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THE ENGINEER'S YEAR-BOOK

OF

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FOR 1949

A COMPENDIUM OF THE MODERN PRACTICE OF

CIVIL, MECHANICAL, ELECTRICAL, MARINE, GAS, AERO, MINE, & METALLURGICAL ENGINEERING

(*Originally compiled by H. R. KEMPE, M.Inst.C.E., M.I.Mech.E., and W. HANNEFORD-SMITH, F.R.S.E., Assoc. Inst.C.E.*)

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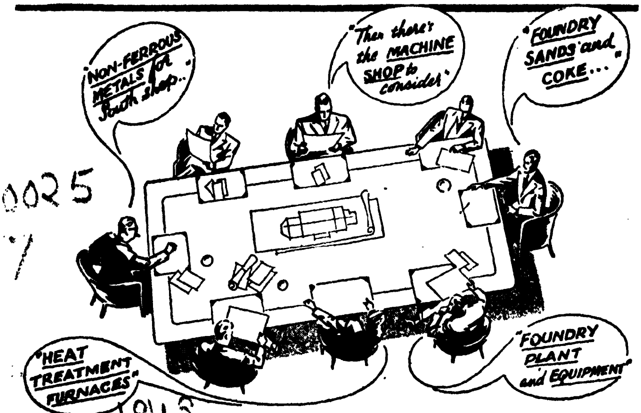

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CONTENTS.

VOLUME TWO

SECTION | PAGE

XXVII. STEAM ENGINEERING.

Part I: PROPERTIES OF STEAM 3

Part II: STEAM GENERATING PLANT 9

Part III: RECIPROCATING STEAM ENGINES 73

Part IV: STEAM TURBINES 103

Part V: STEAM CONDENSERS—AIR PUMPS—FEED PUMPS—COOLING TOWERS—STEAM ACCUMULATORS 167

XXVIII. THE INTERNAL COMBUSTION ENGINE.

Part I: UNITS AND EQUIVALENTS—GAS LAWS, ETC. 187

Part II: GENERAL DESCRIPTION OF TYPES OF ENGINES AND FUELS 197

Part III: DESIGN OF I.C. ENGINES—TORSIONAL VIBRATIONS OF CRANKSHAFTS—SCAVENGING AND SUPERCHARGING—MOUNTINGS—FOUNDATIONS—MAINTENANCE, ETC.—STANDARD SPECIFICATIONS—B.S.I. AND LLOYD'S RULES 215

Part IV: GAS TURBINES 253

XXIX. NAVAL ARCHITECTURE AND MARINE ENGINEERING.

Part I: NAVAL ARCHITECTURE 271

Part II: MARINE ENGINES (STEAM) 287

Part III: MARINE ENGINES (OIL) 305

XXX. AUTOMOBILES AND ELECTRIC ROAD TRACTION 321

XXXI. *Part I:* AIRCRAFT—AERODYNAMICS 381

Part II: AERO ENGINES 445

SECTION		PAGE
XXXII.	RAILWAYS.	
	Part I: PERMANENT WAY	469
	Part II: LOCOMOTIVES AND ROLLING STOCK	515
	Part III: ELECTRIC RAIL TRACTION	585
	Part IV: RAILWAY SIGNALLING	595
XXXIII.	GAS AND GASWORKS	615
XXXIV.	*Part I:* HEATING BY HOT WATER, STEAM, GAS, ETC.—DRYING	659
	Part II: VENTILATION	675
	Part III: LIGHTING	685
	Part IV: FIRE EXTINCTION AND PREVENTION	709
XXXV.	REFRIGERATION AND COLD STORAGE	719
XXXVI.	AIR COMPRESSION	753
XXXVII.	SANITARY ENGINEERING	769
XXXVIII.	*Part I:* CRANES AND CRANE APPLIANCES	801
	Part II: LIFTS AND ESCALATORS	817
	Part III: ROPEWAYS AND CONVEYORS	831
XXXIX.	*Part I:* MINING	847
	Part II: MINERAL VALUATION	941
XL.	EXPLOSIVES	949
XLI.	RADIOCOMMUNICATION	957
XLII.	PAINTS—VARNISHES—ENAMELS—LACQUERS, ETC.	977
XLIII.	PATENTS, DESIGNS AND TRADE MARKS	1001
XLIV.	DEPRECIATION OF PLANT AND MACHINERY—INCOME TAX ALLOWANCES	1017

SECTION PAGE

XLV. Legal Notes for Engineers 1027

XLVI. Mathematics 1039

DESCRIPTIVE SECTIONS 1093

INDEX 1305

Directory of Manufacturers 1379

Index to Advertisers 1403

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SECTION XXVII

STEAM ENGINEERING.

Revised by R. H. Parsons, M.I.Mech.E.

PART I

PROPERTIES OF STEAM . . . . pp. 1-7

PART II

STEAM GENERATING PLANT . . . . pp. 9-71

PART III

RECIPROCATING STEAM ENGINES . . . . . pp. 73-101

PART IV

STEAM TURBINES . . . . pp. 103-166

(Contributed by Sir Henry Guy, C.B.E., D.Sc. F.R.S., M.I.C.E., M.I.Mech.E., and L. S. Robson, M.C., M.Sc. Tech., M.I.Mech.E.)

PART V

STEAM CONDENSERS—AIR PUMPS—FEED PUMPS—COOLING TOWERS—STEAM ACCUMULATORS . . . . . pp. 167-184


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SECTION XXVII

PART I

PROPERTIES OF STEAM—FLOW OF STEAM—POWER PLANT THERMAL EFFICIENCIES.

(Revised by R. H. Parsons, M.I.Mech.E.)

Steam Tables.

Engineers who have to make calculations which involve the properties of steam are recommended to use the Callendar (1939) Steam Tables, published by Messrs. Edward Arnold & Co. London, which cover the range of pressures and temperatures up to 3,200 lb. per sq. in., and 1,000° F. and are recognised as the standard by British manufacturers. The figures in Tables I and II below have been abstracted from the Callendar tables, and are sufficiently complete to enable intermediate values to be obtained with reasonable accuracy by interpolation. Interpolation is best effected by plotting two or three tabulated values on either side of the required value, and drawing a smooth curve through the points so obtained. The required value can then be read from the curve.

Many calculations will be greatly facilitated by the use of a Mollier Diagram, an example of which will be found on page 1305 (Vol. I). A complete diagram, plotted in accordance with the Callendar Tables, is published by Messrs. Edward Arnold, under the title 'The 1939 Heat-Entropy Diagram for Steam.'

Atmospheric pressure in this country is taken as 14·69 lb. per sq. in. At this pressure water boils at 212° F., and has a latent heat of 970·6 B.Th.U. per lb. Gauge pressures, being measured above the atmospheric pressure, are approximately 14·7 lb. less than the absolute pressures for which the tables are computed.

One inch of mercury column at 32° F. is equal to 0·491 lb. per sq. in. This pressure is reduced by one hundredth of 1 per cent. for every extra degree Fahrenheit of the mercury column temperature.

For properties of steam at low pressure, see p. 176.

TABLE I.—PROPERTIES OF SATURATED STEAM (CALLENDAR 1939).

Absolute Pressure. Lb. per Sq. In.	Vacuum or Gauge Pressure.	Temperature. ° F.	Total Heat of Steam B.Th.U.	Heat of Water above 32°.	Volume Cu. Ft. per Lb.	Entropy of Water.	Entropy of Steam.
	(ins. H.G.)						
1	27·96	101·7	1106·0	69·7	334·1	0·1326	1·9779
2	25·91	126·1	1116·4	94·1	173·9	0·1750	1·9202
3	23·87	141·5	1123·0	109·4	119·0	0·2009	1·8872
4	21·83	153·0	1127·5	120·9	90·82	0·2198	1·8628
5	19·79	162·3	1131·3	130·2	73·67	0·2349	1·8445
6	17·75	170·1	1134·4	137·9	62·09	0·2473	1·8296
7	15·70	176·9	1137·2	144·7	53·76	0·2580	1·8170
8	13·66	182·9	1139·7	150·8	47·43	0·2675	1·8062
9	11·62	188·3	1141·9	156·3	42·46	0·2759	1·7966
10	9·58	193·2	1143·8	161·2	38·49	0·2835	1·7883
11	7·54	197·8	1145·6	165·8	35·21	0·2905	1·7805
12	5·49	202·0	1147·2	169·9	32·46	0·2970	1·7736
13	3·45	205·9	1148·8	174·0	30·11	0·3029	1·7673
14	1·41	209·6	1150·3	177·8	28·10	0·3084	1·7613
	(lbs.)						
14·7	0	212·0	1150·7	180·17	26·80	0·3122	1·7572
15	0·31	213·0	1151·6	181·3	26·35	0·3136	1·7557
16	1·31	216·3	1152·8	184·5	24·80	0·3183	1·7504
17	2·31	219·5	1154·0	187·7	23·44	0·3233	1·7454
18	3·31	222·4	1155·0	190·7	22·22	0·3277	1·7408
19	4·31	225·2	1156·0	193·5	21·12	0·3319	1·7366

TABLE I.—PROPERTIES OF SATURATED STEAM (CALLENDAR 1939)—*continued.*

Absolute Pressure. Lbs. per Sq. in.	Gauge Pressure. Lbs. per Sq. in.	Temperature. ° F.	Total Heat of Steam B.Th.U.	Heat of Water above 32°.	Volume Cu. Ft. Per lb.	Entropy of Water.	Entropy of Steam.
20	5·31	228·0	1157·0	196·3	20·13	0·3359	1.7235
22	7·31	233·1	1158·7	201·5	18·41	0·3435	1·7249
24	9·31	237·8	1160·3	206·3	16·97	0·3502	1·7177
26	11·31	242·2	1162·0	210·8	15·74	0·3565	1·7115
28	13·31	246·4	1163·2	214·9	14·68	0·3625	1·7054
30	15·31	250·3	1164·7	218·9	13·75	0·3682	1·6998
40	25·31	267·2	1170·7	236·2	10·51	0·3921	1·6769
50	35·31	280·9	1175·0	250·2	8·520	0·4110	1·6590
60	45·31	292·6	1178·5	262·2	7·180	0·4270	1·6444
70	55·31	302·8	1181·5	272·6	6·212	0·4411	1·6321
80	65·31	311·9	1184·0	282·1	5·480	0·4533	1·6215
90	75·31	320·2	1186·1	290·7	4·902	0·4646	1·6120
100	85·31	327·9	1188·2	298·6	4·438	0·4744	1·6035
110	95·31	334·8	1190·0	305·7	4·050	0·4833	1·5958
120	105·3	341·3	1191·5	312·5	3·732	0·4917	1·5887
130	115·3	347·3	1192·8	318·8	3·458	0·4996	1·5822
140	125·3	353·0	1194·0	324·9	3·222	0·5070	1·5761
150	135·3	358·4	1195·2	330·6	3·015	0·5140	1·5704
160	145·2	363·6	1196·3	336·0	2·834	0·5205	1·5650
170	155·3	368·4	1197·3	341·2	2·675	0·5268	1·5600
180	165·3	373·1	1198·2	346·0	2·532	0·5326	1·5553
190	175·3	377·5	1199·0	350·8	2·404	0·5383	1·5510
200	185·3	381·8	1199·6	355·4	2·288	0·5438	1·5468
220	205·3	389·9	1200·9	364·0	2·087	0·5536	1·5384
240	225·3	397·4	1202·0	372·1	1·920	0·5631	1·5360
260	245·3	404·5	1203·0	379·7	1·777	0·5721	1·5242
280	265·3	411·1	1203·7	386·9	1·654	0·5803	1·5178
300	285·3	417·4	1204·3	393·8	1·545	0·5880	1·5116
320	305·3	423·4	1204·8	400·4	1·451	0·5954	1·5058
340	325·3	429·1	1205·2	406·7	1·366	0·6025	1·5004
360	345·3	434·5	1205·5	412·7	1·292	0·6092	1·4951
380	365·3	439·7	1205·6	418·6	1·224	0·6155	1·4900
400	385·3	444·7	1205·7	424·2	1·163	0·6219	1·4852

TABLE II.

Properties of saturated and superheated steam.

t = temperature in ° F. **H = total heat per lb. in B.Th.U.**

V = volume in cu. ft. per lb. **ϕ = Entropy of steam.**

NOTE.—The first line under each pressure refers to saturated steam.

P = 100 lb. per sq. in. (abs.)				P = 150 lb. per sq. in. (abs.)			
t	H	V	ϕ	t	H	V	ϕ
327·9	1188·2	4·438	1·6035	358·1	1195·2	3·015	1·5704
400	1227·3	4·938	1·6514	400	1219·3	3·228	1·5993
450	1253·3	5·267	1·6806	450	1247·0	3·459	1·6304
500	1278·9	5·588	1·7077	500	1273·8	3·681	1·6588
550	1304·1	5·905	1·7331	550	1299·8	3·899	1·6852
600	1329·1	6·217	1·7571	600	1325·3	4·112	1·7099
650	1353·8	6·525	1·7800	650	1350·6	4·322	1·7335
700	1378·6	6·831	1·8022	700	1375·9	4·530	1·7558
750	1403·6	7·133	1·8234	750	1401·2	4·733	1·7772

TABLE II.—*continued.*

P = 200 lb. per sq. in. (abs.)			
t	H	V	φ
381·8	1199·6	2·288	1·5468
450	1240·3	2·549	1·5917
500	1268·3	2·722	1·6217
550	1295·3	2·891	1·6491
600	1321·6	3·055	1·6717
650	1347·6	3·217	1·6985
700	1373·4	3·376	1·7213
750	1399·1	3·533	1·7430
800	1424·5	3·688	1·7636

P = 300 lb. per sq. in. (abs.)			
t	H	V	φ
417·4	1204·3	1·515	1·5116
450	1225·5	1·640	1·5355
500	1256·9	1·766	1·5692
550	1286·1	1·887	1·5989
600	1314·1	2·002	1·6260
650	1341·3	2·114	1·6511
700	1367·9	2·225	1·6746
750	1394·3	2·333	1·6968
800	1420·4	2·440	1·7179

P = 400 lb. per sq. in. (abs.)			
t	H	V	φ
444·7	1205·7	1·163	1·4852
450	1209·0	1·183	1·4896
500	1244·4	1·286	1·5271
550	1276·3	1·384	1·5592
600	1306·1	1·475	1·5879
650	1334·6	1·564	1·6141
700	1362·2	1·650	1·6385
750	1389·3	1·733	1·6613
800	1416·0	1·815	1·6830

P = 500 lb. per sq. in. (abs.)			
t	H	V	φ
467·1	1205·5	0·930	1·4640
500	1230·4	0·995	1·4904
550	1265·6	1·079	1·5258
600	1297·6	1·157	1·5569
650	1327·6	1·230	1·5845
700	1356·6	1·301	1·6098
750	1384·6	1·371	1·6340
800	1412·2	1·438	1·6564
850	1439·3	1·505	1·6774

P = 600 lb. per sq. in. (abs.)			
t	H	V	φ
486·3	1204·3	0·773	1·4461
500	1215·6	0·799	1·4579
550	1254·4	0·875	1·4970
600	1288·7	0·944	1·5302
650	1320·2	1·009	1·5592
700	1350·2	1·071	1·5856
750	1379·3	1·130	1·6105
800	1407·7	1·189	1·6334
850	1435·5	1·245	1·6549

P = 700 lb. per sq. in. (abs.)			
t	H	V	φ
503·2	1202·2	0·657	1·4305
550	1242·2	0·728	1·4707
600	1279·2	0·791	1·5064
650	1312·5	0·850	1·5371
700	1343·9	0·904	1·5646
750	1376·9	0·958	1·5903
800	1403·2	1·009	1·6137
850	1431·5	1·058	1·6357
900	1459·2	1·106	1·6564

P = 800 lb. per sq. in. (abs.)			
t	H	V	φ
518·3	1199·4	0·569	1·4162
550	1229 0	0 615	1·4458
600	1268·7	0·675	1·4845
650	1304·2	0·729	1·5171
700	1337·3	0·780	1·5458
750	1368·6	0·828	1·5723
800	1398·5	0·873	1·5964
850	1427·5	0·917	1·6188
900	1455·6	0·961	1·6398

P = 900 lb. per sq. in. (abs.)			
t	H	V	φ
532·0	1196·4	0·500	1·4031
550	1214·8	0·526	1·4209
600	1258·7	0·585	1·4637
650	1296·3	0·637	1·4985
700	1330·6	0·684	1·5286
750	1362·7	0·728	1·5556
800	1393·0	0·770	1·5802
850	1422·8	0·809	1·6031
900	1451·6	0·849	1·6248

P = 1,000 lb. per sq. in. (abs.)			
t	H	V	φ
544·6	1192·9	0·445	1·3908
650	1288·0	0·564	1·4810
700	1323·7	0·609	1·5126
750	1356·6	0·649	1·5402
800	1387·7	0·688	1·5653
850	1418·2	0·725	1·5889
900	1447·7	0·760	1·6110
950	1476·1	0·796	1·6317
1,000	1503·9	0·829	1·6511

P = 1,200 lb. per sq. in (abs.)			
t	H	V	φ
567·2	1184·5	0·361	1·3679
650	1268·8	0·449	1·4473
700	1308·9	0·491	1·4822
750	1344·8	0·528	1·5124
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850	1409·6	0·595	1·5640
900	1439·9	0·625	1·5866
950	1469·3	0·655	1·6077
1,000	1497·8	0·685	1·6276

TABLE II.—*continued.*

P = 1,500 lb. per sq. in. (abs.)				P = 2,000 lb. per sq. in. (abs.)			
t	H	V	ϕ	t	H	V	ϕ
596·2	1169·1	0·277	1·3360	635·9	1136·1	0·189	1·2853
650	1237·2	0·332	1·3990	650	1166·0	0·206	1·3113
700	1285·1	0·372	1·4410	700	1237·8	0·248	1·3753
750	1325·8	0·406	1·4754	750	1290·2	0·281	1·4196
800	1362·4	0·435	1·5050	800	1333·4	0·308	1·4550
850	1396·2	0·463	1·5314	850	1371·9	0·331	1·4848
900	1428·3	0·489	1·5552	900	1407·1	0·353	1·5112
950	1458·9	0·514	1·5774	950	1440·6	0·373	1·5352
1,000	1488·5	0·539	1·5981	1,000	1472·3	0·394	1·5575
P = 2,500 lb. per sq. in. (abs.)				**P = 3,000 lb. per sq. in. (abs.)**			
t	H	V	ϕ		H	V	ϕ
668·1	1092·0	0·132	1·232	695·4	1016·4	0·086	1·158
750	1247·2	0·205	1·366	750	1194·4	0·148	1·310
800	1300·4	0·231	1·409	800	1263·4	0·178	1·366
850	1345·1	0·252	1·444	850	1315·9	0·199	1·406
900	1384·6	0·271	1·473	900	1360·2	0·215	1·439
950	1420·7	0·289	1·500	950	1399·6	0·232	1·466
1,000	1454·9	0·306	1·524	1,000	1435·9	0·247	1·493

THE THROTTLING CALORIMETER.

The percentage of moisture carried as vapour in wet steam may be determined experimentally by the use of a throttling calorimeter. This can be made up of ordinary pipe fittings. Steam is taken from the main by a perforated sampling tube, the flow being controlled by a stop-valve. The steam then escapes to atmosphere through a pipe between the flanges of which is fixed a disc pierced with a hole about one eighth of an inch in diameter. Thermometers are inserted in the pipe to measure the temperature of the steam before and after it passes through the orifice in the disc. The steam leaving the orifice at atmospheric pressure should be in the superheated state, and if this is the case the dryness fraction of the steam in its original condition is obtained from the formula:

$$\text{Dryness fraction} = \frac{H + 0{\cdot}47\,(t_2 - t_1) - h}{L}$$

in which H is the total heat per lb. of dry saturated steam at the pressure of the atmosphere, t_1 is the temperature of dry saturated steam at the same pressure, t_2 is the temperature of the superheated steam after passing the orifice, h is the heat in 1 lb. of water at the absolute pressure of the steam in the main, and L is the latent heat of 1 lb. of steam at the absolute pressure in the main.

FLOW OF STEAM FROM AN ORIFICE OR NOZZLE.

Accurate methods of calculating the amount of steam that will flow through an orifice or nozzle under different conditions are given in the Section on Steam Turbines, p. 103. When the absolute pressure on the down-stream side of the nozzle does not exceed 56 per cent. of the absolute initial pressure, the weight of steam that will pass through the nozzle can be found with sufficient accuracy for most practical purposes by the Napier formula:—

$$W = A \times P \div 70$$

in which W is the weight of steam in lbs. per second; A is the cross sectional area of the nozzle in square inches; P is the absolute initial pressure in lbs. per sq. in.

The above formula holds for a well-shaped nozzle. If the orifice is in the form of a short pipe, the calculated flow should be multiplied by 0·93. If the flow is through a hole in a thin plate, or through a safety valve, the calculated flow should be multiplied by 0·63.

QUANTITY OF STEAM REQUIRED TO HEAT WATER.

Let G denote gallons of water to be heated, T and t = temperatures of water after and before heating; H = total heat per lb. of steam (column 4, Table I); and W = lbs. of steam required; then:

$$W = \frac{10 \times G \times (T - t)}{H - T + 32}$$

Note.—One cubic foot of water contains 6·24 gallons. One gallon of water weighs 10 lb.

The Efficiency of Power Stations.

The usual way of expressing the efficiency of a power station is by stating the percentage of the heat in the fuel that is converted into useful work. On this basis the efficiency of the best steam plants in the world is only about 30 per cent., and the great majority are far below this figure. The objection to this method is that it takes no account of what is theoretically possible, having regard to the temperatures and pressures employed, and therefore does not permit of a fair comparison between the performances of different plants, nor does it allow of the comparison of any performance with that of an ideally perfect plant working under the same steam conditions.

The first attempt to meet the difficulty was by using the Rankine Cycle as a standard of comparison. It was employed mainly for the comparison of engine and turbine efficiencies, but it applies, of course, equally to a complete power station, for in the ideal power station the boiler room losses would be nil. The Rankine Cycle is dealt with in the section on Thermodynamics, p. 1299 (Vol. I).

The defect of the Rankine Cycle is that it assumes the feed water to be heated entirely by heat supplied directly from the fuel. This was formerly in accordance with practice, but in all modern stations the feed water is raised regeneratively to a high temperature before entering the boiler, by means of partially expanded steam tapped from the turbine at successively higher pressures. The calculation of the theoretical efficiency of an ideal steam plant working with regenerative feed heating is a very simple matter, provided that the feed-heating is assumed to take place in an infinite number of stages, instead of in three or four stages as commonly adopted in practice. This assumption, moreover, is essential to the conception of an ideal plant, for otherwise there would have to be an irreversible temperature drop at each stage, and this is inconsistent with the notion of a theoretically perfect operation. It follows, then, that all calculations of an ideal cycle that are based on the assumption of a finite number of stage heatings are illogical and unsound, because it would always be possible to obtain a higher cycle efficiency with the same steam conditions, and this is, by hypothesis, impossible. The point is an important one, because attempts are frequently made to calculate cycle efficiencies by taking into account the number of feed-heating stages.

With regenerative feed-heating, the theoretical maximum amount of heat that could be turned into work per pound of steam is given by the expression:

$$\mathrm{U} = \mathrm{H} - h - \mathrm{T}(\phi_1 - \phi_2)$$

in which H represents the total heat of a pound of steam at the upper pressure and temperature, h is the total heat in a pound of water at the final feed temperature; T is the vacuum temperature; ϕ_1 is the entropy of steam at the upper pressure and temperature, and ϕ_2 is the entropy of the final feed water.

The cycle efficiency, expressed as a fraction, is therefore:

$$\text{Cycle efficiency} = \frac{\mathrm{U}}{\mathrm{H} - h}$$

and the performance of the station, expressed as a percentage of its cycle efficiency, which is the true figure of operating merit, is:

$$\frac{3{,}412 \times 100}{\text{Cycle efficiency} \times \text{actual heat consumption per K.W.H.}}$$

The cycle efficiency of modern British power stations is about 44·0 per cent., and the maximum figure so far attained with the highest steam pressures and temperatures yet employed in practice is slightly over 50 per cent. The progress made in steam conditions since 1914 may be illustrated by the fact that at that time a cycle efficiency of less than 30 per cent. was considered good.

SECTION XXVII

PART II

STEAM GENERATING PLANT.

BOILER DESIGN—COMBUSTION—TRANSMISSION OF HEAT IN BOILERS — BOILER EFFICIENCY — BOILER SHELLS — STANDARD CONDITIONS FOR THE DESIGN AND CONSTRUCTION OF MARINE BOILERS — BOILER TUBES — BOILER SETTING — CHIMNEYS AND FLUES—SUPER-HEATERS — SCALE AND CORROSION — FEED-WATER SOFTENING — FEED HEATERS — ECONOMISERS — AIR HEATERS — INSPECTION AND LEGISLATION — BOILER FITTINGS — INJECTORS — SAFETY VALVES — STEAM PIPES—STEAM TRAPS.

(Revised by R. H. Parsons, M.I.Mech.E.)

General.

Commercial steam boilers fall into three main classes: (*a*) the so-called 'tank' boilers, of which the Lancashire, Cornish and Scotch marine are the best known examples; (*b*) locomotive boilers, and (*c*) water-tube boilers such as the Babcock and Wilcox, Stirling, Yarrow, Thornycroft, etc.

Boilers of the Lancashire and Cornish types are very largely used for factory work. They are simple in construction, free from breakdowns, cost little for maintenance, and are easily operated by semi-skilled labour. They are probably the best boilers to use with impure feed water. Another good feature is their large water capacity which renders them immediately responsive to sudden increases on the demand for steam. Their size and weight, however, are disadvantageous for transport, and they are rarely built for pressures of more than about 250 lb. per sq. in. They are practically always used in conjunction with economisers, and then provide a reliable and economical steam raising plant.

The Scotch marine boiler, which has many of the advantages and disadvantages of the Lancashire boiler, is almost exclusively used in mercantile steamers except where the demand for higher pressures has led to the introduction of the water-tube boiler.

The locomotive type of boiler is rarely used for stationary duties, as other types are more efficient, and it is badly adapted for buring low grade fuels, but for purposes where its self-contained nature and range of steaming are important, it is unsurpassed.

For power station work, and whenever really high pressure steam is required in large or small quantities, the water-tube type of boiler is supreme. Single water-tube boilers are now being built with an evaporative capacity of more than a million pounds of water per hour, and units capable of evaporating several hundreds of thousands of pounds per hour are common. There is also no difficulty in building such boilers for the highest commercial pressures, 1,400 lb. per sq. in. being frequently employed nowadays in central station practice.

The Rating of Boilers.

It was formerly the practice to rate boilers according to 'boiler horse-power' and the term is still met with in American publications. The assumptions were made that 34 lb. of steam were required to develop one horse-power in an engine, and that a boiler would evaporate about 3·4 lb. of water per sq. ft. of heating surface. Hence, a boiler with a heating surface of 5,000 sq. ft. was called a 500 h.p. boiler, and so on. The term boiler horse-power is happily becoming obsolete as it has no useful meaning under present conditions.

The proper rating of a boiler is the number of pounds of steam it will produce continuously per hour at a stated pressure and temperature from feed water at a stated temperature, when steaming under its most economical conditions. It is usual also to specify the maximum continuous output of steam obtainable by forcing the boiler.

Evaporation rates of over 15 lb. per sq. ft. per hour (including the economiser surface) are now common, and in the tubes exposed directly to the radiant heat of the furnace, the rate may exceed 100 lb. per sq. ft. of exposed surface per hour.

The output of a boiler will naturally depend upon the rate at which fuel can be burnt in the furnace. In the case of solid fuels this is limited by the grate area and the draught available. For stationary boilers bituminous coal can be economically burnt at a rate of about 25 lb. per hour per sq. ft. of grate surface, though this figure may be greatly exceeded if ample draught is available. For fuels of all kinds the volume of the combustion chamber is important, and in large power station boilers it is customary to arrange for a heat liberation of 15,000 to 50,000 B.Th.U. per cub. ft. of volume per hour, a figure of 20,000 B.Th.U. per cub. ft. being common practice.

Typical proportions of heating surfaces, grate areas and combustion chamber volumes for large power station boilers are given in the two following tables.

Power Station	Hackney	Kirkstall	Deptford West	Battersea
Maker of boiler	Simon-Carves	Stirling	Thompson	Babcock & Wilcox
Normal evaporation (lb. per hour)	125,000	160,000	160,000	440,000
Maximum evaporation (lb. per hour)	150,000	184,000	200,000	550,000
Firing system	Chain-grate stoker	Pulv. fuel	Retort stoker	Retort stoker
Grate area (sq. ft.)	455	—	515	787
Volume of combustion chamber (cub. ft.)	—	11,100	11,600	—
Pressure (lb. per sq. in.)	400	490	375	1,420
Temperature of steam (°F.)	800	750	780	965
Heating surface:				
Boiler (sq. ft.)	18,635	16,540	21,930	18,603
Superheater (sq. ft.)	5,500	4,850	5,300	15,540
Economiser ,,	9,604	6,480	12,870	20,713
Air heater ,,	20,730	8,270 (air) 5,720 (gas)	32,085	170,000
Water walls ,,	—	—	1,683	4,737

MODERN AMERICAN HIGH PRESSURE BOILERS.

Power Station	Logan	Waterside No. 2	West End	Miller's Ford
Location	West. Va.	New York	Cincinnati	Dayton
Operation begun	1937	1937	1937	1937
Make of boiler	C.E. Co.	C.E. Co.	B. & W.	B. & W.
Normal evaporation (lbs. per hour)	1,000,000	500,000	350,000	375,000
Firing system	P. coal	P. coal	P. coal	P. coal
Pressure (lbs. per sq. in.)	1,325	1,325	1,275	1,250
Temperature of steam (° F.)	925	900	925	900
Heating surface (sq. ft.):				
Boiler proper	20,800	7,008	7,176	3,367
Water walls and bottom	12,260	4,800	1,878	2,781
Superheater	24,720	14,000	7,357	10,460
Economiser	25,850	18,850	15,191	21,000
Air heater	90,400	65,400	34,800	26,720
Furnace volume	41,000	20,300	4,750	12,100
Heat release in furnace (B.Th.U. per cub. ft.)	29,800	31,000	48,000	39,200

Proportions and Duty of Lancashire Boilers.

The Lancashire boiler has been in use for more than 100 years and is still a favourite type for many industrial purposes. Its very large water capacity enables it to respond to sudden short and erratic demands for steam without loss of pressure; its operation and maintenance are easy and well understood, while it is more tolerant of impure feed water than boilers of the water tube type. Lancashire boilers are constructed in sizes from 6 ft. diameter by 18 ft. long, to 10 ft. diameter by 30 ft. long, and when hand-fired with bituminous coal will give evaporations from 3,000 lb. per hour for the smallest size, to 12,500 lb. per hour for the largest. Although a large proportion of the boilers are fired by hand, there are on the market several types of mechanical stokers suitable for them which include forced draught equipment, and with these an increase of about 20 per cent. of output can be obtained. The usual working pressure is 160 lb. per sq. in., 200 lb. being rarely exceeded, but modern methods of construction enable Lancashire boilers to be built for pressures up to 300 lb. per sq. in.

The typical Lancashire boiler has two flues, each fitted with from two to five cross tubes, depending on the size of the boiler. The diameter of the flues is generally 9 in. less than the radius of the boiler shell. The ends are generally flanged for riveting directly to the shell plates, and for higher pressures they are dished to a radius equal to the diameter of the shell. The width of each grate may be taken as equal to the diameter of the flue, and the length is about twice the width. The maximum practical length of grate for hand firing is 7 ft., and 6 ft. is generally considered to be quite long enough.

The heating surface of a Lancashire boiler is given approximately by the formula:

$$\text{Heating surface in sq. ft.} = 4 \times \text{length in ft.} \times \text{diameter in ft.}$$

The rate of evaporation is usually reckoned as 6 lb. of water per sq. ft. of heating surface per hour for boilers without economisers, or 7 lb. per sq. ft. per hour if economisers are used. The weight of coal burnt per sq. ft. of grate area will vary from 20 lb. to 30 lb. per sq. ft., according to the draught, etc., but for economical working a rate of 25 lb. per sq. ft. per hour should not be exceeded.

The foundation of the brickwork for the boiler setting should be a solid raft of concrete never less than 12 in. thick, as the boilers are heavy, a Lancashire boiler 8 ft. diameter by 30 ft. long, for a working pressure of 160 lb. per sq. in. weighing about 28·5 tons without the water.

Setting of Lancashire Boilers.

In ordinary practice, as developed by long experience, a Lancashire boiler is set upon two longitudinal walls, parallel to one another, and distant apart half the diameter of the boiler (see figs. 1 and 2). On these walls are laid specially formed seating blocks of firebrick. These blocks should not be less than 9 in. deep and the seating edge should not exceed 3 in. in width. The edge is often made convex in profile so as to touch the boiler plate along a line only. Where the ring seams cross the seating blocks, a rectangular gap should be cut across the seating for inspection purposes. These gaps are closed by special wedge pieces, which should be carefully replaced after an inspection. Seating blocks provided with asbestos cushions to take the weight of the boiler, and to provide a resilient gas-tight packing are supplied by Messrs. Pearson of Stourbridge. Asbestos cushions are also recommended to form the joints between the flue covers and the shell plates, as air leakage is very liable to occur along these joints. The side flues are generally

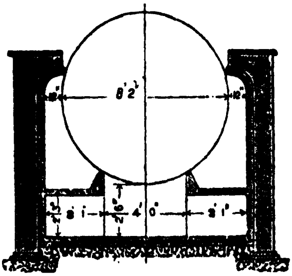

Fig. 1.

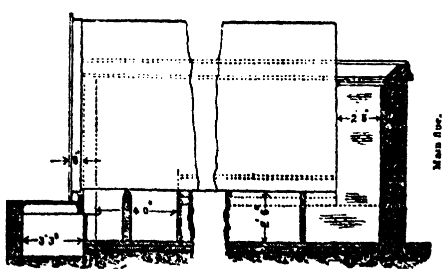

Fig. 2.

made 12 ins. wide at the horizontal diameter of the boiler, and the dividing walls between the flues of adjacent boilers may be 18 ins. thick, so that the distance from centre to centre of boilers in a battery is equal to the boiler diameter plus 3 ft. 6 ins.

The gases leaving the boiler are usually returned to the front end by a flue centrally beneath the boiler, and then returned to the back end by two side flues, from which they pass into the main flue by two separate dampers. Sometimes, however, the gases return to the front end by the side flues, from which they flow into the bottom flue and so to the stack. This latter type of setting may be condemned on theoretical grounds, but it would not be advocated by leading boiler makers were there any real objection to it. The superheater, when used, is placed at the back of the boiler in the path of the gases as they issue from the boiler flues. The economiser is installed separately in the main flue between the boiler and the chimney, where it will serve a number of boilers. The standard type of economiser for Lancashire boilers is constructed of a number of vertical cast-iron pipes assembled in sections, each pipe having a heating surface of 10 sq. ft. As a rule the aggregate economiser surface is equal to the total heating surface of the boilers served. The economiser pipes are usually longer than the height of the flue. In this case the economiser should be set with its lower end level with the bottom of the incoming flue, and its upper end level with the top of the out-going flue. The gases which traverse it will therefore rise during their passage through it, and there will be no gas pocket at either the top or bottom of the economiser. A gas pocket at the top may give cause for a gas explosion, while one at the bottom conduces to the sweating of the pipes.

The flues should be lined with 4½ ins. of firebrick set in fireclay and bonded into the red brick-work at every fourth course. Bricks set in fireclay should be laid with the least possible thickness of clay between them. The clay should be used in a fluid state, and the bricks thoroughly wetted with it and then rubbed into place. Fireclay alone has no 'body' and eventually crumbles away if used in any thickness. A satisfactory fireclay mortar may, however, be made by mixing one part by weight of fireclay with two parts of fire brick dust and a suitable quantity of water. The dust should consist of firebrick crushed to the size of a pea and smaller. Such mortar forms a refractory binding under the action of heat, and is about as strong as the brick itself.

The boiler must be set with a fall towards the front end of half an inch for every 10 ft. of length. The firing floor should be 2 ins. below the lowermost point of the boiler front. The side flues should be not less than 12 ins. wide at the centre line of the boiler, and the underside of the flue covers should be 3 ins. above the crown of the internal flues. The side walls of the end boilers of a range should be not less than 18 ins. thick, and should preferably be faced with white glazed bricks. Access doors and dampers should be arranged so that flue dust may be cleaned out while the boiler is under steam. There should be no sharp corners nor abrupt turns in the main flue, as these may seriously affect the draught.

Dampers.

A Lancashire boiler has two dampers, one at the end of each of the side flues. These dampers may either consist of iron plates sliding in vertical iron guides, or may be of the butterfly type rotating about a vertical spindle. The plate dampers are counterbalanced by weights connected to the dampers by chains passing over guide pulleys, and arranged for operation from the boiler front. The objection to this type of damper is that gas pockets may be formed when it is partly open. Plate dampers usually give a clear opening 12 ins. wide. The opening should be 4 ft. high for boilers up to 7 ft. in diameter, and 6 ft. high for boilers of 9 ft. diameter and over, the height of opening being roughly two-thirds the diameter of the boiler.

Leaky Boiler Settings.

It is impossible to operate a Lancashire boiler efficiently unless the setting is kept in first class condition, for any leakage of air inwards serves both to carry away heat up the chimney and to spoil the draught. For the external surfaces of brickwork settings, white glazed bricks are unsurpassed, as they are not only impervious to air but contribute notably to the lightness and clean appearance of the boiler house. Ordinary unglazed brickwork is very porous and should always be given two or three good coats of paint or tar, to prevent the infiltration of air. Every crack that appears in a setting should be attended to at once, even though it may not appear serious. Leakage can be detected by holding a lighted candle close to the suspected place, when the flame will be drawn towards it if the air is entering in appreciable quantities. If any part of the setting, especially in the neighbourhood of a crack, feels cool to the hand, it is a fairly certain sign of the entrance of air, and any local accumulation of dust is also suspicious. It is difficult to make a satisfactory job of repairing cracks with mortar or cement because these substances shrink and get loose. It is better to fill up the cracks by caulking them tightly with some fibrous material such as asbestos string or rope, or cotton waste saturated with a slurry of fireclay.

Air leakage has to be particularly guarded against at the joints where the flue covers and down-take covers meet the shell. Asbestos packing makes a good joint at these places. Air leakage round the head of the superheater, even when this is covered by a supposedly air-tight casing, is always to be feared, and the possibility of air entering the flue system through the blow-down pit must not be overlooked. The expansion of the boiler, which may amount to as much as three-quarters of an inch in its length, makes the back end very difficult to keep air and gas-

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tight, and there are several special fittings provided for this duty. It must always be remembered that air will leak into the boiler settings and flues at every possible place, and no trouble is too great to keep it out.

SUITABLE FLUE AND CHIMNEY SIZES FOR BATTERIES OF LANCASHIRE BOILERS.
(Edwin Danks & Co., Ltd.)

Dia. of Boiler.	Number of Working Boilers in Battery .	1	2	4	6	8	10
Ft. In.		Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft. In.
	Width of flue . .	2 0	3 0	4 0	5 0	6 0	7 0
7 0	Height of flue . .	5 0	5 6	7 0	8 0	8 6	8 6
	Diameter of chimney .	3 0	4 1	5 3	6 3	7 3	7 9
	Width of flue . .	2 0	3 0	4 0	5 0	6 6	7 6
7 6	Height of flue . .	5 6	6 0	7 6	8 6	8 6	8 6
	Diameter of chimney .	3 3	4 3	5 6	6 6	7 6	8 3
	Width of flue . .	2 0	3 0	4 0	5 6	7 0	8 0
8 0	Height of flue . .	5 6	6 6	8 0	8 6	8 6	9 0
	Diameter of chimney .	3 4	4 6	5 9	6 9	7 9	8 6
	Width of flue . .	2 0	3 0	4 6	5 6	7 0	8 6
8 6	Height of flue . .	6 9	7 0	8 0	9 0	9 0	9 0
	Diameter of chimney .	3 6	4 8	6 0	7 0	8 0	8 9
	Width of flue . .	2 0	3 0	4 6	6 0	7 6	9
9 0	Height of flue . .	6 6	7 6	8 0	9 0	9 0	9
	Diameter of chimney .	3 7	4 9	6 2	7 3	8 3	9

Note: The flue dimensions are for flat-topped rectangular flues.
The chimney diameters are the smallest internal diameter.
For single Cornish boilers, etc., the flue area should equal one-third the grate area.
For single Cornish boilers, etc., the chimney area should equal one-quarter the grate area.

CORNISH BOILERS.

The Cornish boiler is very similar to the Lancashire boiler, the chief difference being that it has only one furnace and internal flue. It possesses all the advantages of the Lancashire boiler as regards steam reserve, low cost of maintenance and tolerance of impure feed water, and is specially suitable for factories requiring only a comparatively small amount of steam. Cornish boilers up to 6 ft. 6 in. diameter by 24 ft. long with an evaporation of 3,160 lb. of steam per hour are obtainable.

VERTICAL BOILERS.

Vertical boilers are characterised by a vertical cylindrical shell containing an internal fire-box. The shell is usually anything from about 3 ft. diameter by 7 ft. high, to 7 ft. diameter by 14 ft. high, though larger sizes are made. The usual working pressure is about 100 lb. per sq. inch and the evaporation ranges from about 250 to 2,500 lb. of steam per hour.

There are two main types, one employing fire-tubes and the other water-tubes. Of the fire-tube boilers, the Cochran is characterised by a hemispherical fire-box from the back of which the gases rise to a dry-back combustion chamber inside the boiler shell, and pass thence through a number of small horizontal fire-tubes to the up-take at the front. The Cradley boiler is generally similar, except that the combustion chamber is surrounded by water on all sides. In another common design, from 50 to 100 small fire-tubes rise from the flat crown of the fire-box, and are expanded into a horizontal plate at the top of the boiler. A steel chimney rises centrally from the smoke-box above.

In the water-tube type, the fire-box is extended upwards, and its upper part is traversed either by two or three large water tubes, or by a large number of smaller tubes, according to the design. A central chimney rises through the steam space and tends to dry the steam.

A great economy is not to be expected from any of these boilers, though the absence of air-leakage is a point in their favour. The more important consideration is the facility that may be afforded for cleaning the tubes inside and out, and the ease with which necessary repairs may be effected.

THE ECONOMIC BOILER.

The so-called 'Economic' boiler is the land counterpart of the Scotch marine boiler. It is a short cylindrical boiler, fitted generally with two furnaces and internal flues from which the gases pass into an external brick-lined combustion chamber forming an extension of the boiler shell at

the back whence they return to the uptake at the front of the boiler through a large number of small tubes. The boiler is thus entirely self-contained and needs no brick work setting. It is supported by cast iron stools at the front and back. The brick lined combustion chamber becomes red-hot when the boiler is working, and facilitates the combustion of all smoke and volatile matter in the gases. Another good feature of the boiler is the absence of flues, and therefore the avoidance of the air leakage which is so detrimental to the efficiency of brick-set boilers.

Economic boilers are particularly favoured for factories where the demand for steam is not too great, and where boiler house space is limited. They should be supplied with softened water, and the tubes brushed out every two or three days. They are constructed in numerous sizes ranging from 11 ft. diameter by 9 ft. 6 ins. long with an evaporative capacity of 2,500 lb. per hour to 11 ft. diameter by 16 ft. long, with an evaporation of 16,000 lb. per hour. Firing can be done either by hand or by one of the mechanical stokers suitable for the furnaces of Lancashire boilers.

Large Water Tube Boilers.

For power station work the water tube boiler holds the field alone. The forms in which such boilers are constructed are so numerous and the variety of arrangements is so great that it is impossible to do more than touch upon the salient features of design. In modern British power stations steam conditions range from 375 lb. pressure at 730° F. to 1,420 lb. pressure at 965° F. at Battersea, and 2,000 lb. pressure at 940° F. at Brimsdown. A pressure of 600 lb. per sq. in. at 800° F. is characteristic of the majority of the modern stations. Outputs of single boilers range up to 550,000 lb. of steam per hour, though boilers with a capacity of over a million pounds of steam per hour are in service in the United States.

The size of modern boilers is a consequence of the increase in the capacity of steam turbines, and of the tendency to reduce the number of boilers per turbine. In America power stations have been built with one boiler per turbine, but, according to Sir Leonard Pearce (*Inst. Mech. Eng.* 1940) a reasonable compromise is to provide two boilers per turbine, these being of such a capacity that 65 per cent. of the full load of the turbine can be carried continuously by one boiler only. The dimensions of modern boilers have increased almost in proportion to their capacity, largely because of the size of furnaces now employed. The Battersea boilers mentioned above require a total head room of 160 ft. measured from the basement floor, this height including the room occupied by the air heaters and fans situated above the boiler proper.

Water tube boilers fall broadly into two classes, according to whether the tubes are straight or curved. The Babcock and Wilcox, Yarrow and Thompson boilers are designed with straight tubes, while the Stirling is typical of the many types of bent tube boilers. The advantage claimed for straight tubes is the greater facility they afford for inspection and cleaning, but with the care now taken to exclude scale forming substances from the feed water, the straightness of the tubes has practically lost its importance.

Boiler drums for Babcock and Stirling boilers range from 36 in. to 48 in. diameter. For pressures above 450 lb. per sq. in., and outputs exceeding 150,000 lb. of steam per hour they are of seamless construction, manufactured either by forging from the solid or by welding from rolled plate. In the latter case the perfection of the weld is confirmed by X-ray examination.

Large furnaces are now frequently water-cooled on all sides. The cooling arrangement consists of a number of parallel tubes forming part of the water circulation system of the boiler, and built into the furnace wall. The Bailey furnace wall is built up of cast iron blocks clamped to the tubes in such a way as to present an unbroken surface towards the fire. The blocks may have smooth or chequered faces, or they may be faced with a layer of refractory material, according to the conditions of use. Asbestos rope packed between the blocks allows for expansion. The tubes are backed with plastic magnesia covered by a thin coating of hard setting cement, the complete wall being 9 ins. thick from face to face. A feature of modern furnaces is the tendency to eliminate all brickwork exposed to the fire, and to adopt dimensions corresponding to a liberation of 20,000 to 30,000 B.Th.U. per hour per cub. ft. of furnace volume. Under such conditions excellent combustion can be obtained, while the maintenance costs of the furnace are very low.

A few years ago it looked as if all power station boilers except those of quite moderate size would be fired by pulverised coal, because of the limitations of mechanical stoking. Stoking machinery, however, has been greatly improved and developed, and is now used for boilers evaporating more than 500,000 lb. of water per hour. Heavy duty chain grate stokers will burn on an average 55 lb. of coal per sq. ft. per hour, and can be operated up to 80 lb. They have been built with grate surfaces as large as 756 sq. ft. Retort type stokers, being of unit construction, can be built up to any size required, the number of retorts being only limited by the furnace width. Each retort will burn from 1,700 to 1,800 lb. of coal per hour, and the grate area of a large stoker may extend to nearly 800 sq. ft. Ashes from mechanical stokers are disposed of by dumping them from the furnace ash-pit, either into trucks or preferably into an ash-sluice whence they are washed away by a rapid stream of water into a settling pit.

When coal is burnt in a pulverised form, it is usually ground to such a fineness that 65 per cent, of it will pass through a sieve with 200 meshes per linear inch. Finer grinding than this is not worth the extra expense. Grinding is effected by ball mills, roller mills, or impact mills. The power consumption may be from about 12 to 21 kW.h. per ton, at the rated capacity of the mill, the figure depending naturally on the class of coal and the kind of mill. Whatever type of mill is used, the wear of the internal parts is a considerable item of expense.

In most of the earlier pulverised fuel installations, the coal was first dried by hot flue gases or otherwise, then pulverised and delivered into a storage bin, whence it was fed to the furnaces as required. Modern practice is to eliminate both drying and storage, the coal being delivered straight from the pulverisers to the furnace. Modern burners are designed to promote the utmost turbulence of the entering coal and air, so as to give rapid combustion with a short flame. Such burners are constructed in sizes that will liberate more than 100,000,000 B.Th.U. per burner per hour.

Furnaces for pulverised coal are either of the dry-bottom or wet-bottom type. In the former there is sometimes a screen of widely spaced water tubes across the lower part of the furnace, which chill the molten ash and cause it to be deposited as dust. The ashes fall into a hopper-shaped bottom, which may have water-cooled sides, and are discharged at intervals in a dry state. When the ash is of a very fusible kind, it may be allowed to run down the furnace walls and collect in a liquid state in the bottom of the furnace. It is kept in a molten condition by the action of the burners which are directed downwards on to it. The melted slag is run off at intervals, and the molten stream is chilled and broken up by jets of water at high pressure, to be finally carried away by a water sluice.

Forced Circulation Boilers

The water in boilers working at moderate pressures is kept in rapid circulation mainly by the action of the rising bubbles. At a working pressure of 200 lb. per sq. in. abs. the density of the water is about 125 times that of the steam in the bubbles, so that the buoyancy of the latter has a powerful effect. At 1,200 lb. pressure the ratio is reduced to less than 16, and the size of the bubbles is also greatly reduced. It is evident, therefore, that at high pressures the forces causing natural circulation are much diminished, and in view of this fact, several types of boilers have been designed for the forced circulation of water through the tubes. The adoption of this principle not only ensures an adequate water velocity over all parts of the heating surface, but enables smaller and therefore thinner tubes to be used for high pressure boilers. It is also claimed that forced circulation tends to prevent deposits in the tubes.

In the La Mont boiler, water is withdrawn from the drum by a pump which forces it through the tube system and back to the drum, the quantity of water thus circulated being always about eight times the maximum continuous steaming capacity of the boiler. In the Velox boiler a similar procedure is carried out, but the rate of circulation is considerably higher.

The Benson, Sulzer and Ramsin boilers may be classed as forced circulation boilers, although the water in them has no circulation in the proper sense of the term. It is merely forced once through a long pipe system by the feed pump, coming out as steam at the far end. These boilers have no steam drums, and water must be fed to them at a rate exactly equal to the demand for steam.

The Loeffler boiler is characterised by an external drum in which water is evaporated by direct admixture with superheated steam. Saturated steam formed by the evaporation of the water, is withdrawn from the drum by a pump which forces it through tubes which constitute radiant heat and convection superheaters. The amount of steam thus pumped is about three times that of the actual output of the boiler. One-third of the superheated steam is taken away as useful output, and the remaining two-thirds are discharged into the drum to evaporate more water. The Loeffler boiler can only be used for pressures above about 1,700 lb. per sq. in. because of the excessive power that would be absorbed in pumping steam at lower pressures and therefore of greater volume. The Sulzer, Ramsin and Benson boilers all work at 1,400 lb. or more, the Benson boiler being originally designed to work at the critical pressure of about 3,200 lb. per sq. in., though now used for lower pressures.

The most important forced circulation boilers in Great Britain are three of the La Mont type, with an output each of 150,000 lb. per hr. at 1,400 lb., 960° F., at the Willesden Station, London, and one of 350,000 lb. capacity at 350 lb. pressure at the Deptford West Station, London. There are two Sulzer boilers with an evaporation of 130,000 lb. per hour at 1,565 lb., 770° F. in the Powell Duffryn Collieries, and three more, somewhat smaller, at a Warrington Paper Mill. There are four Loeffler boilers at the Brimsdown Station, London, the largest pair having a maximum rating of 250,000 lb. per hour, at 2,000 lb. pressure and 940° F.

Automatic Boiler Control.

Quite a number of systems have been devised for the purpose of automatically controlling the operation of large boilers, and thus relieving the attendant of the necessity of continual intervention. The general problem is to regulate the supply of both fuel and air in conformity with the variation in the demand for steam. The water supply is always taken care of by an automatic feed water regulator which keeps the water level constant in the drum at all loads. Since it is an essential feature of good operation that the steam pressure should be kept constant, it is usual to arrange for slight changes in the steam pressure to bring about what other changes are necessary to restore the normal pressure. Alternatively, the changes of the rate of steam flow from the boiler may be used, to actuate some form of relay mechanism which will cause the speed of the mechanical stokers, or of the feeders of the pulverised coal mills to be raised or lowered, and the speed of the draught fans likewise to be adjusted to the new conditions. The superheat temperature is also subjected to automatic control, usually by by-passing more or less of the hot gases.

Some of these automatic systems are very elaborate, and all have to be so arranged that their action may be over-ridden by the attendant in case of necessity. The apparatus also needs skilled supervision, in order that it shall be maintained in a reliable condition. It seems, however, to be demonstrated that automatic control can be productive of an efficiency from 1 to 3 per cent. higher, over considerable periods, than is obtainable by ordinary manual control, while it relieves the operators of the continual interference that would otherwise be necessary to maintain the most efficient combustion conditions at all loads.

Balanced Draught.

When stoker-fired boilers are working with forced and induced draught the air may be delivered under the grates at a pressure of several inches of water, and the pressure in the furnace might therefore exceed that of the atmosphere unless the power of the induced draught fans is sufficiently great. To avoid the escape of furnace gases into the boiler-house it is necessary to keep the furnace pressure slightly below atmospheric, and a suction of from 0·1 to 0·2 in. w.g. is generally aimed at. The object of a balanced draught system is to maintain this furnace pressure automatically constant under all conditions of load. In the application of the system, any variation of the furnace pressure is made to actuate a relay which, by electrical or other means, alters the speed of the induced draught fans or operates dampers in the induced draught circuit so as to restore the desired furnace conditions. The supply of air to the furnace by means of the forced draught fans may therefore be regulated, by hand, to suit the load on the boiler without the necessity for any simultaneous attention to the induced draught. With a fully automatic system of boiler control, both the forced and induced draught fans are regulated in accordance with the load on the boiler and in such a manner that the draught is always balanced.

Superheaters.

Steam leaving a boiler drum is in the saturated condition, and is almost always passed through a superheater before being used in a turbine or engine. In modern power stations the steam may be raised to a temperature as high as 960° F., though from 800° to 900° F. is more usual. A limit to the temperature is imposed by the metal of the superheater, the tubes of which are made of molybdenum steel for steam temperatures of more than 750° F. When the steam is to be used in a reciprocating engine, the practical limit of temperature is 450°–500° F., as lubrication presents difficulties with higher temperatures.

Boilers of the Lancashire type have the superheaters placed in the downtake flue at the back. Water-tube boilers are usually built with integral superheaters located above or between the tube banks, at some point where the temperature is about 300° F. above the desired final temperature of the steam. In locomotive boilers superheater tubes are placed inside the boiler tubes, and the same practice may be followed in boilers of the Scotch marine type.

Superheater tubes are generally about 1·5 ins. external diameter, bent into a hairpin or a zigzag shape with the ends terminating in headers. The length of each tube, and the number of tubes in parallel are determined by the steam temperature required. For temperatures of more than 750° F. the tubes are made of alloy steel, usually molybdenum. As a further safeguard against temperature difficulties with the metal, the saturated steam enters the hottest portion of the superheater, instead of flowing from the coldest to the hottest end, although the necessary surface is somewhat increased by this arrangement. A rapid steam velocity in the tubes is also adopted, and it is usual to ensure this by allowing a pressure drop equal to about 3 to 5 per cent. of the boiler pressure, although a drop up to 15 per cent. is occasionally allowed.

Superheaters that absorb heat from the gases by convection tend to give a superheat which increases with the output of steam. This characteristic tends to give an excessive steam temperature at high loads. To correct this tendency, a portion of the superheater surface is sometimes placed in the furnace where it is exposed to radiant heat only. Since the furnace temperature does not vary greatly at different loads on the boiler, the temperature of the steam leaving the radiant heat superheater tends to fall at high loads because of the greater steam flow. The two parts of the superheater are connected in series with each other, the steam passing first through the radiant heat portion which therefore has the greatest advantage from its cooling effect. This combination of the radiant heat and convection principles results in the production of a fairly constant degree of superheat at all loads, which is a very desirable achievement.

Other methods of providing for a constant superheat temperature consist in by-passing a part of the flue gases by means of a damper, so that the whole quantity does not pass through the superheater at times of heavy load. Alternatively, a de-superheater may be installed. The most usual method of de-superheating consists of spraying a sufficient quantity of water into the superheated steam to lower its temperature to the required degree. The control of the water supply to the de-superheater is effected by a thermostatic device.

It was once considered necessary to flood superheaters during the period when boilers were getting ready to go on load, but this practice has been generally abandoned. Sufficient protection against burning the superheater tubes is obtained by opening the drain and thus allowing a small amount of steam to pass through the tubes and thus prevent overheating.

Each superheater must be provided with a safety valve, set to blow before the safety valve on the boiler drum, after allowing for the pressure drop through the superheater.


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The minimum velocity of steam through superheater tubes should not be less than 40 ft. per second, and the maximum will be determined by the permissible pressure drop. The rate of heat transmission may be anything from 4,000 to 14,000 B.Th.U. per sq. ft. per hour under average operating conditions, with a maximum of 20,000 B.Th.U. Higher rates are employed with superheaters of the radiant type. For such apparatus, tubes with walls ⅜ in. thick have worked satisfactorily with heat transmission rates of 45,000 B.Th.U. per sq. ft. per hour, and thin tubes have been used to transmit heat at the rate of 60,000 B.Th.U. per sq. ft. per hour and over.

According to Gaffert (*Steam Power Stations*, 1940), convection superheaters can be designed on the assumption that the value of K, *i.e.* the rate of heat transmission from gas to steam in B.Th.U. per sq. ft. per hour per °F., will range from 6 to 10 according to the load on the boiler. For radiant heat superheaters the same authority gives a formula which may be written as:

$$K = 12 \cdot 9 \left(\frac{C}{d}\right)^{0.214} \times (MS)^{0.785}$$

in which d = the internal diameter of the tube in inches.

M = the mass flow of the steam in thousands of lb. per hour per sq. ft. of cross-sectional area.

S = the mean specific heat of the steam.

and C = the conductivity of steam = $0 \cdot 0131 + 0 \cdot 000029t$, where t is the temperature in °F.

Economisers.

An economiser is an apparatus designed to heat the feed water by means of the flue gases before it enters the boiler proper. It is a characteristic of economisers that the water passes straight through them, without recirculation as in a boiler. The standard Green's economiser is almost universally used in conjunction with Lancashire and Cornish boilers, and frequently with boilers of the water tube type. Economisers for high pressure water-tube boilers are constructed of steel tubes, and are usually built integrally with the boiler. In some cases they are of the 'steaming' type, in which the tubes discharge a mixture of steam and water directly into the steam drum of the boiler.

In the design and operation of economisers, the following points should be observed. (1) The flue gases must not be cooled to their dew-point or there will certainly be trouble with external corrosion and fouling of the tubes. To prevent this, the feed water should enter the economiser at a temperature never below 100° F. If cold water has to be used as feed, and this should never be necessary, even in the smallest industrial plants, the economiser may be prevented from 'sweating' by delivering the feed water through a sort of injector which draws a certain amount of hot water from the economiser outlet and returns it to the inlet mixed with the fresh feed. (2) Unless the feedwater is properly de-aerated, it should not enter the economiser at less than 180° F., or the dissolved gases are likely to corrode the interior of the tubes. (3) The feed water should be softened and preferably dosed in addition with about 4 parts per million of sodium metaphosphate, which prevents the deposition of any scale in the tubes, even though the water contains some residual hardness. (4) Economisers should not be installed under conditions which would cause them to generate steam, unless they are specially designed to deal with the increase of volume of their contents due to the formation of steam. It is usual to design for a hot water temperature from 20° F. to 50° F. lower than the boiler steam temperature. (5) If the economiser is fitted with by-pass valves, it must also be equipped with a relief valve. (6) The necessity of providing access for inspection, and means of cleaning is obvious. (7) It must be possible to drain the economiser, and blow-off valves are often provided. (8) When a separate economiser is set in a brick flue, care must be taken to avoid gas pockets, as these may result in explosions. When, as usual, the economiser is higher than the flue, the best practice is to make the bottom of it level with the bottom of the incoming flue, and the top level with the top of the out-going flue. (9) The water inlet should be adjacent to the gas outlet so as to get the advantage in heat transmission due to counter flow. (10) The resistance to the flow of gases should be kept as low as possible. It is usual to allow a draught loss equal to 0·25 in. w.g. for an economiser of the Green type, when coal is being burnt at the rate of 20 to 25 lb. per sq. ft. of grate area.

Air Heaters.

The pre-heating of the air supply to boiler furnaces by means of the waste heat of the flue gases is now common practice in power stations. It not only enables more heat to be recovered from the gases than would be possible with an economiser supplied with water at the high feed temperatures now usual, but it has also the advantage of promoting the more efficient combustion of the fuel. The air heater is generally used in conjunction with an economiser, abstracting further heat from the gases after they have passed through the economiser. It has been estimated that an air-heater effects a saving of about one per cent. for every 30° F. rise in the temperature of the air supplied to the furnace. With pulverised coal firing the air may be heated to any degree consistent with the economical design of the whole plant, and temperatures up to 550° F. are employed in practice. With mechanical stoking it is questionable whether there is any net gain by exceeding an air temperature of about 300° F. on account of the increased maintenance charges on the stoker, although temperatures up to 450° F. have been used.

Air heaters are designed either on the recuperative or the regenerative principle. Recuperative heaters may be of either tubular or plate construction. In the former type the hot gas is caused to flow through vertical tubes surrounded by the air to be heated. The tubes, which are 2·5 ins. or 3 ins. in diameter and from 10 to 14 B.W.G. in thickness, are expanded into tube plates above and below. The length of the tubes may be as much as 30 ft. Soot blowers are used for cleaning them internally.

In plate type heaters the gas and air pass through alternate channels separated by the plates of which the heater is composed. In both designs the counterflow of air and gas is provided for as far as possible. With recuperative heaters, whether of the tube or plate type working under ordinary conditions of temperature and draught, the rate of heat transmission will not in general exceed from 2 to 3 B.Th.U. per sq. ft. per hour per degree of mean temperature difference. The heat transmitting surface required is therefore very large, and generally exceeds that of the boiler served.

The rate of overall heat transmission from gas to air in both plate and tubular heaters is given very closely by the formula:

$$K = W^{0.8}$$

in which K has its usual meaning of B.Th.U. per sq. ft. per hour per ° F. of mean temperature difference, and W is the weight of gas, in thousands of lb. per hour passing per sq. ft. of the area of flow.

A type of regenerative air heater which is largely used in power stations is that invented by Mr. Frederick Ljungstrom in 1922. This consists essentially of a large drum mounted on a vertical spindle. The interior of the drum is filled with a honeycomb arrangement of very thin steel sheet. The flue gases pass upwards through one half of the honeycomb, while the air passes downwards through the other half. The drum is kept in constant slow rotation at about 3 r.p.m. by means of a small motor, the power required being insignificant. As the drum rotates the portion heated by the passage of the flue gases moves continually into the path of the incoming air to which it gives up its heat, while the cooled portion continually returns to be heated again by the gases. Special arrangements are made to prevent, as far as possible, leakage between the gas and the air sides.

This type of heater has considerable advantages over the convection heater where space is important. It has been estimated that every inch of the height of the drum is equivalent in effect to a foot in the height of a convection heater. The regenerative type, moreover, can deal safely with gases at a higher temperature than would be advisable in an ordinary heater.

Little information as to details of design have been published, but it would appear that in the earlier apparatus of the Ljungstrom type, the rate of heat transmission from gas to air was about 930 per sq. ft. per hour, and that about 117 sq. ft. of surface were contained in each cu. ft. of the volume of the drum. The value of K from gas to air, was about the same as for convection heaters, which is to be expected, since the necessity of heat transmission from gas to metal and from metal to air is the same in both types.

Efficiency of Air Heaters.

There is a difference of opinion as to the way in which the efficiency of an air-heater ought to be calculated. If the air-heater is regarded as an apparatus for recovering the heat of the flue gases, it is evident that a perfect heater would reduce the gas temperature to that of the atmosphere. The efficiency of an actual heater would then be:

$$\frac{\text{Drop in gas temp.} \times 100}{\text{Gas inlet temp.} - \text{Air inlet temp.}}$$

It should, however, be noted that on this assumption, it might be impossible for even an idea heater to show an efficiency of 100 per cent.

Another way of looking at the matter is to consider that a perfect heater would raise the temperature of the air to that of the inlet gases. According to this view, the efficiency would be:

$$\frac{\text{Increase of air temp.} \times 100}{\text{Gas inlet temp.} - \text{Air inlet temp.}}$$

This is the formula endorsed by the U.S. Code for boiler testing. It is open to the objection that the specific heats of air and gas and the quantities of air and gas are supposed to be equal. To take these factors into account, the expression should be multiplied by W_2S_2/W_1S_1 in which W_1 and W_2 represent respectively the weights of gas and air passing through the heater per hour, and S_1 and S_2 represent the corresponding specific heats.

Air Required for Combustion.

When the analysis of any fuel is known, the quantity of air required for complete combustion and the nature and quantity of the products of combustion can be obtained from the following tables taken from 'Principles of Combustion in the Boiler Furnace,' by A. D. Pratt.

Pounds per Pound of Combustible.

Substance (one lb.)		Air theor. required Lbs.	Products of Combustion. CO_2	H_2O	N_2	CO	SO_2
Carbon (to CO_2)	C	11·49	3·67	—	8·82	—	—
Carbon (to CO)	C	5·75	—	—	4·42	2·33	—
Carbon Monoxide	CO	2·46	1·57	—	1·89	—	—
Sulphur	S	4·31	—	—	3·31	—	2·00
Hydrogen	H_2	34·48	—	9·00	26·48	—	—
Methane	CH_4	17·24	2·75	2·25	13·24	—	—
Acetylene	C_2H_2	13·26	3·38	0·69	10·18	—	—
Ethylene	C_2H_4	14·78	3·14	1·29	11·35	—	—
Ethane	C_2H_6	16·09	2·93	1·80	12·36	—	—
Hydrogen Sulphide	H_2S	6·09	—	0·53	4·68	—	1·88

Cubic Feet per Cubic Foot of Combustible.

Substance (one cub. ft.)		Air theor. required (cub. ft.).	Products of Combustion (cub. ft.). CO_2	H_2O	N_2	CO	SO_2
Carbon Monoxide	CO	2·38	1	—	1·88	—	—
Hydrogen	H_2	2·38	—	1	1·88	—	—
Methane	CH_4	9·52	1	2	7·52	—	—
Acetylene	C_2H_2	11·91	2	1	9·41	—	—
Ethylene	C_2H_4	14·29	2	2	11·29	—	—
Ethane	C_2H_6	16·67	2	3	13·17	—	—
Hydrogen Sulphide	H_2S	7·14	—	1	5·64	—	1

The weight of air theoretically required to burn one pound of fuel can also be obtained from the formula—

$$\text{Weight of air required} = 34\cdot48\left\{\frac{C}{3} + \left(H - \frac{O}{8}\right) + \frac{S}{8}\right\}$$

where C, H, O, and S represent percentages of carbon, hydrogen, oxygen, and sulphur in the fuel.

To secure complete combustion, and prevent the loss due to the presence of combustible gases in the flue gases, it is necessary to allow, in practice, from 30 to 60 per cent. more than the quantity of air theoretically required.

The above formula is not very suitable for gaseous fuels because its use would necessitate reducing the combustible gases to their constituent elements. When dealing with gases it is therefore preferable to make the calculation in terms of volume rather than weight, and the following formula is given by A. D. Pratt for the purpose:

Cu. ft. of air required per cu. ft. of gas

$$= 2\cdot38(CO + H_2) + 9\cdot52\,CH_4 + 11\cdot91\,C_2H_2 + 14\cdot29\,C_2H_4 + 16\cdot67\,C_2H_6 - 4\cdot76\,O_2.$$

The products of combustion can be obtained from the Table.

Effect of Excess Air on the Products.

If W is the weight of air theoretically required for perfect combustion, the effect of supplying x per cent. of excess air will be to add $\frac{W \times x \times 0\cdot232}{100}$ lb. of oxygen and $\frac{W \times x \times 0\cdot768}{100}$ lb. of nitrogen to the products of combustion.

In the case of gaseous fuels, if V is the volume of air theoretically required for perfect combustion, the effect of supplying x per cent. by volume of excess air will be to add $\frac{V \times x \times 0\cdot21}{100}$ cu. ft. of oxygen and $\frac{V \times x \times 0\cdot79}{100}$ cu. ft. of nitrogen to the products of combustion.

CALORIFIC VALUE OF FUEL.

To determine the efficiency of a boiler it is necessary to know the weight of fuel burnt, the calorific value of the fuel and the weight of steam produced. Practically all fuels contain a certain amount of moisture, and of hydrogen in a free or combined form. Consequently water vapour is present in the products of combustion. When such fuels are tested in a bomb calorimeter the water-vapour is condensed, and its latent heat is included in the calorific value of the fuel. The calorific value so determined is called the 'higher calorific value' (H.C.V.), and is generally used in English and American tests. On the Continent, however, the latent heat of the water-vapour is deducted, on the grounds that it is unavailable for use in a boiler, and the calorific value so obtained is called the 'lower calorific value' (L.C.V.). In comparing boiler tests it is important to know whether the H.C.V. or the L.C.V. has been employed, as a considerably higher boiler efficiency is shown by using the L.C.V.

To obtain the L.C.V. it is necessary to have an ultimate chemical analysis of the fuel, in order that its hydrogen content may be known. We have then—

$$\text{L.C.V.} = \text{H.C.V.} - 1{,}055(m + 9\text{H}).$$

where H is the weight of hydrogen, and m the weight of moisture in the fuel as fired.

Typical figures for the difference between the L.C.V. and H.C.V. of various fuels are as follows. Ordinary bituminous coals 3·5 per cent.; undried wood 16·0 per cent.; oil 6·4 per cent.; coke oven gas 11·5 per cent.

When an ultimate analysis of the fuel is available, the calorific value may be obtained very approximately by the use of the formula—

$$\text{H.C.V.} = 14{,}590\ \text{C} + 61{,}340\left(\text{H} - \frac{\text{O}}{8}\right) + 3{,}930\ \text{S}$$

in which C, H, O, and S represent respectively the percentages of carbon, hydrogen, oxygen, and sulphur in the fuel. The determination of the calorific value, however, directly by the calorimeter is the most satisfactory method.

BOILER EFFICIENCY AND HEAT LOSSES.

In a complete boiler test the whole of the heat given out per lb. of fuel as fired has to be accounted for. To do this it is necessary to compute the heat usefully absorbed by the boiler unit and that lost in various ways. The calculations require a chemical analysis of the fuel as fired and a complete determination of the constituents of the dry flue gases by means of an Orsat apparatus.

Heat absorbed by boiler.—If W = weight of steam produced per lb. of fuel, and F = temperature of feed water in deg. Fahr., the heat in B.Th.U. usefully absorbed per lb. of fuel is given by the expression—

$$\text{W} \times \left\{\text{Heat in steam} - (\text{F} - 32)\right\}$$

The heat in steam is the total heat per lb. for the particular pressure and temperature as found in the Steam Tables. The efficiency of the boiler is—

$$100 \times \frac{\text{Heat usefully absorbed}}{\text{Calorific value of fuel}}$$

Heat lost in dry chimney gases.—We have first to obtain the total weight of the dry chimney gases per lb. of fuel burnt, from the formula—

$$\text{W} = \frac{11\,\text{CO}_2 + 8\,\text{O}_2 + 7(\text{CO} + \text{N}_2)}{3\,(\text{CO}_2 + \text{CO})} \times \left(\frac{\text{C}}{100} - \frac{\text{A} \times \text{B}}{10{,}000} + \frac{\text{S}}{183}\right)$$

in which C and S are respectively the percentages of carbon and sulphur in the fuel as fired, A is the percentage of ash formed and B the percentage of combustible in the ash. Also, CO_2, O_2, CO and N_2 are respectively the percentages by volume of these several constituents in the flue gases.

The heat, in B.Th.U., carried away by the dry flue gases per lb. of fuel as fired, equal is to

$$\text{W} \times 0{\cdot}24 \times (\text{T} - t)$$

where T is the temperature of the flue gases and t the atmospheric temperature.

Expressed as a percentage, the heat loss on the dry flue gases is therefore

$$\frac{0{\cdot}24\ \text{W}(\text{T} - t) \times 100}{\text{calorific value}}$$

Loss due to presence of CO.—This is computed from the expression—

$$\frac{\text{CO}}{\text{CO}_2 + \text{CO}} \times \left(\frac{\text{C}}{100} - \frac{\text{A} \times \text{B}}{10{,}000} + \frac{\text{S}}{183}\right) \times 10{,}240$$

which gives the B.Th.U. lost per lb. of fuel due to the incomplete combustion of the carbon. The result, multiplied by 100 and divided by the calorific value of the fuel, is the percentage loss due to the above cause.

***Loss due to carbon in the ash.*—The loss due to carbon in the ashes, expressed as B Th.U. per lb. of fuel as fired, is**

$$\frac{A \times B}{10{,}000} \times 14{,}590$$

where A is the percentage of ash, and B is the percentage of carbon in the ash.

Multiplying the result by 100 and dividing by the calorific value gives the loss as a percentage of the heat in the fuel.

***Loss due to moisture in fuel.*—If m is the percentage of moisture in the fuel as fired, the heat lost due to this is**

$$\frac{m}{100}\left\{(212 - t) + 970 + 0{\cdot}48\,(T - 212)\right\}$$

B.Th.U. per lb. of fuel as fired. This loss may be expressed as a percentage by multiplying the result of the above by 100 and dividing by the calorific value of the fuel.

***Loss due to Hydrogen content of fuel.*—The hydrogen in the fuel combines with oxygen to form water, which passes away as superheated steam in the flue gases. The loss due to this, expressed in B.Th.U. per lb. of fuel as fired, is**

$$\frac{H}{100} \times 9\left\{(212 - t) + 970 + 0{\cdot}48\,(T - 212)\right\}$$

which can be expressed as a percentage by multiplying by 100 and dividing by the calorific value.

HEAT BALANCE.

By adding together the heat absorbed by the boiler per lb. of fuel, and the various losses enumerated above, the result should almost equal the calorific value of the fuel, the difference being accounted for by 'radiation and other losses.'

DETERMINATION OF CHIMNEY LOSSES.

An exact method of determining chimney losses from a knowledge of the economiser temperatures was described by R. H. Parsons in the *Beama Journal* of June 1939. The method avoids the necessity for an analysis of the flue gases, or a knowledge of their CO_2 content, and its accuracy is quite independent of the cleanliness or the efficiency of the economiser.

The chimney losses, in B.Th.U. per lb. of water evaporated in the boiler are given by the formula:

$$\text{B.Th.U.} = (T_5 - T_6) \times (T_2 - T_1)/(T_3 - T_4).$$

in which T_1 and T_2 are the temperatures of water at inlet and outlet of the economiser; T_3 and T_4 are the temperatures of the gases entering and leaving the economiser; T_5 is the temperature at the stack inlet, and T_6 is the temperature of the outside air.

Hence, if W be the weight of water evaporated per lb. of fuel, and C is the calorific value of the fuel, the stack losses expressed in the usual way as a percentage of the heat of the fuel, are:

$$\text{Stack losses} = \text{B.Th.U.} \times W \times 100/C.$$

Also, if R be the lbs. of flue gas per lb. of water evaporated:

$$R = 4{\cdot}167(T_3 - T_4)/(T_5 - T_6)$$

which gives a useful indication of the amount of excess air.

Many calculations relating to combustion are much simplified by a knowledge of the maximum percentage of CO_2 that is theoretically obtainable with the particular kind of fuel being burnt. When this is known a whole range of useful facts can be deduced merely from the readings of any ordinary CO_2 meter.

The maximum theoretical percentage of CO_2 in the dry flue gases, which will be denoted by M, will vary, of course, with the nature and quality of the fuel; representative figures are about 18·7 for bituminous coal, 20·5 for coke and 15·5 for oil. The value of M can be found either from the chemical analysis of the fuel, or more simply from an Orsat test of the flue gases.

From the analysis we get

$$M = \frac{21C + 7{\cdot}875S}{C + 0{\cdot}375S + 2{\cdot}37H - 0{\cdot}296 \times O}$$

in which C, S, H and O denote respectively the percentages by weight of the carbon, sulphur, hydrogen and oxygen in the fuel. This equation assumes that sulphur dioxide is reckoned as carbon dioxide, because neither the Orsat apparatus nor the ordinary CO_2 meter makes any distinction between these gases.

Alternatively we can obtain M from the results of an Orsat test by the formula

$$M = \frac{21(CO_2 + CO)}{21 - O_2 + 0{\cdot}395CO}$$

which simplifies to

$$M = \frac{21CO_2}{21 - O2}$$

when no CO is present.

Having obtained M for any fuel, the amount of excess air, expressed as a percentage of the minimum quantity theoretically required for perfect combustion, is obtained from the reading of the CO_2 meter by the formula

$$\text{Excess air} = \frac{79(M - CO_2)}{CO_2(1 - 0 \cdot 01M)} \text{ per cent.}$$

With complete combustion there is neither CO nor H_2 present in the flue gas, and in such a case the weight of the dry gases (including excess air) per lb. of combustible matter actually burnt is given by

$$\text{Weight of dry gas} = \frac{721M + CO_2(4M - 21)}{CO_2(1 \cdot 734M + 26 \cdot 58)} \text{ lbs.}$$

The weight of the water formed by combustion will be :

$$\text{Weight of moisture} = \frac{9(21 - M)}{21 + 1 \cdot 37M} \text{ lbs.}$$

and the total weight of the products of combustion, together with excess air, per lb. of combustible matter burnt, will be

$$\text{Weight of total gases} = \frac{569 \cdot 59M - CO_2(5 \cdot 84M - 172 \cdot 41)}{CO_2(21 + 1 \cdot 37M)} \text{ lbs.}$$

When a fuel can be considered as a pure hydro-carbon, which is approximately the case for oils, its composition may be obtained from the CO_2 in the flue gases, by the formula

$$\frac{H}{C} = \frac{2 \cdot 37M}{21 - M}$$

provided that its combustion is complete. C/H is the carbon-hydrogen ratio of the fuel.

Approximate Combustion Formulæ.

Accurate formulæ concerning combustion have been given above, and should be used for all important calculations. There are, however, a number of approximate formulæ which are sometimes useful, although they should be employed with discretion as they are not applicable to all fuels.

Weight of air theoretically required per lb. of coal $= \dfrac{\text{Calorific value}}{10{,}000} \times 7 \cdot 2.$

Weight of air entering furnace per lb. of coal $= \dfrac{132 \times \text{Calorific value}}{CO_2 \times 10{,}000}$

Weight of flue gases per lb. of fuel = (lbs. of air + 1)

Percentage of excess air $= \left(\dfrac{18 \cdot 9}{CO_2} - 1\right) \times 100.$

(*N.B.*—This applies to normal bituminous coals, burnt with no CO present in the flue gases.)

Percentage loss in flue gases $= 0 \cdot 35 \left(\dfrac{T - t}{CO_2}\right)$

in which T = exit temperature of gases and t = temperature of air.

Rate of Combustion and Draught.

With natural or chimney draught the air pressure at the grate depends on the height of the chimney, the temperature of the gases in the chimney, and the lay out of the flues. A typical figure is 0·65 in. of water per 100 ft. of chimney height, above the grate. The effective draught at the grate for a Lancashire boiler battery with economiser will be about one-third of this figure, and rather less for a water-tube boiler installation with economiser.

The coal burnt per hour per square foot of grate will be about :—

20 lbs. average steam coal ; 5–10 lbs. anthracite very thinly spread.

For other chimneys and draughts the rate of combustion increases not quite in proportion to height.

The draught at the base of the chimney is due to the difference of the weight of the column of hot gases in the chimney and the weight of a similar column of external air.

If t_1 = temperature of external air ; t_2 = mean temperature of gases in chimney ; h = height of chimney in feet ; d = draught in inches of water, measured at foot of chimney ; density of gases (of typical composition) at 32° F. and atmospheric pressure = ·084 lb. per cu. ft. ; density of air = ·0807 lb. per cu. ft., then,

Mean density of gases at chimney temperature $= \cdot 084 \times \dfrac{493}{t_2 + 460} = \dfrac{41 \cdot 4}{t_2 + 460}$;

Density of external air $= \cdot 0807 \times \dfrac{493}{t_1 + 460} = \dfrac{39 \cdot 8}{t_1 + 460}$

Pressure difference at base of chimney in lbs. per sq. foot $= h\left(\frac{39\cdot8}{t_1 + 460} - \frac{41\cdot4}{t_2 + 460}\right)$;

or

d in inches of water gauge $= h\left(\frac{7\cdot64}{t_1 + 460} - \frac{7\cdot95}{t_2 + 460}\right)$.

This is the 'theoretical' draught. The actual available draught is less than the theoretical by that required to produce the velocity of flow of the gases and by that required to overcome friction in the flues, economiser, and chimney. The loss due to velocity (in inches w.g.)

$$= d_v = 0\cdot10 \times \text{density of gases} \times \text{pressure} \times \frac{V^2}{t_2 + 460}$$

$$= \text{for typical gases } 0\cdot123 \frac{V^2}{t_2 + 460}$$

where

V = velocity of gases in chimney.

The friction loss (in inches w.g.) in the chimney

$$= d_f = 0\cdot40 \times f \times \text{density} \times \text{pressure} \times \frac{V^2}{t_2 + 460} \times \frac{L}{D}$$

where

f = friction coefficient = 0·016 say;

L = length of gas path in feet;

D = diameter of chimney in feet;

$$\therefore \text{ friction loss } d_f = 0\cdot0079 \frac{V^2}{t_2 + 460} \frac{L}{D}$$

It is convenient to relate the losses in the chimney to the weight of gas flowing rather than to the velocity.

If W = rate of flow of gas in lbs. per sec., the velocity loss d_v becomes

$$d_v = 0\cdot000117 \frac{W^2 (t_2 + 460)}{D^4}$$

and the friction loss

$$d_f = 0\cdot0000075 \frac{W^2 (t_2 + 460) L}{D^5}$$

Example.—A boiler battery consists of six boilers with a grate area of 38 sq. ft. each. What will be the draught at the foot of a chimney 150 ft. high and 7 ft. diameter, when the rate of combustion is 25 lbs. per hour per sq. ft. of grate area and the mean chimney temperature is 600° F.?

The coal burned per hour is 5,700 lbs.

The flue gases will amount to about 19 lbs. per lb. coal, or 108,000 lbs. per hour = 30 lbs. per sec.

The 'theoretical draught' (with external air at 60° F.)

$$= 150\left(\frac{7\cdot64}{520} - \frac{7\cdot95}{1060}\right) = 1\cdot08 \text{ inches w.g.}$$

$$\text{Loss due to velocity} = 0\cdot000117 \times \frac{30^2 \times 1060}{7^4}$$

$$= 0\cdot046 \text{ inches w.g.}$$

$$\text{Loss due to friction} = 0\cdot0000075 \times \frac{30^2 \times 1060 \times 150}{7^5}$$

$$= 0\cdot064 \text{ inches w.g.}$$

∴ available draught at foot of chimney

$$= 1\cdot08 - (0\cdot046 + 0\cdot064)$$

$$= 0\cdot97 \text{ inch w.g.}$$

From this, the friction losses at flues, economiser (if any) and dampers must be deducted in order to find the draught at the boiler furnaces. Note, in the above example, if there were an economiser, the chimney temperature would be lower than 600° F., and hence the 'theoretical draught' itself would be less. The economiser thus reduces draught both by reducing chimney temperature and by its additional frictional resistance.

Effect of Altitude on Draught.

At high altitudes, the barometric height is less than at sea-level, and hence the density of the atmosphere is less. Hence also the 'theoretical draught' of a given chimney is reduced.

In order to obtain the same draught the height of the chimney must be increased.

The approximate ratio of the height required to the height required at sea-level is given in the following table:

Altitude above sea-level (ft.)	500	1,000	1,500	2,000	3,000	4,000
Ratio of heights	1·02	1·04	1·06	1·08	1·12	1·16

The diameter of the chimney at altitudes should also be increased by about 1½ per cent. for each 1,000 ft. of altitude.

DRAUGHT AND RATE OF COMBUSTION.

Even if the draught at the grate is accurately known, this alone does not fix the rate of combustion, which is influenced by the kind of fuel and thickness of fire; the following tables, due to W. S. Hutton, must therefore be regarded as approximate only.

RATE OF COMBUSTION FOR DIFFERENT DRAUGHT PRESSURES.

Height of Chimney above Grate, in Feet.	Total Draught Pressure, in Inches of Water.	Rate of Combustion, in Pounds per Square Foot of Grate per Hour.	Height of Chimney above Grate, in Feet.	Total Draught Pressure, in Inches of Water.	Rate of Combustion, in Pounds per Square Foot of Grate per Hour.
25	·182	10	130	·948	30
50	·364	16	140	1·029	34
60	·437	17	150	1·095	40
70	·512	18	180	1·313	50
80	·583	19	200	1·459	60
90	·657	20	225	1·641	70
100	·729	22	250	1·825	80
110	·802	24	300	2·189	90
120	·875	27	400	2·553	112

TOTAL DRAUGHT REQUIRED FOR EFFICIENT COMBUSTION OF DIFFERENT KINDS OF FUEL.

Fuel.	Total Draught, in Inches of Water.	Fuel.	Total Draught, in Inches of Water.
Straw	0·20	Slack, very small	0·7 to 1·1
Wood	0·30	Coal-dust	0·8 to 1·1
Sawdust	0·35	Semi-anthracite coal	0·9 to 1·2
Peat, light	0·40	Mixture of breeze and slack	1·0 to 1·3
Peat, Heavy	0·50	Anthracite, round	1·2 to 1·4
Sawdust, mixed with small coal	0·60	Mixture of breeze and coal-dust	1·2 to 1·5
Steam coal, round	0·4 to 0·7	Anthracite slack	1·3 to 1·8
Slack, ordinary	0·6 to 0·9		

In hand-fired plants, using coal for fuel, the rate of combustion should not be permitted to fall below 15 lbs. per sq. ft. of grate surface per hour. With stokers the rate of combustion may be as high as the design of the plant will permit. When firing with fine steam-size coal, the rate should not be less than 10 lbs. per sq. ft. of grate area per hour.

A typical working figure for the rate of combustion in stationary boilers with good natural or moderate assisted draught is 25 lbs. coal per hour per sq. ft. of grate area. For locomotives the rate is about 60 to 70, but may rise to 200.

Transmission of Heat in Boilers.

The transfer of the heat developed by the combustion of the fuel in the furnace to the water in the boiler takes place in two ways: (1) by radiation from the incandescent fuel and the hot gases; and (2) by convection and conduction—i.e. conduction through the materials in the path of the heat flow and by the contact of the gases and water with the metal separating them.

The transmission of heat to that part of the heating surface which is directly exposed to the fire is practically entirely by radiation, and that to the remainder of the heating surface practically entirely by convection.

Transmission by Radiation.

The heat transmitted by direct radiation in most boilers is strikingly large, so that the fire-box surface is much the most powerful evaporating surface.

The amount of heat radiated from a fire of temperature T_1 (° F. absolute) to the fire-box surface at T_2 is

$$1{,}600 \times \frac{T_1^4 - T_2^4}{1{,}000^4} \text{ B.Th.U. per hour per sq. ft.}$$

or practically, since T_2^4 is always negligible in comparison with T_1^4,

$$\text{Radiant heat} = 1{,}600 \left(\frac{T_1}{1{,}000}\right)^4.$$

Whether the radiating surface to be taken into account in calculating the total heat radiated is the grate area, or a larger amount, is debatable. The formula indicates, however, the importance of the direct heating surface, and also the importance of high combustion temperature for high rates of transmission.

Radiant Heat Transmission in Boiler Furnaces.

In a discussion of heat transmission by radiation in boiler furnaces in *The Engineer*, May 16, 1930, a conclusion is reached endorsing the formula given originally by Hudson and modified by Orrok, viz.:

Heat in B.Th.U. per hour entering one sq. ft. of water-cooled surface exposed to radiant heat is equal to

$$\mathrm{OH}\left(\frac{1}{1 + \frac{A\sqrt{O}}{27}}\right)$$

where O = lbs. coal burnt per sq. ft. of heat absorbing surface per hour

H = calorific value of coal;

A = lbs. air per lb. coal.

This formula agrees well with the results of Wohlenberg's analysis (for which see *Trans. Amer. Soc. Mech. Eng.*, 1925, 1926, 1928, 1929).

Transmission through Plates or Tubes.

The transmission of heat from a fluid to another fluid on the other side of a metal tube depends on a large number of factors, including, as well as the area of tube surface and the temperatures, the velocities of the fluids, the diameter of the tube, and the nature of the fluids and of the tube surfaces. The change of temperature of the fluid passing through the tube is, of course, itself dependent on the factors named, so that the factors are not all independent. Differences of conductivity and thickness of the metals of the tube are negligible, the resistance to the passage of heat through the metal being a very small portion of the total resistance from fluid to fluid.

The coefficient of heat transmission is the number of heat units transmitted per unit time per unit area per degree of mean temperature difference (see p. 170, and for determination of mean temperature difference see p. 172).

For the conditions prevailing in the tubes of fire tube boilers, such as boilers of the locomotive type, waste-heat boilers, and air heaters, that is, where the current of hot gas—and for an air heater, the current of air being heated—is along the tubes, the coefficient of heat transmission can be related approximately to $\frac{w}{a}$, where w is the rate of gas flow in lbs. per sec., and a the cross section of the gas path in sq. ft. The results of investigations on different boilers and heaters vary widely, but for values of $\frac{w}{a}$ between 1 and 4, mean values of the transmission coefficient h are given by formulae of the type $h = c\left(\frac{w}{a}\right)^n$ where h = B.Th.U. transmitted per second per sq. ft. of heating surface per degree F. of temperature difference *between the gas and the metal*.

For cases in which the heat is transferred mainly by convection, say for temperatures up to 700° F., c may be taken as 0·0014, n being 0·67.

Where radiation also assists the heat transference, as in the tubes of boilers exposed to gases at or near flame temperatures, the formula $h = 0\cdot0025\left(\frac{w}{a}\right)^{0\cdot67}$ may be used.

Note that these formulae are to be used in conjunction with the temperature *between gas and metal*, and not that between the two fluids separated by the metal wall.

If, as in a boiler, one side of the tube metal is in contact with water, the temperature difference between metal and water is negligible compared with that between gas and metal, and in this case the temperature difference between gas and water may be used instead of that between gas and metal.

When both the substances separated by the tube wall are gaseous, the temperature differences on *each* side of the tube must be considered, or else the overall coefficient of transmission calculated.

Schack (in *Die Industrielle Wärmeübertragung, Berlin,* 1929) gives a formula, which, in British units, becomes.

$$h = 0\cdot000165\,\frac{v_0^{\,0\cdot8}}{4\sqrt{d}}$$

where v_0 = velocity of gas, reduced to 32° F., in feet per sec.
d = tube diameter in inches.

For air, this becomes
$$h = 0\cdot00124\,\frac{\left(\frac{w}{a}\right)^{0\cdot8}}{4\sqrt{d}}$$

and for flue gas,
$$h = 0\cdot0019\,\frac{\left(\frac{w}{a}\right)^{0\cdot8}}{4\sqrt{d}}$$

The procedure that may be adopted is illustrated by the following examples :

Example 1.—A tubular boiler, in which the gases pass through the tubes at the rate $\frac{w}{a} = 2$; temperature of water (and of steam) = 350° F., gas temperatures 1,000° F. at inlet to 450° F. at outlet.

Mean temperature difference (M.T.D.) between gas and water

$$= \frac{(1{,}000 - 350) - (450 - 350)}{\log_e \dfrac{1{,}000 - 350}{450 - 350}} = 294.$$

This may also be taken as the M.T.D. between the gas and the tube metal.

$$\text{Coefficient } h = 0\cdot0014 \times 2^{0\cdot67}$$
$$= 0\cdot0022 \text{ B.Th.U. per sec. per sq. ft. per } ^\circ\text{ F.}$$

∴ heat transmission rate = $0\cdot0022 \times 294 = 0\cdot646$ B.Th.U. per sec. per sq. ft. (the heating surface being measured on the *gas* side of the tube).

If the tube is 2 ins. internal diameter, $a = 3\cdot14$ sq. ins. $= 0\cdot0218$ sq. ft.

$$\therefore w = \frac{w}{a} \times a = 2 \times 0\cdot0218 = 0\cdot0436 \text{ lbs. per sec.}$$

Taking $0\cdot25$ as the mean specific heat of the gases, the heat given up in each tube

$$= 0\cdot0436 \times 0\cdot25 \times (1{,}000 - 450) = 6 \text{ B.Th.U. per sec.}$$

$$\therefore \text{tube surface required (internal)} = \frac{6}{0\cdot646} = 9\cdot3 \text{ sq. ft.}$$

$$\therefore \text{ tube length} = \frac{\text{surface}}{\text{perimeter}} = \frac{9\cdot3 \times 12}{6\cdot28} = 17\cdot8 \text{ ft.}$$

Evaporation (from and at 212° F.) per tube

$$= \frac{6 \times 3{,}600 \text{ (B.Th.U. per hour)}}{970 \text{ (B.Th.U. per lb. steam)}} = 22\cdot2 \text{ lbs. steam per hour.}$$

∴ for 2,000 lbs. steam per hour

Number of tubes required = 90

Total tube heating surface = 836 sq. ft.

Evaporation per sq. ft. of heating surface = 2·49 lbs. per hour (from and at 212° F.)

Example 2.—A tubular air heater, with tubes 2 ins. internal and 2¼ ins. external diameter, gas flowing inside tubes, air flowing in counter current direction outside tubes: $\frac{w}{a}$ for gas = 2; $\frac{w}{a}$ for air = 1·86; gas temperature 700° F. and 450° F. at inlet and outlet; air temperatures 60° F. and 334° F.

h_g = coefficient of heat transmission on gas side = 0·0022.

h_a = do. on air side = 0·00212.

Mean temperature difference between gas and air

$$= \frac{(700 - 334) - (450 - 60)}{\log_e \frac{700 - 334}{450 - 60}} = 378.$$

It would be incorrect to take either h_g or h_a multiplied by 378 as the heat flow per sq. ft. per sec., since h_g and h_a refer respectively to the transmission between gas and *metal*, and *metal* and air only.

It can be shown that the mean temperature difference (M.T.D.) between gas and metal

$$= \text{M.T.D. between gas and air} \times \frac{1}{1 + \frac{h_g \times A_g}{h_a \times A_a}}$$

where A_g, A_a are the metal surfaces exposed to gas and air respectively.

In the example, $\frac{A_g}{A_a} = \frac{\text{internal diameter of tube}}{\text{external diameter of tube}} = \frac{2}{2 \cdot 25}$

$$\therefore \quad \text{M.T.D. gas to metal} = 378 \frac{1}{1 + \frac{0 \cdot 0024 \times 2}{0 \cdot 00228 \times 2 \cdot 25}} = 195 \cdot 3$$

$$\text{Similarly, M.T.D. metal to air} = 378 \frac{1}{1 + \frac{0 \cdot 00228 \times 2 \cdot 25}{0 \cdot 0024 \times 2}} = 182 \cdot 7$$

∴ heat transmitted per sec. = 0·0022 × 195·3 B.Th.U. per sq. ft. of surface exposed to gas = 0·430 B.Th.U. per sq. ft.

Also heat transmitted per sec. = 0·00212 × 182·7 = 0·388 B.Th.U. per sq. ft. of surface exposed to air.

These heat transmission rates must be inversely proportional to the areas on the gas and air sides, *i.e.* $\frac{0 \cdot 430}{0 \cdot 388}$ should be $\frac{2 \cdot 25}{2 \cdot 00}$, which is fulfilled very closely.

The length of tube required may be calculated as follows:

Rate of flow of gas in tube = $\frac{w}{a}$ × area of tube

$$= 2 \times \frac{3 \cdot 1416}{144} = 0 \cdot 0436 \text{ lbs. per sec.}$$

Specific heat of gas = 0·25.

∴ heat given up by gas in tube

= 0·0436 × 0·25 × (700 − 450) B.Th.U. per sec.

= 2·725 B.Th.U. per sec.

∴ heating surface required (on gas side) = $\frac{2 \cdot 725}{0 \cdot 430}$ = 6·32 sq. ft.

Internal perimeter of tube = 3·1416 × 2 = 6·283 in. = 0·5236 ft.

∴ tube length = $\frac{6 \cdot 32}{0 \cdot 5236}$ ft. = 12·1 ft.

The temperature differences are the chief factors in determining the length of tube (for usual values of $\frac{w}{a}$).

The number of tubes is determined from the total gas flow compared with the flow per tube.

Thus if the gas flow is 4·4 lbs. per sec., the number of tubes for the above example would be $4 \cdot 4 \div 0 \cdot 0436 =$ say 100.

Example 3.—An alternative method of finding the heat transmission rates in the previous example is as follows:

If the overall coefficient of heat transmission from gas to air (neglecting the relatively small resistance of the metal) $=$ H,

then $$\frac{1}{H} = \text{resistance to heat flow}$$
$$= \frac{1}{h_a} + \frac{1}{h_g} = \frac{1}{0 \cdot 0022} + \frac{1}{0 \cdot 00212}$$
therefore $$H = 0 \cdot 00108.$$

The overall mean temperature difference is 378°.

∴ rate of heat transmission

$$= 0 \cdot 00108 \times 378 \text{ B.Th.U. per sec. per sq. ft. of mean area}$$
$$= 0 \cdot 408 \text{ B.Th.U. per sec. per sq. ft. of mean area.}$$

The ratio of the areas on the gas and air sides is 2 : 2·25, or the areas are respectively 6 per cent. below and above the mean area.

The transmission rates are thus $0 \cdot 408\,(1 - 0 \cdot 06) = 0 \cdot 384$ and $0 \cdot 408\,(1 + 0 \cdot 06) = 0 \cdot 441$ B.Th.U. per sec. per sq. ft. of the respective areas, agreeing with the former calculation.

General Remarks on Tubular Heaters.

For a given range of temperatures, and a given value of $\frac{w}{a}$, the required tube length is proportional to tube diameter.

Hence if the length for a particular diameter has been calculated, the lengths for other diameters can be very readily found.

The tube length is *not* proportional to the drop of temperature of the heating medium; the extra tube length for each additional ten degrees say of temperature reduction at the outlet end becomes successively greater. Thus, if in a waste heat boiler, with water at 350° F., and initial gas temperature 1,000° F., the tube length is 12·2 ft. for outlet temperature 450° F., it will be 13·7 for 430° F., and 15·5 ft. for 410° F. (the rate $w \div a$ and the diameter remaining the same)

The loss of pressure in the tubes, in inches of water gauge, may be taken as approximately

$$\left\{ 0 \cdot 0000216 \frac{l}{d} \mathrm{T}_m + 0 \cdot 000075 \left(\frac{\mathrm{T}_1}{2} + \mathrm{T}_2 \right) \right\} \left(\frac{w}{a} \right)^2;$$

where

$l =$ tube length in ft.;

$d =$ internal diameter of tube in ins.;

$\mathrm{T}_m =$ mean gas temperature, ° F. absolute $= \frac{\mathrm{T}_1 + \mathrm{T}_2}{2}$;

$\mathrm{T}_1 =$ inlet gas temperature, ° F. absolute;

$\mathrm{T}_2 =$ outlet gas temperature, ° F. absolute;

$\frac{w}{a} =$ lbs. of gas per sec. $\div$ area of section of tube in sq. ft.

In this formula, the pressure of the gas is assumed to be atmospheric within a few ins. water gauge.

Fry's Empirical Formula.

In the case of boiler tubes with flue gases passing through them, the change of temperature of the gases, and thus also the heat transferred, can be calculated from the following equation (Lawford H. Fry, *Engineering*, Aug. 27, 1920):—

$$\log\log. \frac{\mathrm{T}_1}{t} - \log\log. \frac{\mathrm{T}_2}{t} = \mathrm{M}x \quad . \quad . \quad . \quad (1),$$

in which T_1, T_2 are the mean flue gas temperatures, ° F. absolute, at two sections of the tube x feet apart, t the mean temperature (absolute) of the tube wall, and 'lolog.' means 'logarithm of the logarithm.'

M is given by $\log. M = B - m \log. \dfrac{W}{p}$. (2) $\log. (B + 1\cdot3) = \bar{1}\cdot71 - 0\cdot54 \log. d$. (3)

$$\log. m = \bar{1}\cdot36 + 0\cdot37 \log. d \quad . \quad . \quad . \quad (4)$$

where,

d = internal diameter of tube in inches; p = perimeter of tube ($= d \times 3\cdot1416$); W = rate of flow of gas in lbs. per hour.

Having determined the fall of temperature of the gases from the above, the heat given up —*i.e.* the heat transferred—is equal to

$$W \times (T_1 - T_2) \times \text{mean specific heat of gases.}$$

The following table gives the values of M for a number of tube diameters and rates of flow. These values may be used directly in equation (1).

VALUES OF M.

d inches.	W in lbs. per hour.				
	100	200	400	800	1,600
0·5	0·133	0·117	0·104	0·092	0·081
1·0	0·074	0·063	0·054	0·046	0·039
2·0	0·050	0·040	0·033	0·027	0·022
4·0	0·040	0·030	0·023	0·018	0·014
8·0	0·037	0·026	0·019	0·013	0·009

EXAMPLES.

(1) To find fall of temperature in flue tube 2 ins. diameter and 12 ft. long, with rate of gas flow of 400 lbs. per hour, boiler pressure 180 lbs. per sq. in., and temperature of gases 2,000° F. at entrance of tubes.

The temperature of the metal of the tube is nearly that of water in the boiler—*i.e.* nearly that of steam at 180 lbs. per sq. in.; thus t = 380° F., about. (Slight differences in t have very little influence as a rule.)

From the table, for 2-in tube, and W = 400 lbs. per hour, M = 0·033;

$$\therefore \text{lolog.} \frac{2{,}000 + 460}{380 + 460} - \text{lolog.} \frac{T_2}{380 + 460} = 0\cdot033 \times 12$$

$$\therefore \text{lolog.} \frac{T_2}{840} = \bar{1}\cdot273;$$

$$\therefore \log. \frac{T_2}{840} = 0\cdot1875.$$

$$\therefore T_2 = 1{,}293° \text{ F. abs., or } 833° \text{ F.}$$

(2) To find the heat transferred.

The mean specific heat of flue gases at the temperature in question may be taken as 0·31. Then the heat transferred in B.Th.U. per hour

$$= 400 \times (2{,}000 - 833) \times 0\cdot31 = 145{,}000.$$

The tube surface is 6·29 sq. ft., and heat transferred per sq. ft. = 23.100 B.Th.U. per hour.

(3) To find length of tube required to reduce the gas temperature to 700° F., rate of flow and diameter being as before:

$$\text{lolog.} \frac{2{,}000 + 460}{380 + 460} - \text{lolog.} \frac{700 + 460}{380 + 460} = 0\cdot033 \times \text{length.}$$

From which, length = 15·8 ft.

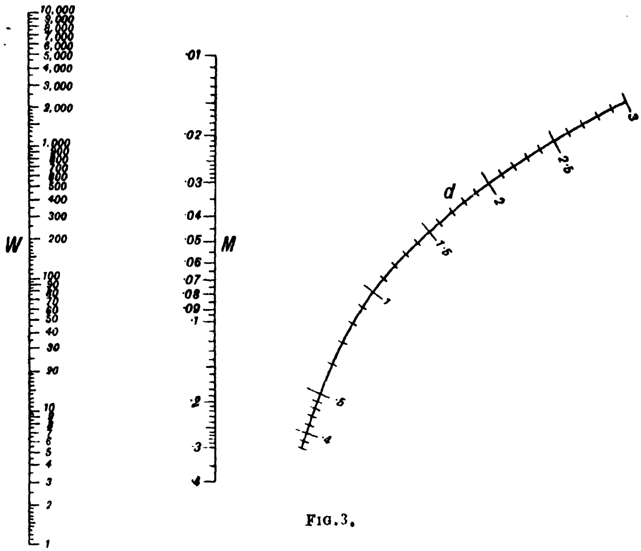
W
10,000
5,000
4,000
3,000
2,000
1,000
500
400
300
200
100
50
40
30
20
10
5
4
3
2
1
M
·01
·02
·03
·04
·05
·06
·07
·08
·09
·1
·2
·3
·4
d
·4
·5
1
1·5
2
2·5
3

FIG. 3.

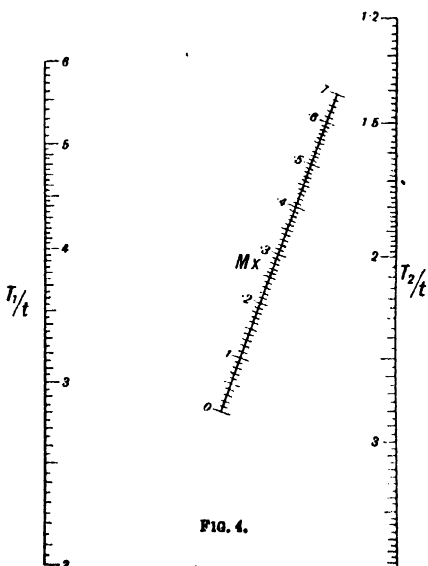
T_1/t
6
5
4
3
2
Mx
0
·1
·2
·3
·4
·5
·6
·7
T_2/t
1·2
1·5
2
3
4

FIG. 4.

GRAPHIC CALCULATION OF HEAT TRANSMISSION THROUGH BOILER TUBES.

Fig. 3 (p. 30) replaces equations (2), (3) and (4), and gives directly, without calculation, the value of M for any given values of W and d, or if required the value of any one of the three quantities W, M, d when the other two are known. In fig. 3 any straight line will cut the three scales graduated in terms of W, M, d at graduations which satisfy equations (2) (3) (4).

Fig. 4 enables equation (1) to be solved graphically. The values of $\frac{T_1}{t}$ and $\frac{T_2}{t}$ are marked on the two end scales and values of Mx on the middle scale. Any straight line cuts these scales at graduations which satisfy equation (1).

Example 1 :—To find the length of flue necessary for a temperature drop from 2,000 deg. F. to 720 deg. F. in a half-inch flue, with a wall temperature of 380 deg. F. and where 100 lb. of gas are passing per hour.

In fig. 3 draw a straight line joining the points W = 100 and $d = 0{\cdot}5$. This line cuts the M scale at the required value of M = 0·132.

Since

$$\frac{T_1}{t} = \frac{2{,}000 + 460}{380 + 460} = 2{\cdot}93, \text{ and } \frac{T_2}{t} = \frac{720 + 460}{380 + 460} = 1{\cdot}41,$$

therefore a straight line drawn in fig. 4 joining these graduations on the $\frac{T_1}{t}$ and $\frac{T_2}{t}$ scales, will cut the Mx scale, at the required value of Mx, which is seen to be 0·50. Therefore, since $M = 0{\cdot}132$, $x = \frac{0{\cdot}50}{0{\cdot}132} = 3{\cdot}78$ ft., *i.e.*, a flue 3·78 ft. in length will be required for the given temperature drop.

Example 2 :—To find what drop in temperature will take place in a half-inch flue, 5 ft. in length, when 1,600 lb. of gas are passing per hour and the temperature of flue gas at entry is 2,000 deg. F., the flue wall temperature being 420 deg. F.

From fig. 1, by joining W = 1,600 and $d = 0{\cdot}5$, we get $M = 0{\cdot}081$, therefore,

$$Mx = 0{\cdot}405.$$

But $\frac{T_1}{t} = \frac{2{,}000 + 460}{420 + 401} = 2{\cdot}80$, therefore, joining $\frac{T_1}{t} = 2{\cdot}80$, and $Mx = 0{\cdot}405$, in fig. 2, we get

$$\frac{T_2}{t} = 1{\cdot}50,$$

therefore,

$$T_2 = 1{\cdot}50 \times 881 = 1{,}322° \text{ C.} = 861° \text{ F.}$$

Actually, lines need not be drawn on the diagrams, as it is sufficient to lay a rule on the scales through the required points.

(A. J. V. Umanski, M.Sc., *Engineering*, June 10, 1921.)

Experiments made by Yarrow and Co., Ltd. (*W. W. Marriner, in discussion on paper by Dorey, 'Tubes for Water Tube Boilers,' Inst. Marine Engineers*, 1930) indicated the following rates of heat transmission in boiler tubes: ordinary boiler, evaporating 6 to 8 lb. steam per sq. ft. of H.S. per hour, in fire rows, between 40,000 and 60,000; in outer rows between 1,000 and 3,000 B.Th.U. per hour per sq. ft.

In destroyer boiler at full power, evaporating 16 to 18 lb. per sq. ft. per hour, fire rows between 100,000 and 120,000; outer rows, between 2,000 and 8,000 B.Th.U. per hour per sq. ft.

Heat Transfer in Cross Flow.

The following formulæ are considered to be the most satisfactory in practice for determining the transfer of heat from gases to tubes when the gases flow at right angles to the tubes. They are taken from 'Industrial Heat Transfer,' by Schack, Goldschmidt and Partridge (1939), which

contains valuable information on all heat transfer questions. In the formulæ, K = rate of heat transmission in B.Th.U. per hour per sq. ft. of tube surface, per ° F. of temperature difference between gas and tube. V = the velocity of gas in feet per sec. between the pipes, supposing its volume to be reduced to that corresponding to 32° F. and atmospheric pressure. D = external diameter of tubes in feet.

For a single tube $K = 0 \cdot 7 \dfrac{V^{0 \cdot 56}}{D^{0 \cdot 44}}$

For banks of tubes, not staggered $K = A \times \dfrac{V^{0 \cdot 654}}{D^{0 \cdot 346}}$

where A = 0·615 for two rows ; 0·635 for three rows ; 0·65 for four rows, and 0·66 for five rows.

For staggered banks of tubes $K = A \times \dfrac{V^{0 \cdot 69}}{D^{0 \cdot 31}}$

where A × 0·70 for two rows ; 0·79 for three rows ; 0·86 for four rows ; 0·91 for five rows ; and 0·99 for ten or more rows.

CONSTRUCTION OF BOILERS.

Boiler Shells.

Mild steel is now almost invariably used for boiler shells. As a rule, in order to provide a margin for wear and tear, plates below $\frac{3}{8}$ in. thick should never be used.

The bending of plates should be done by rolls, and not by hammering. Rivet holes should be drilled after the plates have been bent and fixed together. The plates should then be taken apart and the rough edges removed by a countersinking tool.

Steel for Boilers.

The steel is usually required to be made by the open-hearth acid process. The tensile strength is about 26 to 30 tons per sq. in. for shell plates, and 24 to 28 tons per sq. in. for plates for furnaces, flue tubes, and endplates. An elongation of not less than 23 per cent. is advised.

Strength of Boiler Shells.

Let

P = working pressure in lbs. per sq in.;
T = thickness of plate in inches;
D = diameter (internal) of boiler in inches
E = efficiency of longitudinal joint—i.e. proportionate strength of joint to plate;
F = factor of safety ;
f = tensile strength of plate in lbs. per sq. in.;

then,

$$P = \frac{2E f T}{DF}, \text{ or } T = \frac{PDF}{2Ef}.$$

The factor of safety is usually between 4 and 6. Frequently rules are used in which the safe working stress is substituted for $\frac{f}{F}$ in the above.

For permissible values of f, F, or $\frac{f}{F}$, the rules of the Boiler Survey or Insurance Society concerned should be consulted.

The National Boiler and General Insurance Co. rule is

$$P = \frac{S \times (t - 2) \times K}{D \times 16}$$

where t = thickness of plate in thirty-seconds of an inch, and K = effective area of joint ÷ area of solid plate.

S = 13,100 for double riveted butt joints at longitudinal seams.

S = 12,500 for double or treble riveted lap joints at longitudinal seams.

S = 10,500 for single riveted lap joints at longitudinal seams.

For 'Standard Conditions for Marine Boilers,' see page 34.

The proportionate strength of joint to plate—i.e. the efficiency of the joint—depends upon the type of joint, and also on the pitch and diameter of the rivets, and thickness of plate.

Typical percentage figures for single riveted lap joints are 40 to 58; for double riveted lap 55 to 70; for double butt (a butt strap each side of the plate and double riveted) 72 to 77; for treble riveted butt, each alternate rivet in outer rows being omitted, 82 to 85.

For the method of calculating the efficiency of riveted joints, see page 37.

There is a strong tendency now in favour of butt joints for the longitudinal seams, by which the circular form of the shell is maintained, and there is less liability to grooving. The butt strap also protects the joint from corrosion.

All longitudinal seams break joint, as it is termed—that is, the seam in one ring of plates comes opposite to the solid plate in the next ring.

Furnaces and Flue Tubes.

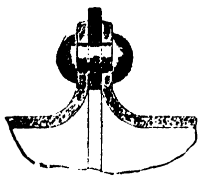

Fig. 5.—Flanged Seam.

For boilers of the Lancashire type, the furnace or flue tubes are usually made with the Adamson flanged seam, with welded longitudinal joints, so that no rivets are exposed to the fire in the furnace.

Increased provision for expansion can be given by making one section of each flue tube of corrugated form.

For proportions of flue sections and thicknesses the rules of the Boiler Insurance Companies should be consulted.

Pressures Required for Riveting and Flanging.

The pressure required for riveting is 75 to 125 tons per sq. in. of rivet area, preferably the lower value. Excessive pressure causes local deformation of the plate. The riveting pressure should be maintained for a short period, so that the rivet may cool down a little, otherwise the spring of the plate may stretch the rivet while it is still hot.

The pressure required for flanging boiler plates may be found from the formula:

$$F = \frac{2\cdot 5\,(L \times t^2)}{L + 10},$$

where,

F = press load in tons; L = length run of flange in feet; t = thickness of plate in sixteenths of an inch. (*Mechanical World Year Book.*)

Welded Pressure Vessels.

In May, 1937, Lloyd's Register of Shipping issued 'Requirements for Fusion Welded Pressure Vessels intended for Land Purposes.' It recognises two classes of welded vessels. The welding of Class I vessels is to be done only by firms which have demonstrated to the Society's surveyors that they are capable of making efficient welds consistently, whilst those of Class II may only be done by firms which satisfy the surveyors that their works are properly equipped for the welding of pressure vessels.

Detection of Fine Cracks in Steel Plates.

A method of detecting cracks in steel plates which are not visible of themselves, is to apply a powerful magnet to the plate, and pour upon its cleaned surface a liquid consisting of a mixture of light oils with finely divided iron in suspension.

The iron dust accumulates along any cracks owing to the change in local distribution of the magnetic flux.

BOILER RULES AND REGULATIONS.

'Standard Conditions for the Design and Construction of Marine Boilers' have been recommended by a committee of representatives of the Board of Trade, British Corporation, Lloyd's Register, and Bureau Veritas. An abstract of the rules drawn up is given in the following pages. For boilers constructed to the requirements of a particular surveying body or boiler insurance company, the rules of the body concerned should be consulted.

Standard Conditions for the Design and Construction of Marine Boilers.*

(Abstract.)

MATERIALS OF CONSTRUCTION.

(Based on British Standards.)

Process of Manufacture.—Structural steel for marine boilers shall be made by the open-hearth process, acid or basic.

Freedom from Defects.—The finished material shall be free from cracks, surface flaws, and lamination. It shall also have a workmanlike finish, and must not have been hammer-dressed.†

Tensile Tests: Plates.—The tensile breaking strength of steel plates for shells and girders, determined from Standard test pieces, shall be between the limits of 28 tons and 35 tons per square inch, but a range of not more than 4 tons per square inch shall be permitted in any one case. For plates intended for flanging or welding, and for combustion chambers and furnaces, the tensile breaking strength shall be between the limits of 26 tons and 30 tons per square inch. The elongation, measured on a Standard test piece having a gauge length of 8 ins., shall be not less than 20 per cent. for material of $\frac{3}{8}$ in. in thickness and upwards, required to have a tensile breaking strength between the limits of 28 tons and 35 tons per square inch; and not less than 23 per cent. for material of $\frac{3}{8}$ in. in thickness and upwards, required to have a tensile breaking strength between the limits of 26 tons and 30 tons per square inch.

Stay Bars.—The tensile breaking strength of longitudinal stays shall be between the limits of 28 tons and 35 tons per square inch, with an elongation of not less than 20 per cent. measured on the Standard Test Piece B, but a range of not more than 4 tons per square inch shall be permitted in any one case. For steel bars for combustion chamber stays the tensile breaking strength shall be between the limits of 26 tons and 30 tons per square inch, with an elongation of not less than 23 per cent. measured on the Standard Test Piece B.

Where stay bars are tested on a gauge length of 4 times the diameter (Test Piece F), the elongation shall be 24 per cent. and 28 per cent. respectively.

The tensile breaking strength of angle and tee bars shall be between the limits of 28 and 32 tons per square inch, with an elongation of not less than 20 per cent. measured on the Standard Test Piece A.

For material under $\frac{3}{8}$ in. in thickness the elongation may be 3 per cent., but not more than 8 per cent., below the above-named elongations.

Wherever practicable the rolled surfaces shall be retained on two opposite sides of the test piece.

Rivet Bars.—The tensile breaking strength of rivet bars shall be between the limits of 26 tons and 30 tons per square inch of section, with an elongation of not less than 25 per cent. measured on the Standard Test Piece B or 30 per cent. measured on the Standard Test Piece F. The bars may be tested the full size as rolled.

Bend Tests: Cold Bends.—Test pieces shall be sheared lengthwise or crosswise from plates or bars, and shall not be less than $1\frac{1}{4}$ ins. wide, but for small bars the whole section may be used. For rivet bars bend tests are not required.

Temper Bends.—The test pieces shall be similar to those used for cold bend tests. For temper bend tests the samples shall be heated to a blood-red colour and quenched in water at a temperature not exceeding 80° F. The colour shall be judged indoors in the shade.

In all cold bend tests, and in temper bend tests on samples 0·5 in. in thickness and above, the rough edge or arris caused by shearing may be removed by filing or grinding, and samples 1 in. in thickness and above may have the edges machined, but the test pieces shall receive no other preparation. The test pieces shall not be annealed unless the material from which they are cut is similarly annealed, in which case the test pieces shall be similarly and simultaneously treated with the material before testing.

For both cold and temper bends the test piece shall withstand, without fracture, being doubled over until the internal radius is equal to $1\frac{1}{2}$ times the thickness of the test piece, and the sides are parallel.

For small sectional material these bend tests may be made from the flattened bar.

Bend tests may be made either by pressure or by blows.

* Published by His Majesty's Stationery Office.

† The Board of Trade has more detailed instructions which it proposes to retain.

Tests for Manufactured Rivets.—**Rivets selected by the Inspector from the bulk shall withstand the following tests:—**

(*a*) The rivet shanks are to be bent cold, and hammered until the two parts of the shank touch, without fracture on the outside of the bend.

(*b*) The rivet heads are to be flattened, while hot, in the usual manner, without cracking at the edges. The heads are to be flattened until their diameter is 2½ times the diameter of the shank.

FORMS OF BRITISH STANDARD TENSILE TEST PIECES.

For Plates and other Structural Material.

TEST PIECE A.

X.—For thicknesses over ⅞ths in.: Maximum width allowed = 1½ ins. Y.—For thicknesses ⅜ths in. to ⅞ths in.: Maximum width allowed = 2 ins. Z.—For thicknesses under ⅜ths in.: Maximum width allowed = 2½ ins.

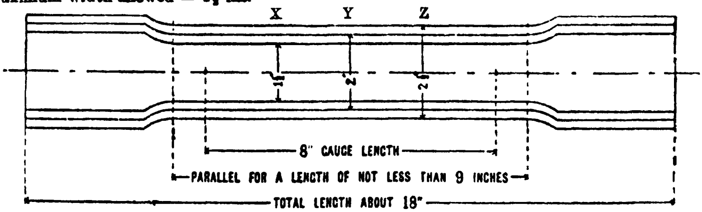

FIG. 6.

NOTE.—It will be observed that the widths given above, being maxima, do not exclude the use of the usual 1½ ins. by 8 ins. test pieces.

TEST PIECE B.

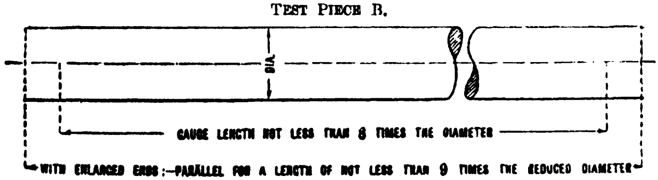

FIG. 7.

TEST PIECE F.

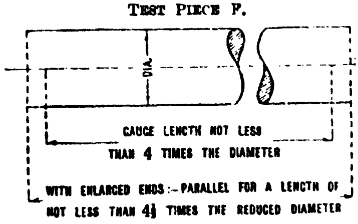

FIG. 8.

SPECIAL IRON FOR SCREW STAYS FOR COMBUSTION CHAMBERS

In order that iron screw stays may be approved of the same size as would be required for mild steel, the iron must withstand the following tests:—

Tensile Tests.—**The tensile breaking strength shall not be less than 21½ tons per square inch, with an elongation of not less than 25 per cent. measured on the Standard Test Piece B, or of 30 per cent. measured on the Standard Test Piece F.**

Bend Tests.—**Test pieces either of the bar as rolled, or turned down to 1 in. diameter, shall stand bending cold until the sides are parallel and the space between the two sides is not greater than the diameter of the test piece.**

Diameters and Thickness of Plain and Stay Tubes.—The following table may be worked to :—

Outside Diameter in Inches.	Standard Thicknesses in L.S.G.				Suitable for Working Pressures of Lbs. per Square Inch.			
Inches.	A	B	C	D	A	B	C	D
2	—	11	10	9	—	155	215	300
2¼	11	10	9	8	140	190	260	315
2½	11	10	9	8	125	175	230	300
2¾	11	10	9	8	110	160	215	275
3	10	9	8	7	140	190	250	300
3¼	10	9	8	7	130	180	230	280
3½	10	9	8	7	120	165	215	260

The minimum thickness of stay tubes measured under the threads shall be ¼ in. for marginal stay tubes, and $\frac{3}{16}$ in. for other stay tubes.

Thickened Ends.—If stay tubes are required to have their thickness increased at the screwed ends, so that the thickness at the bottom of thread is practically the same as in the body of the tube, the thickening is to be attained by upsetting and not by any welding process, and the tubes are to be annealed after the upsetting.

Screwed Threads to be Continuous.—Stay tubes must be screwed at both ends with a continuous thread, which should not be finer than 10 threads per inch. It is desirable, however, that they should be screwed to the standard 9 threads per inch, therefore, after 30th June, 1921, this should be the rule.

MARINE BOILERS, CYLINDRICAL TYPE.

General Conditions as to Design, Workmanship, Hydraulic Test, etc.

In the design and construction of marine boilers the following conditions should be observed :—

Riveting of End Seams.—The riveting of the seams joining the end plates to the cylindrical shell shall be not less than 42 per cent. of that of the solid plate. Where the shell plates exceed ⅝ in. in thickness the seams connecting the shell plates to the end plates are to be double riveted.

Other Circumferential Seams.—The circumferential seam at or near the middle of the length of single-ended boilers should have a strength of joint not less than 60 per cent. of the solid plates. The inner circumferential seams of double-ended boilers should have a strength of joint not less than 62 per cent. of the solid plate. In any case there shall be three rows of rivets when single-ended boilers have shell plates over 1⅜ ins. in thickness, and when double-ended boilers have shell plates over 1$\frac{5}{16}$ ins. in thickness. Where the shell plates exceed ⅞ in. in thickness the intermediate circumferential seams of double-ended boilers are to be at least double riveted.

Welding and other Treatment of Plates.—No steel plates subject to a direct tensional stress are to be welded except where the weld is covered by a butt strap or straps. For small steam domes, by special permission where the welding is done by hammer and the plates do not exceed ⅜ in. in thickness, the straps may be omitted. The strength shall, in such cases, be assumed to be 50 per cent. of that of the solid plate. All steel plates which are welded, dished, flanged, or locally heated, are to be afterwards efficiently annealed.

Butt Straps.—Butt straps must be cut from plates, and not from rolled strip.

Rivet Holes.—All rivet holes must be drilled 'fair,' and as far as possible they should be drilled in place. After drilling the plates the burrs should be removed and the faying surfaces of the plates cleaned, and the sharp outer edges of holes removed also.

Stays not to be Welded. Annealing of Fire-worked Stay Bars.—No steel stays are to be welded. If plus threads are desired, the ends of the stay bars may be upset or the bars may be drawn down in the central portions from bars originally of the size of the ends. In either of these two cases the bars must be subsequently annealed throughout.

In double-ended boilers the through longitudinal stays must be supported at or near the middle of their length.

Threads of Screw and other Stays.—All longitudinal stays and also screw stays should have threads in accordance with the British Standard Specification, and true to pitch, *viz.* :—
All screw stays 1¼ ins. in diameter and upwards should have 9 threads per inch, and all stays 2 ins. in diameter and above passing through plates and secured by nuts on each side of the plate should have not more than 6 threads per inch.

Compensation Rings to Manholes.—Where the cylindrical shell is cut for a manhole, compensation must be provided and must be such that the strength in way of the hole is not less than that required for the longitudinal joint.

Hydraulic Tests of New Boilers.—In all new boilers working at pressures up to 100 lbs. per square inch the hydraulic test must be twice the working pressure. For boilers working at pressures greater than 100 lbs. per square inch the hydraulic test pressure must be $1\frac{1}{2}$ times the working pressure plus 50 lbs. per square inch.

RULES FOR DETERMINING THE WORKING PRESSURE TO BE ALLOWED ON VARIOUS PARTS OF CYLINDRICAL MARINE BOILERS.

CYLINDRICAL SHELLS.

Formula for Working Pressure of the Shell.—For the cylindrical shells of steel marine boilers the maximum working pressure (which is designated by W.P. and is in lbs. per square inch) to be allowed shall be calculated from the following formula :—

Where the thickness of the shell plates does not exceed $1\frac{3}{8}$ ins.,

$$\text{W.P.} = \frac{(t-2) \times S \times J}{C \times D}$$

Where the thickness of the shell plates exceeds $1\frac{3}{8}$ ins. and double butt straps are fitted

$$\text{W.P.} = \frac{t \times S \times J}{2 \cdot 85 \times D}$$

where,

t is the thickness of the shell plates in 32nds of an inch,

S is the minimum tensile strength of the steel shell plates in tons per square inch,

J is the percentage of strength of the longitudinal seams calculated by the methods described below.

C is a coefficient, which is 2·75 when the longitudinal seams are made with double butt straps; 2·83 when the longitudinal seams are made with lap joints and are treble riveted; 2·9 when they are made with lap joints and are double riveted; and 3·3 when they are made with lap joints and are single riveted.

D is the inside diameter of the outer strake of plating of the cylindrical shell measured in inches.

N.B.—The factor of safety must be in no case less than 4.

Methods of Calculating the Strength of Riveted Joints.—The percentage of strength of a riveted joint (J) is found from the following formulæ (I), (II), (III): (I) and (II) are applicable to any type of joint; (III) is applicable only to that type of joint in which the number of rivets in the inner rows is double that in the outer row. The lowest value given by the application of these formulæ is to be taken as the percentage of strength of the joint.

(I) Percentage of strength of plate at joint as compared with solid plate $= \dfrac{100\,(p-d)}{p}$

(II) Percentage of strength of rivets as compared with the solid plate $= \dfrac{100\,(S_2 \times a \times n \times C)}{S_1 \times p \times T}$

(III) Percentage of combined strength of the plate at the inner row of rivet holes and of the rivets in the outer row

$$= \frac{100\,(p-2d)}{p} + \frac{100\,(S_2 \times a \times C)}{S_1 \times p \times T},$$

where,

p = pitch of rivets at outer rows in inches;

d = diameter of rivet holes in inches;

a = sectional area of one rivet in square inches;

n = number of rivets which are fitted in the pitch p;

T = thickness of plate in inches;

C = 1·0 for rivets in single shear as in lap joints;

C = 1·875 for rivets in double shear as in double butt-strapped joints;

S_1 = minimum tensile strength of plates in tons per square inch;

S_2 = shearing strength of rivets, which is taken generally to be 23 tons per square inch, and may be 85 per cent. of the minimum tensile strength of the rivet bar.

Thickness of Butt Straps.—Where the longitudinal seams are fitted with double butt-strapped joints, the outer butt strap should have at least 0·625 of the strength of the plate, and should be of sufficient thickness to permit of efficient caulking of its outer edges. The inner butt strap should be $\frac{4}{32}$ in. thicker than this.

In cases where the number of rivets in the inner rows is double the number in the outer rows this will require the thickness of the outer strap to be

$$t = \frac{5 \times (p-d)}{8 \times (p-2d)} \times T,$$

and that of the inner strap to be at least

$$t_n = \frac{5 \times (p-d)}{8 \times (p-2d)} \times T + \frac{4}{32} \text{ in.}$$

Distances between rows of Rivets and between Rivets and Plate Edges.—In all cases the clear space between a rivet hole and the edge of a plate should not be less than the diameter of the rivet hole—i.e. the centre of the rivet hole should be at least $1\frac{1}{2}$ diameters distant from the edge of the plate.

In joints, whether lapped or fitted with butt straps, in which there are more than one row of rivets and in which there is an equal number of rivets in each row, the distance between the rows of rivets should be not less than $0 \cdot 33p + 0 \cdot 67d$, with zigzag riveting, or $2d$ with chain riveting.

In joints in which the number of rivets in the outer rows is one-half of the number in each of the inner rows, and in which the inner rows are chain riveted, the distance between outer rows and the next rows should be not less than $0 \cdot 33p + 0 \cdot 67d$ or $2d$, whichever is the greater, and the distance between the rows in which there are the full number of rivets should be not less than $2d$.

In joints in which the number of rivets in the outer rows is one-half of the number in each of the inner rows, and in which the inner rows are zigzag, the distance between the outer rows and the next rows should be not less than $0 \cdot 2p + 1 \cdot 15d$, and the distance between the rows in which there are the full number of rivets should be not less than $0 \cdot 165p + 0 \cdot 67d$.

In the above p is the pitch of the rivets in the outer rows.

Maximum Pitch of Rivets in Longitudinal Joints.—The maximum pitch of the rivets in the longitudinal joints of boiler shells is to be—

$$\text{Maximum pitch in inches} = C \times T + 1\tfrac{5}{8} \text{ ins.}$$

where T is the thickness of the plate in inches and C is a coefficient as given in the following table:—

Number of Rivets per Pitch.	Coefficients for Lap Joints.	Coefficients for Double Butt-strapped Joints.
1	1·31	1·75
2	2·62	3·50
3	3·47	4·63
4	4·14	5·52
5	—	6·00

FURNACES.

Plain Furnaces.—The working pressure to be allowed on plain furnaces or furnaces strengthened by Adamson or other joints, and on the cylindrical bottoms of combustion chambers, is to be determined by the following formulæ, the least pressure obtained by either formula being taken:—

$$\text{W.P.} = \frac{C\,(t-1)^2}{(L+24) \times D}, \text{ or W.P.} = \frac{C_1}{D} \times [10\,(t-1) - L].$$

where D is the external diameter of the furnace or combustion chamber bottom in inches;

t is the thickness of the furnace plate in 32nds of an inch;

L is the length of the furnace or of combustion chamber bottom or the length between points of substantial support, in inches, measured from the centres of rivet rows or from the commencement of flange curvature, whichever is applicable;

C = 1,450 where the longitudinal seams are welded, and 1,300 where they are riveted;

C_1 = 50 where the longitudinal seams are welded, and 45 where they are riveted.

W.P. = working pressure in lbs. per square inch.

Corrugated Furnaces.—The working pressure to be allowed on corrugated furnaces is to be determined by the following formula:—

$$\text{W.P.} = \frac{C(t-1)}{D},$$

where D is the external diameter measured at the bottom of the corrugations in inches;

t is the thickness of the furnace plate in 32nds of an inch, measured at the bottom of the corrugation or camber;

C is a coefficient which is 480 for the Fox, Morison, Deighton, Purves, and other similar furnaces, 510 for the Leeds Forge Bulb Suspension furnace.

N.B.—No furnace, plain or corrugated, should exceed $\frac{13}{16}$ in. in thickness.

FLAT PLATES SUPPORTED BY STAYS SECURED IN VARIOUS WAYS.

Flat Plates supported by Screwed Stays.—The working pressure to be allowed on flat plates supported by stays is to be calculated by the following formula:—

$$\text{W.P.} = \frac{(t-1)^2 \times C}{a^2 + b^2},$$

but if steel of less tensile strength than 26 tons per square inch is used the working pressure allowed shall be correspondingly reduced.

In this formula,

W.P. is the working pressure in lbs. per square inch;

t is the thickness of the flat plate in 32nds of an inch;

a is the distance apart of the rows of stays in inches;

b is the pitch of the stays in the rows in inches;

C is a coefficient which varies with the method of fixing the stays as follows:—

Where the plates are exposed to flame and the stays are screwed into the plate and their ends are riveted over, C = 50.

Where the plates are not exposed to flame and the stays are screwed into the plate and their ends are riveted over, C = 57.

In these cases the thickness of the plate must be at least half the diameter of the stay required by the rule.

Where stay tubes are screwed into tube plates and expanded, C = 52. If they are fitted with nuts, C = 72.

Where the plates are exposed to flame and the stays are screwed into the plate and fitted with nuts on the outside, C = 75. Where the plates are not exposed to flame, C = 86.

Where the stays pass through the plates not exposed to flame and are fitted with nuts inside and outside, C = 96.

Where plates are stiffened by flanging, the inner radius of which is not greater than $2\frac{1}{2}$ times the thickness of the plate, for the support thus given C = 110 when the plates are not exposed to flame, and C = 96 where they are exposed to flame. The pitch is to be reckoned from the commencement of the curvature.

GIRDERS.

Girders Supporting Combustion Chamber Tops.—For girders supporting the tops of combustion chambers the following formula is to be used:—

$$\text{W.P.} = \frac{C \times d^2 \times t}{(L-P) \times D \times L} \times \frac{S}{28},$$

where,

d is the depth of the girder at centre in inches;

t is the thickness of the girder at centre, when this is a forging, or the sum of the thicknesses of the plates when the girder is made of two plates, measured in 32nds inch;

L is the length of the girder in inches, measured between the tube plate and back chamber plate inside, or between tube plates in chambers common to two opposite furnaces;

P is the pitch of stays supported by the girder, in inches;

D is the distance apart of the girders, centre to centre, in inches;

S is the minimum tensile strength of the steel plates forming the girder, in tons per square inch. In the case of forged girders S is to be taken as 24 for iron and 28 for steel;

C is a coefficient as follows:—

$$C = \frac{n}{n+1} \times 495,$$

when the number of stays in each girder is odd, and

$$= \frac{n+1}{n+2} \times 495,$$

when the number of stays in each girder is even, n being the number of stays to each girder.

STAYS.

Iron or Steel Screw Stays to Combustion Chambers.—For screw stays with threads not coarser than 9 threads per inch, made of steel or of special wrought iron tested to the requirements, the following formula is to be used:—

$$\text{W.P.} = \frac{(d - 0{\cdot}267)^2 \times 8{,}250}{a},$$

where

d is the diameter of the stay over the thread in inches;

a is the area in square inches supported by one stay.

But in no case must the stress exceed 9,000 lbs. per square inch of section.

Steel Longitudinal Stays.—For steel longitudinal stays with threads not coarser than 6 threads per inch, the working pressure is to be calculated from the following formula:—

$$\text{W.P.} = \frac{(d - 0{\cdot}340)^2 \times 9{,}500}{a} \times \frac{S}{28},$$

where d is the diameter of the stay over the thread in inches;

a is the area in square inches supported by one stay;

S is the minimum tensile strength of the steel in tons per square inch.

But in no case must the stress exceed 11,000 lbs. per square inch of section, when steel of a minimum tensile strength of 28 tons per square inch is used.

In cases where longitudinal stays are made with enlarged ends, and the body of the stay is smaller in diameter than at the bottom of the thread, and in cases where coarser threads than 6 per inch are used, the working pressure is to be calculated from the following formula:—

$$\text{W.P.} = \frac{(d_1 - 0{\cdot}125)^2 \times 9{,}500}{a} + \frac{S}{28},$$

where d is the diameter of the stay at the bottom of the thread or at the smallest part of the body.

Stay Tubes.—On stay tubes, whether of wrought iron or of lap-welded steel, a working stress of 7,500 lbs. per square inch of the net sectional area at the bottom of the thread is permitted.

Dished Ends (Convex Outside).—For ends of steam chests, etc., dished to partial spherical form, the following formula is to be used:—

$$\text{W.P.} = \frac{15 \times S(t-1)}{R},$$

where

W.P. is the working pressure in lbs. per square inch;

t the thickness in 32nds of an inch;

R the inner radius of curvature of the end in inches, which shall not exceed the diameter of shell;

S the minimum tensile strength of plates in tons per square inch.

Safety Valves.—At least two safety valves must be fitted to each boiler. They must be arranged so that the springs and valves are cased in, that the valves cannot be overloaded when steam is up, that they can be lifted by easing gear, and turned round on their seats by hand, and in case of fracture of springs they cannot lift out of their seats.

The easing gear must be arranged to lift all the safety valves on a boiler together, and must be workable from some accessible place free from steam danger.

All the safety valves of each boiler may be fitted in one chest, which must be separate from any other valve chest and which must be connected direct to the boiler by a strong and stiff neck, the passage through which should have a cross-sectional area at least equal to one-half the aggregate area of the safety valves in the chest.

Each safety valve box shall have a means of draining it; the pipe shall lead to the bilge or tank clear of the boiler.

Size of Safety Valves.—The minimum aggregate area of the safety valves of the ordinary type in each boiler, whether coal-fired or oil-fired, and whether working under natural, forced or induced draught, shall be found by the following formula:—

Aggregate area of safety valves in square inches

$$= \text{Total heating surface of boilers in square feet} \times \left(\frac{K}{p + 15}\right),$$

where p is the working pressure in lbs. per square inch.

For coal-fired boilers K is 1·25, for oil-fired boilers and boilers with closed stokehold forced draught K = 1·5.

Accumulation Test of Safety Valves.—All safety valves must be set to the required pressure under steam. During a test of 15 minutes with the stop valves closed, and under full firing conditions, the accumulation of pressure must not exceed 10 per cent. of the loaded pressure. During this test no more feed-water should be supplied than is necessary to maintain a safe working water level.

Superheater Safety Valves and Drains.—Where a superheater is fitted which can be shut off from the boiler it must have a separate safety valve fitted with easing gear, etc. The valve as regards construction must comply with the regulations for ordinary safety valves, but the easing gear may be fitted to be workable from the stokehold only.

The superheater must be fitted with a drain cock or valve to free it from water when necessary.

BOILER TUBES.

Approximate Weight of Iron Boiler Tubes.

PER FOOT RUN.

External Diameter.	11 S.W.G.	10 S.W.G.	9 S.W.G.	8 S.W.G.	¼-inch	5/16 inch.
Ins.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
2½	2·89	3·18	3·55	3·92	5·89	7·16
2¾	3·20	3·51	3·93	4·34	6·54	7·97
3	3·50	3·85	4·30	4·75	7·20	8·79
3¼	3·80	4·18	4·68	5·17	7·85	9·61
3½	4·11	4·52	5·06	5·59	8·50	10·4
3¾	4·41	4·85	5·43	6·01	9·16	11·2
4	4·72	5·19	5·81	6·43	9·81	12·0
4¼	5·02	5·52	6·19	6·85	10·4	12·8
4½	5·33	5·86	6·57	7·27	11·1	13·7
4¾	—	6·19	6·94	7·69	11·7	14·5
5	—	6·53	7·32	8·11	12·4	15·3
5¼	—	6·86	7·70	8·52	13·0	16·1
5½	—	—	8·07	8·94	13·7	16·9
5¾	—	—	8·45	9·36	14·4	17·8
6	—	—	8·83	9·78	15·0	18·6

Surface of Boiler Tubes.

PER FOOT RUN.

External Diameter.	Surface.	External Diameter.	Surface.	External Diameter.	Surface.
Ins.	Sq. Ft.	Ins.	Sq. Ft.	Ins.	Sq. Ft.
3/4	·1964	1 3/4	·4582	3 1/4	·8509
7/8	·2291	1 7/8	·4909	3 1/2	·9163
1	·2618	2	·5236	3 3/4	·9818
1 1/8	·2945	2 1/8	·5563	4	1·047
1 1/4	·3273	2 1/4	·5891	4 1/2	1·178
1 3/8	·3600	2 1/2	·6545	5	1·307
1 1/2	·3927	2 3/4	·7200	5 1/2	1·440
1 5/8	·4254	3	·7584	6	1·571

British Standard Specification for Charcoal Iron Lapwelded Boiler Tubes.*

(No. 43—1927.) (*Abstract.*)

1. The tubes shall be lapwelded and shall be made from genuine charcoal iron of the best quality. 2. The tubes shall be sound, clean, smooth and well finished, free from surface defects, rust and longitudinal grooving, both internally and externally, and the ends shall be clean and square. 3. The tubes shall be carefully annealed at both ends. 6. The tubes or strips cut from the tubes shall show a tensile strength between the limits of 19 and 24 tons per square inch inclusive, with a contraction of area of the metal of not less than 45 per cent. 7. The tubes shall stand bulging both hot and cold with a parallel drift (fig. 9) without showing either crack or flaw, until the diameter of the bulged end exceeds the original diameter of the tube by not less than 15 per cent. when tested hot and by not less than 10 per cent. when tested cold. 8. A piece

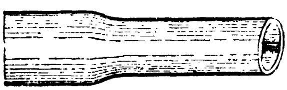

FIG. 9.

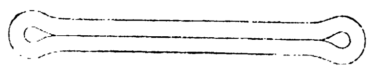

FIG. 10.

of tube 2 ins. long shall be placed on end and shall stand hammering down until it is reduced to 1 3/8 in. high, without showing crack or flaw. 9. The tubes shall, when cold, stand flattening (fig. 10) without showing crack or flaw until the interior surfaces of the tubes meet. 10. Each tube shall be tested by an internal hydraulic pressure of at least 750 lbs. per sq. in., and any tubes failing to stand this test shall be rejected.

British Standard Specification for Cold Drawn Weldless Steel Tubes for Locomotive Boilers.*

(No. 53—1927.) (*Abstract.*)

Quality of Material.

1. The tubes shall be cold drawn and weldless, and shall be manufactured from steel of the best quality made by the Open-Hearth process. They shall not show on analysis more than 0·035 per cent. of sulphur or more than 0·030 per cent. of phosphorus. The manufacturer shall supply an analysis when required to do so.

Freedom from Defects.

2. The tubes shall be perfectly sound, clean, smooth, and well finished, free from surface defects, rust and longitudinal grooving, both internally and externally, and the ends shall be clean and square.

Annealing.

3. The tubes shall be carefully annealed throughout their length after the operation of drawing and the ends shall be carefully annealed after the process of enlarging or decreasing the diameter.

* By permission of the British Standards Institution.

Permissible Variation in Weight.

4. The actual weight of each tube shall not differ from the calculated weight by more than 5 per cent. above or $2\frac{1}{2}$ per cent. below the weight calculated from the specified dimensions, allowing 489·6 lbs. per cubic foot of metal in the tubes.

Tensile Test.

6. The tubes or strips cut from the tubes shall, without further annealing, show a tensile strength of not less than 20 tons per sq. in. nor more than 26 tons per sq. in. with an elongation of not less than 28 per cent. in 8 ins. for tubes, and not less than 22 per cent. in 8 ins. for strips cut from the Tubes. If strips are cut for testing from tubes over $2\frac{1}{2}$ ins. diameter they may be annealed before testing.

Crushing down Test.

8. A piece of tube 2 ins. long shall be placed on end and shall, when cold, stand hammering down until it is reduced to $1\frac{1}{4}$ ins. high, without showing either crack or flaw.

Flattening Test.

9. The tubes shall, when cold, stand flattening (fig. 10), without showing either crack or flaw until the interior surfaces of the tubes meet or, when the tubes are over 11 S.W.G. in thickness, until the interior surfaces are brought to a distance apart equal to the thickness of the wall of the tube.

Hydraulic Test.

10. Every tube shall be tested by an internal hydraulic pressure of at least 1,000 lbs. per sq. in., and any tubes failing to stand this test shall be rejected.

Tubes for Water-Tube Boilers and Superheaters.

The 'Board of Trade Instructions as to the Survey of Passenger Steamships,' 1928, states :—

The minimum thickness of tubes for pressures up to 350 lbs. per sq. in. shall be determined by the following rule :

$$t = \frac{\text{W.P.} \times d}{F} + 7$$

where W.P. = working pressure in lbs. per sq. in.

d = external diameter in ins.

t = thickness in $\frac{1}{100}$ths of an in.

For the two rows of tubes next the fire and round the gaps formed in the nest of tubes for the outflow of hot gases from the fire F = 55. For all the other tubes F = 75.

Superheater tubes.—The mean thickness of the heating tubes shall be determined from the formula $t = \frac{\text{W.P.} \times d}{75} + 5$. The symbols have the meanings given above.

The recent extensions of design to plants to work at very high pressures call for the development of a new formula for high pressure and high temperature combined.

Dorey proposes the following formula for high pressure water-tube boilers for marine purposes :

For fire row tubes $\quad t = \left[\frac{\text{W.P.} \times d_1}{100} + \frac{4000}{\text{W.P.}}\right] \frac{5000}{5000 + \text{W.P.}}$.

For ordinary row tubes $\quad t = \left[\frac{\text{W.P.} \times d_1}{100} + \frac{2000}{\text{W.P.}}\right] \frac{5000}{5000 + \text{W.P.}}$.

where t = thickness in hundredths of an inch,

W.P. = working pressure in lb. per sq. in.

d_1 = external diameter of tube in inches.

The terms $\frac{4000}{\text{W.P.}}$ in the first formula, and $\frac{2000}{\text{W.P.}}$ in the second are not to exceed 11 and 7 respectively.

CHIMNEY AND FLUE DIMENSIONS.

Apart from any consideration of production of draught, the height of a chimney must be such that the smoke and noxious gases are discharged at a level at which they will not be a nuisance. The minimum height is usually fixed by the bye-laws of local authorities.

For relation between draught in inches of water gauge and chimney height see page 22.

The area of the chimney depends on the amount of coal burned per hour on the grates of the boilers which it serves, and on the velocity of the gases in the chimney, and their temperature.

The gas velocity again depends on the draught pressure, and on the frictional resistances in the flues and chimney. Thus the area of the chimney for a given size of boiler or battery of boilers should take into account many factors, some of which are not capable of exact determination. This accounts for the empirical character of the rules given by various authorities, and for the variations in them.

For typical conditions, say about 600° F. chimney temperature, flues not abnormal in length, and of area about 20 to 40 per cent. greater than chimney area, and 25 to 28 lbs. of coal burned per sq. ft. of grate area per hour, the following rules give suitable areas:—

H = height of chimney in feet; A = area at top of chimney in sq. ft.; G = grate area.

$$A = \frac{k\,G}{\sqrt{H}};$$

here $k = 1\cdot5$ for one boiler, $1\cdot0$ for two, $0\cdot8$ for three, $0\cdot7$ for four, $0\cdot5$ for five or more.

At 28 lbs. coal per sq. ft. of grate area per hour this corresponds to

$$A = \frac{k\,W}{28\sqrt{H}};$$

where W = lbs. coal burned per hour.

For a 100-ft. chimney, and $k = 1\cdot4$,

A = 0·5 sq. ft. per 100 lbs. coal burned per hour

Other formulæ are:

(*a*) $$A = \frac{W}{14\sqrt{H}}$$

(evidently for lower rate of combustion than the above, as it would agree with $k = 1\cdot5$ and rate of combustion 21).

(*b*) **Area in sq. in.** $= \dfrac{\text{lbs. coal per hour} \times 12}{\sqrt{H}}$ . . . . . (*Bourne*).

or $A \text{ (sq. ft.)} = \dfrac{W}{12\sqrt{H}}$

(*c*) **Area in sq. ins. = 1·5 per lb. coal per hour** . . . . (*Bourne*).

or $A = \dfrac{1\cdot5}{144}W$

corresponding to $k = 1\cdot5$ if $H = 100$ and combustion rate $= 14\cdot4$).

(*d*) **Area in sq. ins.** $= \dfrac{\text{Area of grate}}{\sqrt{H} \times 1\cdot58}$ . . . . (*Elswick*)

or $A = \dfrac{1}{1\cdot58}\dfrac{G}{\sqrt{H}}$

corresponding to $k = 0\cdot63$).

(*e*) **Kent gives formulæ which correspond to the following:—**

Effective area E = A — 0·6 √A;

$W = 16\cdot65\,E\sqrt{H}$; $\quad D = 13\cdot54\sqrt{E} + 4$;

$E = \dfrac{0\cdot06\,W}{\sqrt{H}}$; $\quad S = 12\sqrt{E} + 4$;

where,

D = diam. of round chimney in inches; S = side of square chimney in inches.

The table on page 1419 has been calculated from the foregoing formulæ of group (*e*).

(*f*) **Sometimes the area of the chimney top is taken as $\frac{1}{8}$ to $\frac{1}{10}$ that of fire grate, the height not being taken into account.**

Flue areas should be about 20–40 per cent. greater than chimney area.

For stability of chimneys see Section XIII, page 464 (Vol. I).

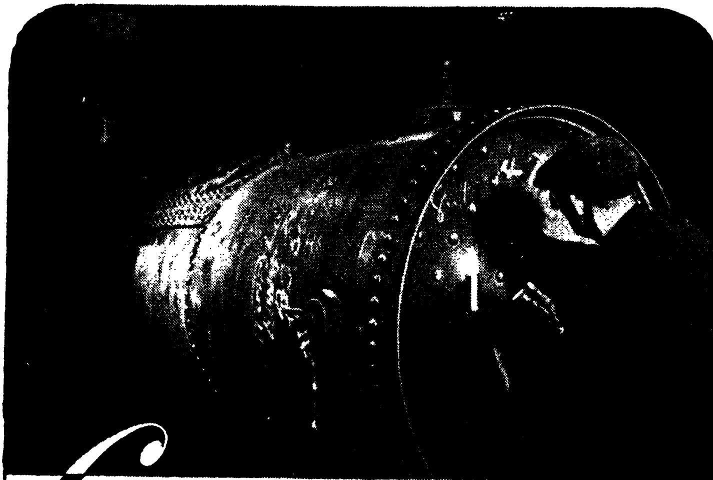

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TABLE OF SIZE OF CHIMNEYS FOR BOILERS.

Diameter. Inches.	Area A in sq. feet.	E.	Height of Chimney in Feet.														Side of Equivalent Square Chimney.
			50	60	70	80	90	100	110	125	150	175	200	225	250	300	
			Pounds of Coal burned per Hour.														
18	1·77	0·97	115	125	135	145											16
21	2·41	1·47	175	190	205	220											19
24	3·14	2·08	245	270	290	310	330										22
27	3·98	2·78	325	360	390	415	440										24
30	4·91	3·58	420	460	500	535	565	595									27
33	5·94	4·48		575	625	665	705	745	780								30
36	7·07	5·47		705	760	815	865	910	955	1,020							32
39	8·30	6·57			915	980	1,040	1,095	1,145	1,225							35
42	9·62	7·76			1,080	1,155	1,225	1,290	1,355	1,445	1,580						38
48	12·57	10·44				1,555	1,650	1,740	1,825	1,945	2,130						43
54	15·90	13·51					2,135	2,245	2,360	2,515	2,755	2,975					48
60	19·64	16·98					2,680	2,825	2,965	3,160	3,460	3,740					54
66	23·76	20·83						3,470	3,640	3,880	4,245	4,590	4,905				59
72	28·27	25·08						4,175	4,380	4,670	5,115	5,525	5,905	6,265			64
78	33·18	29·78							5,190	5,535	6,060	6,550	7,000	7,425	7,825		70
84	38·48	34·76							6,070	6,470	7,090	7,655	8,180	8,680	9,150	10,025	75
90	44·18	40·19								7,480	8,195	8,850	9,465	10,040	10,580	11,590	80
96	50·27	49·01								8,560	9,380	10,135	10,835	11,490	12,115	13,270	86
102	56·75	52·23								9,720	10,650	11,500	12,290	13,045	13,750	15,060	91
108	63·62	58·63								10,450	11,995	12,960	13,855	14,695	15,490	16,965	96
114	70·88	65·83									13,425	14,500	15,500	16,440	17,330	18,985	101
120	78·54	73·22									14,930	16,130	17,240	18,285	19,275	21,115	107
132	95·03	89·18									18,185	19,645	21,000	22,275	23,480	25,720	117
144	113·10	109·72									21,760	23,505	25,130	26,655	28,090	30,775	128

TABLE V.

PROPORTIONS OF BRICK CHIMNEYS.

(*Edwin Danks & Co., Ltd.*)

Chimney Height above Ground Level.	No. of Bricks Thick at Bottom.	Thickness at Bottom.	Thickness of Lining at Bottom.	Clearance of Lining.	Diameter of Base of Circular Chimneys for internal diameters ranging from 2′ 6″ to 10′ 0′.								
Ft.		Ft. In.	Ins.	Ins.	Ft. In. 2 6	Ft. In. 3 0	Ft. In. 4 0	Ft. In. 5 0	Ft. In. 6 0	Ft. In. 7 0	Ft. In. 8 0	Ft. In. 9 0	Ft. In. 10 0
20	1	9	—	—	—	—	—	—	—	—	—	—	—
40	1½	1 2	—	—	—	—	—	—	—	—	—	—	—
60	2	1 7	4½	3½	7 0	7 6	—	—	—	—	—	—	—
80	2½	1 11	4½	3½	7 6	8 0	9 0	—	—	—	—	—	—
100	3	2 4	9	3½	9 3	9 9	10 9	11 9	—	—	—	—	—
120	3½	2 9	9	6	10 6	11 0	12 0	13 0	14 0	—	—	—	—
140	4	3 1½	9	6	—	11 9	12 9	13 9	14 9	15 9	—	—	—
160	4½	3 6	9	6	—	—	13 6	14 6	15 6	16 6	17 6	—	—
180	5	3 10½	9	6	—	—	—	15 3	16 3	17 3	18 3	19 3	—
200	5½	4 3	9	6	—	—	—	—	17 0	18 0	19 0	20 0	21 0
220	6	4 7½	9	6	—	—	—	—	—	18 9	19 9	20 9	21 9
240	6½	5 0	9	6	—	—	—	—	—	—	20 6	21 6	22 6

NOTES.—Under no circumstances must the height of a circular chimney exceed 12 times the diameter of the shaft at ground level. The corbelling at the bottom and below ground level must be at least equal to the thickness of the shaft brick work, and the angle of the corbelling must not be less than 45°. The batter of the chimney must not be less than 2½ ins. in diameter for every 10 ft. of height. The thickness of the brickwork must not be less than that given in the table above, with half-brick offsets every 20 ft. It is advisable that an access door should be placed on the base of each chimney to allow for cleaning.

Notwithstanding the above information, the design of a brick chimney must comply with the local by-laws of the district in which it is to be erected.

THE EMISSION OF DUST AND ASH FROM CHIMNEYS.

The following notes are largely based upon the Report of a Technical Committee appointed by the Electricity Commissioners to study the various methods adopted in Great Britain and on the Continent for the suppression of the dust nuisance from power station chimneys.

Power station chimneys should have a height not less than two and a half times that of the highest point of the station roof, or adjacent building. With this height it is found experimentally that such dust as is discharged will be carried away by an air-stream which will not subsequently come into contact with the ground. Sulphurous gases from the fuel will likewise be rendered harmless. With chimneys of the height stipulated there should be no need for any dust-extraction apparatus in stoker-fired plants provided that the gas velocities and the rate of combustion are moderate. Where these conditions do not exist, or where low-grade fuel is burnt, the most suitable apparatus for eliminating the dust is the multiple cyclone type of extractor. For pulverised fuel installations, some form of dust extraction plant is always necessary, and the most suitable type is the electrostatic precipitator. While 'wet' types of dust extractors are capable of preventing the dust nuisance, they have many practical disadvantages.

THE CONCENTRATION OF GRIT IN FLUE GASES.

A stoker-fired plant burning a fuel with an ash content of 12 per cent. may pass to the chimney 10 per cent. of the ash in the fuel, or about 0·3 grains of dust per cub. ft. of flue gas. A pulverised fuel plant burning similar fuel might pass to the chimney 70 per cent. of the ash or about 2·62 grains per cub. ft. of gas. There are methods of coal cleaning which will reduce the ash content, and therefore the dust emission to a very low figure, but although practically ash-free coal might be advisable for pulverised fuel plants, fuels with a very low ash content are found by experience to make the conditions too severe for mechanical stokers.

The solid particles in the flue gases are usually graded according to their size in microns (0·001 millimeter) as determined by standard sieves. This method, however, is very unsatisfactory as sieve measurements are unreliable below 63 microns, and impossible below 40 microns, while they ignore the most important factor in creating a dust nuisance, namely the specific gravity of the particles. For these reasons grading by means of air-elutriation is preferable. The dust produced by pulverised fuel plant contains about 50 per cent. of particles less than 20 microns in diameter, and as these will remain suspended in the air almost indefinitely they are unlikely to cause a nuisance. An extraction plant should therefore be capable of eliminating practically all particles over this size, besides a good proportion of those smaller. A modern electrostatic plant will remove 90 per cent. of all dust, and 93 per cent. of that which escapes will be less than 30 microns in diameter.

The quantity of dust in flue gases cannot be calculated from the ash content of the fuel as given by analysis because dissociation of the ash takes place at furnace temperature, resulting in a loss of weight that may amount to 10 per cent. The efficiency of a dust extraction plant has therefore to be found by determining the weight of dust per cub. ft. of gas before and after the apparatus. This is done by drawing a sample of the gas through a filter and weighing the dust deposited on the filter.

Types of Dust Extraction Plant.

There are four types of dust extraction plant which may be referred to as (1) the water-film type; (2) the combined spray and water-film type; (3) the electrostatic type, and (4) the cyclone type. The first two depend upon the use of water, while in the others the gases are dry and uncooled throughout. The wet systems are cheaper to instal and operate provided that the effluent can be disposed of without the use of costly filtration plant. They have the effect of reducing the gas temperature to 140° F. or less at the base of the stack, but the induced draught fans cannot take advantage of this as the fans have to be placed at the inlet side of the dust extractor to avoid corrosion from the wet gases, which is more serious than the erosion by the grit in the untreated gases. In the dry systems the fans may be placed at the outlet of the extractor where they only have to handle clean non-erosive gases.

Water Film Type of Dust Extraction Plant.

In one of the best-known designs of this type, the elements consist of hollow four-sided vertical cast iron pipes, arranged in staggered formation in a horizontal flue. Water is supplied to the interior of each pipe and overflows in a continuous trickle down the outsides, which are slightly concave in plan. The gas passes transversely among the set of pipes and the dust collected by their wet sides is washed down to a tank beneath. The draught loss is about 0·75 in. of water, and the gases leave at a temperature of about 120° F.

Combined Water Spray and Film Type of Plant.

In a typical plant of this kind the gases enter a chamber in which they are first subjected to the action of a series of atomising water sprays. They are then turned upwards and backwards through a nest of staggered vertical baffles. The latter are V-shaped in cross-section, and are kept wet by secondary water jets playing on their upper ends. The effluent is in the form of a fine slurry containing about 1·5 per cent. of solid matter. The draught loss is about 1·0 in. w.g. and the outlet temperature of the gases about 140° F.

Electrostatic Type of Dust Extractor.

In electrostatic dust extractors the flue gas is caused to pass through tubes or between planes which form the collecting electrodes. These are maintained at earth potential, and receive the dust extracted from the gas. Down the centre of each pipe, or between adjacent plates, is a discharge electrode, consisting of a wire or rod furnished with discharge points. The discharge electrodes are maintained at a constant negative potential of 60,000 to 70,000 volts. The dust particles in the gas are ionised by the brush discharge from these electrodes and consequently travel across to the surface of the collector electrodes where they give up their charges. The particles then fall by gravity into the hoppers below, any tendency for the dust to adhere to the surfaces being counteracted by a hammer gear which raps the electrodes from time to time. The high negative voltage required for the discharge electrodes is obtained by a transformer and rectifier controlled by a low tension switchboard. The power consumption is very low, amounting only to about 2 kW.h. per million cub. ft. of gas cleaned. The draught loss is about 0·25 in. w.g. and the temperature of the gas is practically unchanged by passing through the apparatus. The draught fans are placed on the outlet side of the plant so that they work with clean gas only and are thus not subjected to erosion. The dust is delivered dry and in very fine particles. It may be removed in this state or carried away by flushing with water, as may be most convenient.

The Cyclone Type of Dust Extractor.

The essential feature of this type of extractor is a casing of volute shape into which the dust-laden gases enter tangentially. Beneath the volute are a number of truncated cones alternating

with cylindrical parts, terminating in a dust outlet at the bottom. The dirty gases travel downwards in a spiral path, throwing out their solid particles by centrifugal force against the inner wall of the casing. The core of clean gas at the centre of the spiral flows upwards to the central outlet at the top. The fan handles the clean gas leaving the apparatus and so works under the best conditions as regards erosion. The draught loss amounts to about 2 ins. w.g. and the heat loss is very small. For dealing with the gases from a 270,000 lb. pulverised fuel boiler, a dozen cyclones may be used in parallel, some of them being shut off during periods of light load.

The Removal of Sulphur from Flue Gases.

Untreated flue gases contain not only grit and dust, but also sulphurous gases derived from the sulphur in the coal burnt, and these gases form acids which have a very deleterious effect upon vegetation and buildings. Although there can be little doubt that by far the greater part of the sulphurous gases in the atmosphere is derived from domestic fires, there is a justifiable demand that power stations in urban areas shall not add to the pollution of the air and many plants have incurred heavy expenses in their efforts to comply with this demand. From the findings of the Committee appointed by the Electricity Commissioners to enquire into the question of dust and grit emission, it would seem that sulphurous gases would not be harmful if discharged from chimneys having a height of at least two and a half times that of the highest point of the nearest roof, as they would then enter a current of air that would not subsequently come into contact with the ground. At certain important power stations, however, such as those at Battersea and Fulham, elaborate flue gas washing apparatus has been installed. At Battersea washing is effected with river water, and also with limed water for the absorption of the sulphur compounds. Thames water is used, and as the liquid effluent is returned to the water, the plant has to include means of rendering this effluent innocuous.

The gases from the boilers pass first through grit arresters, then through 'primary chambers' where they are sprayed with water to remove any dust before they enter the main flue. This flue runs the length of the boiler house, and terminates in large chimney towers at each end. Banks of scrubbers, formed of steel channels are placed in the main flue. The sulphites first obtained are oxidised to sulphates by contact with the iron oxide on the surfaces of the channels, and by the use of aerated wash water, which is re-circulated. From the main flue the gases pass into downtakes in the chimney towers, where they come into contact with limed water and wooden scrubbers. The gases then enter the uptakes leading to the chimneys, passing through more scrubbers supplied with alkaline water, before being finally discharged. The effluent is treated to limit the salts in solution, and also filtered before being sent back to the river. The Battersea flue gas washing plant is described in a paper read by Messrs. Hewson, Pearce, Pollitt and Read before the Society of Chemical Industry in July 1933. During the year 1938 this plant removed 3,680 tons of sulphur from a total of 465,000 tons of coal burnt.

The gaswashing plant at Fulham is of the I.C.I.—Howden type, and is characterised by the absence of any effluent. The gases are scrubbed with an alkaline liquor from which the solids are settled out, and removed as sludge. A plant of this type, installed at the Swansea power station, was described in a paper by Pearson, Nonhebel and Ulander, published in the *Journal of the Institute of Fuel*, 1935.

The Treatment of Boiler Feed Water.

The treatment of raw water to make it fit for boiler feeding is a matter that can only be decided on in the light of a chemical analysis of the water concerned. Expert advice on the subject can always be obtained from the reputable firms who specialise in water-softening, and such firms, moreover, will always be ready to guarantee the results of the treatment they recommend. There are, nevertheless, certain things that all operating engineers should know about water, including a number of simple tests that can be carried out in any plant.

The scale-forming properties of water are due to its hardness The total hardness is made up of the so-called 'temporary hardness' and 'permanent hardness.' The temporary hardness, which can be removed by boiling, is due to the presence of bicarbonates of calcium and magnesium in solution, while the permanent hardness, which cannot be disposed of in this way, is caused mainly by the sulphates, chlorides and nitrates of the same elements. The hardness of water, whether temporary, permanent, or total, is measured by 'Degrees Clark,' one degree denoting a hardness equal to that which would be caused by the presence of one grain of calcium carbonate per gallon. This is equivalent to 14·2 parts per million. The total hardness of the public water supply of 50 British towns has been found to range from 0·80° to 39·18°, the average hardness being 13·20°. As a general rule all waters with more than a few degrees of hardness should be given chemical treatment before being used as feed, even for boilers of the tank type, while extremely soft natural waters should also be looked on with suspicion.

There are on the market innumerable 'boiler compounds' which their makers recommend should be mixed with the raw water with the object of preventing scale, or at least of ensuring that any scale should be of a soft and easily removable nature. These compounds are usually dribbled at a constant rate, into the feed-water in a heater of the open type, in the hope that, on

the average, the right amount will be supplied. Boiler compounds differ greatly in composition, their principal ingredients, apart from water, being usually soda ash, caustic soda, tri-sodium phosphate, and often some organic material such as tannin which has the property of causing the lime salts to be precipitated as a soft sludge. When water requires softening, it is better and cheaper to do it outside the boiler by means of apparatus that can be relied on to dose the water with known chemicals in the exact quantities required to give the correct results. The feeding of chemicals into the boiler is apt to cause foaming and priming as well as involving excessive blowing down.

The Soap Test for Hardness of Water.

The simplest way of testing water for hardness consists of shaking up a sample of the water with increasing quantities of standard soap solution, and noting the quantity of the soap solution required to form a permanent lather. To carry out the test, 70 c.c. of the water are placed in a stoppered bottle, into which the soap solution is run, 1 c.c. at a time, from a graduated burette. The stopper is replaced, and the bottle violently shaken after each addition of soap. The total hardness of the water, in Clark degrees, is given by the number of c.c. of soap solution used to form a permanent lather on the water after shaking up. The standard soap solution can be obtained from any firm of manufacturing chemists, and is standardised by testing with water of standard hardness.

In carrying out the soap test there are two points that need attention. Firstly the addition of soap solution to the sample must be continued until a lather is formed that will last for five minutes. The reason for this is that with some waters a temporary lather may be formed which will quickly disappear, and must not be mistaken for the true lather. The second point of importance is the necessity of taking care that the hands are clean and that the fingers are not used in place of the stopper to close the bottle when it is shaken, for the slightest contamination of the sample with oil or grease will prevent a lather being formed.

The Lime and Soda Water Softening Process.

The commonest method of softening water is by means of the lime and soda process. The treatment is carried out in automatic water softeners which add a properly adjusted mixture of lime and soda solution in proportion to the quantity of water passing through the apparatus. The amounts of lime and soda required are best determined by a previous analysis of the water. The lime serves to destroy the temporary hardness, while the soda destroys the permanent hardness. If the water is treated cold, the softener should be large enough to hold a four hours' supply. The process is very much more rapid when the water is hot. A smaller softener can therefore be used if the water is heated, which is sometimes done by turning exhaust steam into the softener itself.

When the temporary and permanent hardness of the water are known, the approximate quantities of lime and soda needed may be found from the following formula:

Lbs. lime per 1,000 galls. water = 0·096 times the temporary hardness.
Lbs. soda-ash per 1,000 galls. water = 0·159 times the permanent hardness.

Allowances of 20 per cent. in the case of lime and 5 per cent. in the case of soda-ash are included in the formulæ to cover impurities of the commercial materials.

A more accurate way of determining the quantities of lime and soda required, is to calculate them from the analysis of the water. This can be done by the use of the following table, which gives the weight in lbs. of pure calcium hydrate, $Ca(HO)_2$, and of pure carbonate of soda Na_22CO_3, theoretically required per 1,000 gallons of water, to neutralise one grain of the various impurities per gallon.

Impurity.	Lbs. required for 1,000 gallons for each grain of impurity per gallon. Calcium Hydrate.	Sodium Carbonate.
Sulphate of Calcium	—	0·111
Sulphate of Magnesia	0·088	0·126
Nitrate of Magnesia	0·071	0·102
Chloride of Magnesia	0·111	0·159
Nitrate of Calcium	—	0·092
Chloride of Calcium	—	0·136
Free Carbonic Acid	0·240	—
*Carbonate of Calcium	0·105	—
*Carbonate of Magnesia	0·126	—

* These two items are to be disregarded if free CO_2 is listed in the analysis.

The weights of the reagents calculated from the table are the weights of the pure chemicals theoretically required. The totals obtained should be increased by 10 per cent. in order to make sure that the water is fully softened, and is definitely alkaline. When purchasing lime and soda-ash for use in water softeners, guarantees should be obtained as to the actual percentages of calcium hydrate and of sodium carbonate in the respective commercial materials, and these percentages must, of course, be taken into account when deciding on the quantities to be used. The lime should contain not less than 10 per cent. of calcium hydrate, and not more than 3 per cent. of magnesium salts.

When the proper amounts of soda and lime have been decided on, the materials are mixed together with water in the proportion of about 1 lb. of mixture to 1 gallon of water, and it is this solution which is automatically dosed into the raw water by the softener.

A water properly softened by the lime and soda process should have a total hardness not exceeding 2°, and should be of the correct alkalinity. The alkalinity is tested in the following manner. A 70 c.c. sample of the softened water is placed in a white porcelain dish, and two drops of phenolphthalein indicator are added. This should turn the water purple, if it does not do so the water needs more lime and soda. Standard sulphuric acid solution, of strength N/50 is dropped slowly into the purple water from a graduated burette, the water being stirred meanwhile, until the colour vanishes. The number of c.c. of acid used, as read from the burette, is called the alkalinity to phenolphthalein. The same sample of water is now coloured yellow by adding 2 drops of methyl orange indicator, and more acid is dropped in until the yellow colour changes to a faint pink. The number of c.c. of acid required to effect this change is called the alkalinity to methyl orange, and the total number of c.c. of acid used in the two tests is called the total alkalinity. The alkalinity to phenolphthalein should be slightly more than half the total alkalinity. If the alkalinity to phenolphthalein should exceed 3°, *i.e.* if more than 3 c.c. of acid are required to discharge the purple colour, both lime and soda should be reduced. If the methyl orange alkalinity should exceed 2°, the lime should be increased and the soda decreased.

A properly softened water will have a total hardness of between 1 and 2°, an alkalinity to phenolphthalein of about 2·5°, and an alkalinity to methyl orange of a further 2°.

The alkalinity due to sodium hydroxide is called 'Caustic Alkalinity' and is equal to the difference between twice the phenolphthalein alkalinity and the total alkalinity.

Question.—What quantities of lime and soda would be required to soften water having the following analysis.

Silica	0·19
Oxide of Iron	0·15
Sulphate of Lime	4·92
Carbonate of Lime	17·50
Sulphate of Magnesia	3·78
Sulphate of Sodium	3·40
Chloride of Sodium	4·10
	34·04 grains per gallon.

Hardness . . 24·25°.

Answer.—The silica, oxide of iron, chloride of sodium and sulphate of sodium may be disregarded, as they play no part in the hardness of the water. For the other constituents, we see from the table on p. 49 that the quantity of pure calcium hydrate required would be:

For the carbonate of lime	17·50 × 0·105 = 1·817 lb. per 1,000 gallons
For the sulphate of magnesia	3·78 × 0·088 = 0·333 lb. per 1,000 gallons
	2·150

The amount of pure sodium carbonate would be:

For the sulphate of lime	4·92 × 0·111 = 0·546 lb. per 1,000 gallons
For the sulphate of magnesia	3·78 × 0·126 = 0·476 lb. per 1,000 gallons
	1·022

Increasing these quantities by 10 per cent. to ensure complete softening, we find that 2·356 lb. of calcium hydrate, and 1·124 lb. of sodium carbonate are needed for every 1,000 gallons of water.

Supposing that the commercial lime available contains 80 per cent. of pure calcium hydrate, and that the commercial soda-ash contains 98 per cent. of pure sodium carbonate, the final requirements are:

Lime = 2·365/0·80 = 2·95 lb. per 1,000 gallons of water.
Soda = 1·124/0·98 = 1·15 lb. per 1,000 gallons of water.

CAUSTIC EMBRITTLEMENT OF BOILER PLATES.

The failure of boiler plates by cracking between and around rivet holes has been the subject of much investigation during recent years. The cracking has nothing to do with the quality of the plate, and since it takes place between the crystals and not through them, it cannot be attributed to cold working of the metal. The generally accepted explanation is that, when steel under stress is in contact with a concentrated solution of caustic soda at boiler temperature, the soda is decomposed with the evolution of hydrogen which causes the embrittlement of the metal. The caustic soda which gives rise to the mischief, is produced by the decomposition of the soda-ash in the boiler water. The fact that caustic embrittlement only takes place along riveted seams is explained by the possibility of an exceptionally high concentration of caustic soda occurring in conjunction with a high stress at such places.

To prevent caustic embrittlement, it has been recommended that the ratio between sodium sulphate and total alkalinity should be not less than unity for boiler pressures up to 150 lb., than 2 for pressures from 150 to 250 lb., and not less than 3 for higher pressures. It should be noted that the total alkalinity rather than the caustic alkalinity is taken into account because of the conversion of carbonate of soda into caustic soda in the boiler.

The maintenance of the above ratios has proved effective in preventing embrittlement in a large number of cases. Further investigations carried out by the U.S. Bureau of Mines have shown that embrittlement may be prevented by maintaining the concentration of sodium chloride in the boiler water at a value greater than 0·6 times the total alkalinity, and the sodium sulphate concentration in excess of the total alkalinity. For pressures from 500 to 1,400 lb. per sq. in. it appeared necessary to have soluble salts containing iron and aluminium oxides in the water, in concentrations greater than 0·6 times the concentration of sodium silicate present, for sodium silicate had been found to be conducive to embrittlement.

In discussing these results, F. J. Matthews, in *Boiler Feed Water Treatment* (Hutchinson), states that chloride is present in most boiler waters. The requisite sulphate introduction varies according to the method of treatment. The lime-soda process usually requires the soda ash to be carried to slight excess. If the sulphate content is high enough before treatment to keep the required ratio with this excess of soda ash, no additive treatment is necessary. If not, sodium sulphate should be added. Acid treatment should not be adopted in the lime-soda process. The soda ash is carried to excess to facilitate softening, and if sulphuric acid were added this excess would be reduced and softening retarded.

When the sodium-zeolite treatment is used, if the carbonate hardness is higher than the sulphate, the treated water will be higher in sodium carbonate than in sodium sulphate. The usual procedure has been to acid treat the water in such a case. When natural waters, having free sodium carbonate and low sulphate are used, they are usually treated with lime and then sufficient sulphuric acid to maintain the ratio, or by lime and aluminium or iron sulphate. In all cases care must be taken not to overtreat with acid or sulphate salts.

Among other agents which have the power of inhibiting embrittlement, tannic acid has been said to be fifty times as effective as sulphate, and sodium phosphate to be 1,500 times as effective as sulphate. It should be remembered, however, that the only phosphate that counts is that actually present in solution in the boiler water. This must be kept at over 4 parts per 100,000 for boilers working up to 250 lb. pressure, and the same concentration is said to be effective at higher pressures when the alkalinity of the water is kept below 130 parts per 100,000. In view of certain contradictory experiments, however, sodium sulphate is the inhibitor most frequently employed.

DETERMINATION OF CAUSTIC ALKALINITY OF BOILER WATER.

A method of determining the caustic alkalinity of water by means of the phenolphthalein and methyl orange tests has already been given. An alternative method, given by F. H. Mathews, is as follows. Take 100 c.c. of the boiler blow-down sample, and add 25 c.c. of a 10 per cent. solution of barium chloride. Dilute the mixture with water if necessary, so that a pink colour is produced when a few drops of phenolphthalein are added. Add N/10 nitric acid drop by drop from a burette until the pink colour just disappears. The number of c.c. of acid required, multiplied by 5, gives the parts of caustic alkalinity expressed as parts of calcium carbonate per 100,000.

This method may also be used for determining caustic alkalinity in softened water, but in this case the quantity of barium chloride should be reduced to 10 c.c.

DETERMINATION OF TEMPORARY HARDNESS OF RAW WATER.

Take 70 c.c. of the raw water, add a few drops of methyl orange, and drop in N/10 nitric acid drop by drop from a burette, until the colour just changes from yellow to pink. Five times the number of c.c. of acid used gives the temporary hardness in Clark degrees, or in grains of calcium carbonate per gallon.

The Conditioning of Boiler Feed Water.

Conditioning is the term used to denote a supplementary treatment of feed water designed to ensure that such deposits as occur in the boiler shall be of the less troublesome kind, such as calcium carbonate and calcium phosphate which do not cause a hard scale. An excess of sodium carbonate will bring about this state of affairs, but unfortunately at modern boiler pressures the sodium carbonate dissociates and forms sodium hydroxide thus causing an undue amount of caustic alkalinity in the boiler. To avoid this difficulty, sodium phosphate has been used. This salt results in a soft scale, but unless it is pumped directly into the boiler, it is liable to react with the residual hardness in the feed water and form deposits in the feed piping or the economiser. Sodium metaphosphate can be used without any of the disadvantages of the tri-sodium phosphate, and is now extensively employed for water conditioning. It can be added to the feed water without risk of trouble with the feed pipes or economiser, as it delays the precipitation of the calcium salts until the boiler is reached, and then brings them down as sludge. It protects the pipes from corrosion and has the further advantage of dissolving any calcium scale already formed in them, while its action in the boiler is to reduce the alkalinity, and therefore the amount of salts in the water necessary to maintain the required sulphate-alkalinity ratio. The use of 2 parts of sodium metaphosphate per million parts of water is said to be sufficient to prevent deposits in feed-lines, etc.

Water Softening by the Zeolite Process.

The zeolite used for water softening is an artificially prepared hydrated alumino-silicate in granular form. When hard water is passed through it, the calcium and magnesium salts are replaced by corresponding salts of sodium, with the result that the water is completely softened. When the zeolite ceases to be effective, its softening power can be fully restored by flushing with an ordinary brine solution, and washing out the calcium chloride thereby formed.

A zeolite softener is a closed steel cylinder coated with protective enamel. It is half filled with zeolite, and provided with the necessary connections for operating and flushing. The Permutit Company have introduced a completely automatic plant in which regeneration takes place every time a predetermined quantity of water has been treated.

Zeolite softening produces zero hardness. No adjustment is needed for variation in the quality of the raw water, and no chemicals are used except the salt required for regeneration. No precipitate is formed, and the whole apparatus is simple and easy to operate.

On the other hand, the zeolite softened water will actually contain something like five per cent. more solids in solution than the raw water, whereas softening by the lime and soda process effects a great reduction in the dissolved solids. The zeolite method of softening therefore tends to increase the concentration of salts in the boiler, and consequently, the risk of foaming, priming, corrosion due to the CO_2 evolved, and caustic embrittlement.

The zeolite process is not applicable to acid, ferruginous, or turbid waters, and is not advisable for waters with much temporary hardness. To avoid the high alkalinity produced in the latter case, the softened water is sometimes treated with sulphuric acid, and the carbon dioxide formed is removed by steam. A better way still is to do the main part of the softening by the lime and soda process and to use zeolite softening only to remove the residual hardness.

Water Conditions Accepted as Good Practice.

The following conditions are given as representative of good practice by Messrs. Babcock & Wilcox, Ltd.

Low and Medium Pressure Boilers: Feed Water.—Dissolved oxygen should be less than 0·1 c.c. per litre; chlorine not over 0·3 grains per gallon. Alkalinity should be definitely caustic soda alkalinity, but not more than 0·5 grain per gallon, and should have a hydrogen ion concentration *p*H of about 8·4. Total solids should be kept to the minimum possible amount.

Boiler Water Conditions.—Caustic soda alkalinity should be not greater than 50 grains per gallon; total solids not greater than 500 grains per gallon; chloride not over 30 grains per gallon. Hardness as near zero as possible. Oil, no indication of even the slightest trace. In no case should the ratio of sodium sulphate to alkalinity be less than the figures stipulated on p. 51 in the section on 'Caustic Embrittlement.'

For high-pressure boilers (600 lb. per sq. in. and over) feed water conditions:

Dissolved oxygen	0·02 c.c. per litre (maximum value)
Caustic soda alkalinity	0·1 grain per gallon "
Sodium Sulphate	0·4 " " " "
Other salts	0·2 " " " "
Hardness	Nil.
No Oil.	

Boiler water conditions:	600 to 850 lb. per sq. in.	850 to 1,500 lb. per sq. in.
Caustic Soda, grains per gallon maximum	25	15
Sodium Sulphate " " "	100	60
Sodium Phosphate " " "	5 to 10	5 to 10
Other salts " " "	15 to 20	15 to 20
Total maximum	150	100

The Elimination of Oil from Feed Water.

It is of the greatest importance that feed water should be free from oil, the merest trace being a source of danger with modern water-tube boilers. When condensate from reciprocating engines is used as a boiler feed it is always likely to be contaminated by the oil used for the cylinder lubrication. A mechanical oil-separator, which must be of adequate size, placed in the exhaust pipe of the engine will remove a great part of the oil before it reaches the condenser, and such a separator is often relied on in industrial plants. But if the condensate contains oil in emulsion, it must be effectively purified before being used as boiler feed. The oil particles in an emulsion are so minute that the oil will neither settle out, nor can it be removed by any ordinary process of filtration, unless previously coagulated. This can be done by the addition of sulphate of alumina and soda-ash to the water. The effect is to produce a flocculent precipitate which is easily filtered out. A de-oiling apparatus, to supply measured quantities of these reagents automatically to the oily water is often combined with the ordinary lime-soda softening plant. Alternatively, the oil in emulsion may be removed by an electrical process of de-oiling. The water, in this case, is caused to flow slowly between iron plates across which a small difference of potential is maintained. The oil is then deposited on the plates in the form of sludge.

BOILER INSPECTION.

Boilers must be inspected by a competent person at least once in every fourteen months. The owner or user may be made responsible for the effects of an explosion.

To prepare a boiler for inspection the manhole and mudhole doors should be removed, the blow-off cock closed, the steam valve locked to prevent tampering and danger to cleaners and inspectors, the hot feed valve locked, the bars and bridges removed from the furnace, and the flue doors removed and the hearth pit plates lifted away.

The flues and boiler plates in the flues should be swept clean of soot and dust.

The amount and nature of the internal scale should be observed, and the boiler thoroughly cleaned inside and scale removed.

Brickwork must be removed to the extent required by the person making the examination. It should, in any case, be removed to the extent necessary to expose the seams of shell boilers and headers of water-tube boilers, not less frequently than once in every six years in the case of boilers situated in the open or exposed to weather or damp, and not less frequently than once in ten years in the case of every other steam boiler.

The special points to be observed by the inspector are the condition of the plates along the seatings and all points of contact with bricks, the dryness of the flues, the absence of lime mortar near the plates, the dryness of the top covering, leakages at seams, circularity of furnaces, blisters, signs of overheating. Internally: grooving at plate junctions, around the angle irons of shell or furnace tube connection to end plates, at the root of angle irons or flanged seams or other points of stress or flexure, tautness of gusset stays, easy slackness of longitudinal stay bolts if present, condition of plates as to cleanliness, corrosion &c., condition of low-water appliances, scum troughs, feed pipes, gauge openings, and generally all points bearing on the safety of the boiler that may by some unlikely and hard to conceive contretemps lead to mishap of any sort.

In old boilers the openings in the shell should be looked to in order to ascertain the weakening due to removal of so much plate, and the degree of strength given back by added rings or other reinforcements. It is also desirable to note the quality of the attendance and workings of the boiler periodically, and to test the accuracy of the steam gauge and working of the safety valves.

The foregoing remarks apply more particularly to shell boilers. For water-tube boilers, the caps of the water tubes must all be removed, and each tube examined and seen through for scale deposit, and as far as possible externally for blisters. Locomotive-type boilers are not easy to inspect, should never have riveted shell seams below water line, and must be largely judged by the appearance of such parts as are visible. Every few years the shell should be examined internally, the fire tubes being removed for the purpose. The accessibility of a boiler for complete inspection ought to be a considerable factor in determining the choice of a boiler, and more particularly so when the water is dirty or corrosive in quality.

Experience is necessary to make a good inspector. He must possess judgment and sense of proportion, and be able to order a boiler not to be worked until further notice, without it being possible for further enquiry to show that there were not good grounds for his fears.

BOILER LEGISLATION.

The Boiler Explosions Acts, 1882 (45 & 46 Vic. ch. 22) and 1890 (53 & 54 Vic. ch. 35) provide for inquiry into boiler explosions. The Acts extend to the whole of the United Kingdom. The term 'boiler' means any closed vessel used for generating steam, heating any liquid, or receiving steam for heating, steaming or boiling, excepting boilers used exclusively for domestic purposes or in the service of his Majesty. On the occurrence of a boiler explosion, notice must be sent within 24 hours to the Board of Trade by the owner, user, or person acting for him. This notice to state the name of premises or works, postal address, day and hour of explosion, number of persons killed and injured, description of boiler, purpose used for, particulars of failure, working pressure, by whom last inspected, and by whom insured.

The Act of 1890 makes the Act of 1882 applicable to sea and mine boilers.

BOILER FITTINGS.

Valve bodies, tees and other fittings should be of cast steel for superheated steam, and also for saturated steam if the pressure exceeds 120 lb. per sq. in.

Injectors.

To Compute the Diameter of Throat and Volume of Discharge.

d = diameter of throat in millimetres; V = volume of water required per hour in gallons; P = pressure of steam in pounds per sq. in.

$$d = \cdot 709\sqrt{\frac{V}{\sqrt{P}}} \qquad V = 1\cdot 985\, d^2 \sqrt{P}.$$

Example.—The pressure of steam is 200 lbs., and the volume of water required to be delivered per hour is 1,010 gallons.

Then $\cdot 709\sqrt{\frac{V}{\sqrt{P}}} = \cdot 709 \times 8\cdot 45 = 6$ mms.; and $1\cdot 985\, d^2 \sqrt{P} = 1\cdot 985 \times 36 \times 14\cdot 142 = 1{,}010$ gals.

The above particulars apply to all properly proportioned live steam injectors.

Notes on Injectors.

A very frequent cause of trouble is that by one means or another the injector becomes overheated and refuses to lift its water. This may be due to steam leaking past the check valve into the body of the injector, and then passing to the suction pipe, so that the latter becomes too hot. Under these conditions it is impossible to make the injector work. Another cause of overheating is sometimes to be traced to the proximity of the fitting to the hot boiler shell, which allows heat from the latter to pass to it by conduction and radiation. In the first case the remedy is to fit a well-made check valve, combined with a plug cock, between the injector and the boiler. This should be done, even if the former is designed with a self-contained back-pressure valve. Further, the plug cock should be shut off when the injector is not in use. In the second instance, it sometimes happens that an injector is employed which is fitted with its own back-pressure valve, and it is then bolted directly to the boiler. This is very bad practice, as a single valve is not dependable in the first place, and a further point is, that the injector is much too close to the hot boiler. This is a fruitful cause of trouble. Also, should anything go wrong with the injector, it is necessary to blow down the boiler before it can be put in order; whereas, with a combined check valve and plug cock, the latter can be closed, and the injector removed under steam.

An injector will not work satisfactorily if supplied with wet steam; therefore, in order to ensure that this is as dry as possible, it should be taken from the highest available point in the boiler.

Another defect sometimes arises through the check valve not having sufficient lift, or on account of some sort of obstruction in the delivery pipe. The effect of this is that the injector will lift the water, but on account of the greater pressure due to the obstruction against which it has to work, it will be unable to force the water into the boiler, which will consequently find an escape through the overflow instead.

In some cases the water supply to the injector is taken from a tank placed beside the boiler, on the boiler-house floor, the supply to the tank being from an external source, and controlled by a ball cock with a float attached, so that the level of the water in the supply tank is constant. This is a very satisfactory arrangement. In other instances the source of supply may be some little distance away, and connected to the injector by means of pipes. Further, if the water is taken from a well the level may vary between considerable limits. Hence, when purchasing an injector, it is of the utmost importance that the manufacturer should be given all possible data, such as the greatest static head and the length of pipe with the number of bends, so that he can obtain the total suction lift and supply an injector to suit the conditions.

Much trouble will be experienced if the feed-water is allowed to become hot. The lower the temperature the better. There are many so-called hot-water injectors, but even these will not prove very satisfactory if the temperature of the feed-water is allowed to rise too near to the maximum temperature specified by the makers. In order to prevent as much as possible any rise in temperature, it is of the greatest importance that the overflow from the injector should not be allowed to drain back into the feed-tank. An overflow pipe should be fitted that will lead this hot water clear of the feed supply. Such a pipe should not be closed, but should enter an open cup, similar to the overflow of the circulating water in a gas-engine jacket. It is then under constant observation of the man in charge, and if any irregularity occurs it is quickly detected.

Occasionally an injector will refuse to work, due to a leak in the suction pipe. If this defect is present it may be located by fixing a plug in the suction pipe and opening the steam supply to the injector, when the position of the leak will be easily detected.

Lastly, dirt and grit are a source of trouble at times. To prevent this as far as possible, a straining rose should be fitted to the suction pipe. This will not exclude all grit, and occasionally small grains of solid matter suspended in the feed-water will find their way through and lodge on the internal surface of the injector. Also if the water contains soluble salts of magnesium and calcium, it becomes heated on passing through the injector, and some of these salts are deposited, forming incrustations which may cause the injector to refuse to work, either because it has become totally choked up, or else small particles of the incrustations may break away and become lodged under the back-pressure valve or check valve and prevent its closing, when the injector will refuse through overheating.

It is always advisable to keep a spare injector in stock, and if the water contains a considerable amount of impurity, instructions should be given to change the injector at regular periods. The length of time that one should be allowed in continual service must be governed by experience with the particular conditions under consideration. In order to loosen the deposit the injector may be left to soak in diluted hydrochloric acid.

If the above very easily followed points are given attention, injectors will be found most reliable, and few complaints will be heard.

('*Mechanical World.*')

SAFETY VALVES:

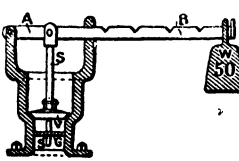

FIG. 11. B, lever; A, short arm of lever; SS, stem; V, valve; G, guide; W, weight.

WEIGHT ON SAFETY VALVES.

If A = area of valve in sq. ins., P = pressure in lbs. per sq. in., l_1 = distance of valve from fulcrum, l_2 = length of lever, w_1 = weight of lever in lbs., l_3 = distance of centre of gravity of lever from fulcrum, w_2 = weight of valve and stem in lbs.

Then W, the weight required at the end of the lever

$$= \frac{P \times A \times l_1 - w_1 \times l_3 - w_2 \times l_1}{l_2}.$$

SAFETY VALVE AREAS.

Since the steam production of a boiler is sensibly the same at all pressures, and since the density of steam increases very nearly as the pressure, and since steam escapes at an outlet at practically constant volume, it appears that safety valves, like boiler stop valves, may be reduced in size as pressure increases. Ordinary valves have a very small lift. Special valves in which the steam reaction is utilised to increase the lift will give a much greater opening. If a valve lift a height equal to a fourth of its diameter the area of opening is equal to the valve area, and no greater lift would give greater output.

It is not sufficiently well known that boilers working at a low pressure require larger safety-valves than boilers at a high pressure. Thus, for example, a boiler working at 15 lbs. by the steam gauge requires three times the area of safety valve opening that is required by a boiler at 75 lbs. This is the more important when it is remembered that in a boiler which is safe only at low pressure, an additional 15 lbs. pressure doubles the strain, whilst in the boiler at 75 lbs. a rise of 15 lbs. adds but a fifth to the stress on the plates. This point as to the safety-valves on old low-pressure boilers is too little known or recognised.

Safety-valve areas should be inversely as the absolute pressure.

In well-proportioned dead-weight safety valves it is usually expected that if a pressure of 60 lbs. per sq. in. opens the valve, a pressure of 70 lbs. will keep it sufficiently open for the escape of all the steam produced.

The former Board of Trade rules are largely followed, and as they are more detailed, bringing out many points of importance, extracts are given below, also extracts from the rules of Lloyd's and Bureau Veritas.

BOARD OF TRADE RULES FOR SAFETY VALVES.

(Extracts from.)

The valve chest should be placed directly on the boiler; the neck, or part between the chest and the flange which is bolted to the boiler, should be as short as possible, and cast in one with the chest.

The minimum aggregate area of the safety valves should not be less than that obtained from the following formula, but in no case should valves less than $1\frac{1}{2}$ ins. in diameter be passed without the special sanction of the Board of Trade.

$$A = \text{THS} \times \frac{k}{p + 15}$$

where A = aggregate valve area in sq. ins.;
THS = total heating surface of the boiler in sq. ft.
p = working pressure in lbs. per sq. in.;

$$k = \begin{cases} 1 \cdot 25 \text{ for coal-fired, cylindrical boilers in open stokeholds.} \\ 1 \cdot 5 \text{ for coal-fired, cylindrical boilers with closed stokehold and forced draught} \\ 1 \cdot 5 \text{ for oil-fired, cylindrical boilers.} \\ 1 \cdot 1 \text{ for coal-fired, water-tube boilers with natural draught.} \\ 1 \cdot 25 \text{ for coal-fired, water-tube boilers with forced draught.} \\ 1 \cdot 25 \text{ for oil-fired, water-tube boilers.} \end{cases}$$

The safety valves should be fitted with lifting gear, so arranged that the two or more valves on any one boiler can at all times be used together, without interfering with the valves on any other boiler. The gear should be arranged so that it can be worked by hand from some accessible place free from steam danger, and the arrangement should permit of the valves being turned round on their seats by hand.

Care should be taken that the safety valves have a lift equal to one-fourth of their diameter; that the opening for the passage of steam to and from the valves, including the waste-steam pipe, have each an area not less than 1·1 A, and the area of the main waste-steam pipe should not be smaller than the combined area of the branch pipes. Each valve box should have a drain-pipe fitted at its lowest part.

In the case of lever valves, if the holes in the lever are not bushed with brass, the pins must be of brass; iron and iron working together must not be used.

Too much care cannot be devoted to seeing that there is proper lift, also that free means of escape for the waste steam are provided, as it is obvious that unless the means for escape of the waste steam are ample, the effect is the same as reducing the area of the valves or putting an extra load upon them. The valve seats should be secured by studs and nuts.

The valves should have a clearance in the seats of at least $\frac{1}{32}$ in. on the diameter, and should not project below.

For spring loaded safety valves, (1) at least two valves are to be fitted to each boiler, (2) the valves are to be of the prescribed size, (3) the springs and valves are to be so cased in and locked up that they cannot be tampered with, (4) provision is to be made to prevent the valves lifting out of their seats in the event of the springs breaking, (5) screw lifting gear is to be provided to ease all the valves, (6) the size of the steel of which the springs are made is to be in accordance with the formula $d = \sqrt[3]{\frac{S \times D}{C}}$, where S = load on spring in lbs., D = diameter of spring (from centre to centre of wire) in inches, d = diameter, or side of square, of wire in inches, C = 8,000 for round steel, 11,000 for square steel, (7) the springs are to be protected from the steam and impurities issuing from the valves, (8) when valves are loaded by direct springs, the compressing screws are to abut against metal stops or washers when the loads sanctioned by the Surveyor are

on the valves, (9) the springs are to have a sufficient number of coils to require a compression under the working load of at least one-quarter of the diameter of the valve.

The valves are to be tested under full steam, and full firing, with the feed water shut off and stop valve closed. The duration of the tests for accumulation of pressure shall be, on cylindrical boilers, 15 mins., on water-tube boilers, for as long as the water supply in the boiler permits, with a maximum of 7 mins.

Lloyd's Rules for Safety Valves.

(Abridged.)

At least two safety valves are to be fitted to each boiler, arranged so that springs and valves are cased in, that valves cannot be overloaded while steam is up, that the valves can be lifted by easing gear and turned round on their seats, and that valves cannot lift out of their seats if the springs break.

Vertical boilers with 100 square feet or more of heating surface to have two safety valves each not less than 1·5 in. diameter; those with less than 100 square feet heating surface one valve not less than 2 ins. diameter.

All safety valves of one boiler may be fitted in one chest connected direct to boiler by passage having cross-sectional area not less than half the aggregate area of the valves. Valve chest to be drained. The minimum aggregate area of safety valves of a boiler, coal- or oil-fired, in square inches

$$= \text{total heating surface of boiler in square feet} \times \frac{C}{p+15},$$

where

p = pressure in lbs. per square inch;
C = 1·25 for boilers using coal;
C = 1·5 for boilers using oil and for those with closed stokehold forced draught.

Waste-steam pipe to have area not less than 1·1 times area given by above formula.

Safety valves to be set under steam. During test of fifteen minutes with stop valve closed and full firing conditions accumulation of pressure is not to exceed 10 lbs. per square inch.

Bureau Veritas Rules and Regulations for Safety Valves.

At least two spring safety valves of an approved design must be fitted to each main boiler The total area of the safety valves is given by the formula

$$A = \frac{8{\cdot}7}{\sqrt[4]{(P-18)^3}},$$

where

A = sectional area of safety valve in sq. ins. per sq. ft. of grate surface; P = working pressure in lbs. per sq. in. No safety valve to be less in diameter than 1¼ in.

When forced draught or oil fuel is provided for, the area is to be increased in proportion to the increased evaporative power of the boilers.

Suitable arrangements and gear to be fitted in connection with the safety valves, whereby they may be lifted from the deck as well as from the stoke-hole floor.

Defects in Water Gauges.

As ordinarily fitted, the gauge is usually attached directly to the boiler and provided with three taps, one connecting to the steam space, the second to the water space, and the third controlling the discharge to atmosphere. The latter should always discharge inside the boiler-house, where the effluent can be seen, and not be as it sometimes is, with a mistaken idea of cleanliness, coupled to a waste pipe, led outside, where the evidence of its action is not visible when the attendant opens the tap. It is of value to have duplicate gauges, as the cost is comparatively trifling, and one gauge serves as a check on the other. All three taps should be tested independently, that is to say, the bottom tap should be shut while the top one is opened and *vice versâ*, the top one shut while the bottom is opened, so as to give a thorough 'blow through' of the steam and water connections to the boiler. Unless this is done there is no certainty of the gauge being in working order, while there is risk of the top and bottom connections becoming choked and causing a false reading. More than one serious accident has been traced to defects in the taps themselves, owing to the holes in the plugs of the taps not coinciding with the thoroughfare of the fitting, so that when the position of the handle suggests the tap is open it is in reality nearly closed. Another cause of false readings is the blocking of the passage in the glass tube through the indiarubber packing ring working over the ends of the glass, owing to the glass being cut too short and the gauge not being provided with a nipple in the bottom of the gland to protect the glass against this risk. Having regard to the serious results which may attend overheating from shortness of water, it is advisable whenever possible to supplement the water gauge with a fusible plug, and if the boiler is a large one, with a low-water safety-valve as well. (*Mechanical World.*)

GAUGE GLASSES.

It is important to have the fittings exactly in line with each other, and to have them large enough so as not to cause the glass to bind or to be cramped in the least. When the fittings are not in line the glass is apt to be cramped either when putting it into the sockets in the fittings or when tightening the packing nuts. Wrenches should not be used for tightening the packing nuts around the gauge glass. When a glass is packed with the proper kind of packing and the nuts have been screwed up just tight enough to prevent leakage when both the glass and the packing have been warmed, the glass may readily be turned around in the fittings with merely the thumb and forefinger.

After putting in a new glass turn on a very little steam, leaving the drip at the bottom of the glass open so that the steam will gradually fill the glass and escape through the drip-cock. The glass is thus warmed thoroughly before the water is let into it and before it is subjected to pressure. When the glass has become thoroughly hot, open the steam valve wider, then open the water valve a like amount, and continuing to open both valves alike until fully open, then slowly close the drip at the bottom. This method not only allows the fittings to be thoroughly blown out, but enables the operator to increase the pressure on a new glass and new packing more gradually. It is best to anneal all the glasses before using them. This is generally accomplished by means of a water and steam bath, the water and the glasses being heated by steam. It is cheaper in the long run to cut glasses with a gauge-glass cutter than

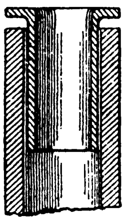

FIG. 12.

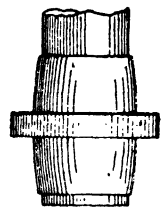

FIG. 13.

to attempt to cut or break them off with a broken file, or by similar means, because a slight nick in the glass at the wrong place may mean the early destruction of the glass when subjected to pressure and varying temperature. Some engineers fit a thin brass or copper thimble into the upper end of the tube, as shown in fig. 12, which prevents scale and water from coming in contact with the glass when blowing out the latter, and hence it reduces the eroded and pitted appearance commonly seen at the upper end of unprotected glasses. Tubes made of aluminium may be obtained of dealers in engineers' supplies which accomplish the same result.

These tubes and thimbles, it will be seen, protect the inner surfaces of the glass only, and as the glass is liable to be crowded to one side or the other in the packing nut as well as in the fitting itself, it is a good plan to also protect the glass on the outside.

A simple means of protecting the glass on the outside is also obtained from dealers in gauge glasses. The arrangement consists of a rubber sleeve adapted to slide over the end of the glass, as shown in fig. 13. The glass cannot come in contact with the socket in the fitting, neither can the packing nut touch the glass when the packing is screwed up. The combination of the thimble and the latter device prevents the rapid wear of the glass at the top, and also retains the glass clear of all metallic contacts, which has been found to be very desirable where glasses are subjected to high pressure and temperature and sudden changes in the latter.

When the rubber protector is employed, separate packing rings are not needed, because the packing forms a part of the sleeve, as shown. Annealed gauge glasses protected inside and out by this or similar means will last as long under any circumstances as it is possible to make them by any means thus far made known.

If packing 'rings' are used, then in order to secure a tight joint with the least possible pressure, thus avoiding any tendency to break or to cramp the glass due to the packing, it is necessary to obtain a form of packing that will fit snugly in the space in the nut.

(*H. B. Waverly*, '*Mechanical World.*')

The following arrangement, fig. 14, for packing is said to give very satisfactory results:— A fibre washer $\frac{1}{16}$ in. thick with an outside diameter to allow an easy fit into the gauge-cock nut,

and an inside diameter equal to that of the glass, is put into each nut before the rubber gasket. The fibre washer so fitted prevents the rubber gasket being pushed out between the glass and

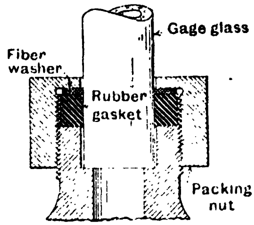

FIG. 14.

hole in the nut. The fibre does not scratch the glass, and therefore will not cause the latter to crack and break, as would a metal washer or a nut with too snug a hole in it.

(*F. R. Rawson*, *Power*.)

Superheated water acts on most varieties of glass as a violently corrosive agent, and for gauge glasses ordinary glass cannot be used. A special glass, under the brand 'Hercules,' to resist high-pressure steam, is made by Joseph Tomey & Sons, Ltd., Birmingham. It is specially constructed of boro-silicate, and is made after the 'Jena' formula. The glass is practically white in colour and has withstood the following remarkable tests:—

> After the glass had been submerged in iced water for five minutes it was plunged into molten lead, and showed no signs of fracture. It has withstood 4000 lbs. hydraulic pressure, and 750 lbs. steam pressure.

Although the glass was specially manufactured to meet the requirements of high-pressure and superheated boilers, it has been used for a great number of low-pressure boilers, because at low pressure the material is practically unbreakable and is considerably cheaper in the long run than the ordinary quality gauge glass.

Cutting Gauge Glasses.

A simple and reliable method of cutting gauge glasses to any desired length consists in filing a notch, then wrapping the tube with a strip of wet filter paper or paper towel on each side of the notch, leaving the space between the strips $\frac{1}{16}$ to $\frac{1}{8}$ in. wide. The strips of paper may be $\frac{1}{4}$ to $\frac{3}{8}$ in. wide and wrapped on about three layers deep. The fine flame of a blast burner is directed against the tube between the papers and at the notch for a second or two when the tube is slowly rotated in the flame. The tube should be turned rapidly enough to avoid burning the wet paper. At the completion of the revolution the tube will break off with a clean and even cut. Short tubes for oil-sight feeds may be cut with ease by this method when it would be impossible to break them off from just a file notch. Bottles and large tubes are equally easy to cut.

In breaking a tube from a file notch, which is the universal method for small-diameter tube and rod, much depends on the nature of the file mark. Just as in cutting window glass, what is wanted is a cut and not a scratch, and in filing a notch completely around the tube, even the inexperienced is likely to make a real cut at some point on the circumference, while in his hands the single file notch may not register the cut necessary for the clean break.

REDUCING VALVES.

These are very needful fittings in connection with steam appliances when steam has to be reduced from a high to a uniform low pressure. It is a common mistake to order reducing valves too large, which leads to their excessive wear. No reducing valve should be implicitly trusted; but wherever a rise in steam-reduced pressure is liable to bring about serious results a relief valve should follow the reducing valve.

Reducing valves which are too large for the amount of steam they are to pass are usually unsatisfactory; either they will open and close continually and chatter violently, or will stand in a position nearly closed and become steam-cut and leaky. It is more satisfactory to use two small valves in parallel, one set slightly heavier than the other, so that one will supply steam up to its maximum capacity before the other opens. Another way is to use one reducing valve equal to about one-half the maximum demand, and a by-pass of about the same capacity with a hand-operated valve in it, so that when the reducing valve reaches its capacity the by-pass valve may be opened wide and the reducing valve be closed down and make up the shortage. This, of course, requires that an attendant shall observe the pressure gauge occasionally.

STEAM PIPES.

Steam pipes should be made of mild steel, preferably weldless. The steel should be of boiler plate quality. For pressures up to 250 lb. per sq. in. the following are suitable minimum thicknesses:

Bore of pipe, inches .	2	$2\frac{1}{2}$	3	4	5	6	7	8	9	10	12
Thickness, inches .	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{7}{16}$

Bends for pipes 6 ins. and larger should be $\frac{1}{16}$ in. thicker. Flanges for pipes up to 6 ins. may be screwed on, but for larger sizes, flanges should be welded or riveted on.

Flanges should be of wrought steel forged from the solid. The ends of the pipes should be expanded into screwed flanges.

It is essential in steam pipe arrangements to make adequate provision for (*a*) drainage, and (*b*) expansion.

The pipes if possible should not rise from the valve on the boiler to the main steam pipe, as this causes a water pocket.

If the junction valve has to be below the level of the main steam range, suitable drain connections must be fitted. These drains should be tested by blowing to atmosphere at least twice a day.

The best steam pipe arrangement between engine and boiler is that the boiler stop valve should stand at the top of a vertical pipe on the boiler and there should be a steady slope in the direction of flow of about $\frac{3}{8}$ in. per 100 ft., and a drop-leg or separator should be provided at the end of the range.

Rupture from water hammer occurs from failure of pipe drains, or from neglect in their use. If possible, pipe lines should be so designed as to render drains unnecessary except at drop-legs or separators.

The closing of no valve should be able to cause water to accumulate against it. Where it is impossible to avoid drainage points, these should be carefully arranged, and unduly small drain pipes must be avoided lest they choke with rust or dirt.

Long lengths of drain piping, from steam mains to traps, should be avoided; they are a continual source of waste, and with superheated steam the water they drain away has largely been formed by condensation in the drain piping itself. Therefore the pipes from steam main to traps should be as efficiently lagged as the main steam line.

Idle parts of the steam range should be isolated when not in use, and carefully drained, as valves often leak when shut, through scoring of faces and grit wedging in, preventing tight closing.

All live steam drains should be returned to the feed tanks.

The coefficient of expansion of steel being 0·000006 per °F. (at moderate temperatures), the expansion at a pressure of 190 lb. per sq. in. or a temperature of 384° F., from an initial temperature of 60° F. is about 2·33 in. per 100 ft.

At higher temperatures the coefficient of expansion increases: at 800° F. it is about 20 per cent. greater than at 200° F. This is taken into account in calculating the expansion of pipe lines. Pipes should be arranged so that about 2·8 inches for saturated, or about 4·2 in. for superheated steam per 100 ft. of length can be allowed for without straining the pipes or their connections. Flexibility should be obtained by the use of steel bends or spring legs, rather than by expansion joints.

In erecting pipe lines, the pipes are frequently cold sprung in a direction opposite to that of expansion to about half the expected expansion so as to minimise the expansion stresses, the bends being then equally stressed in opposite directions when hot and when cold.

The pipes should be marked when cold opposite marks on the walls, and again when hot. Anchors should be at the points where least movement is shown. Pipes are best carried on brackets. Where they rest they should have rubbing pieces firmly attached, and they should not rest on rollers as the friction helps to check vibration.

The centre of a main steam pipe should be anchored, to divide the longitudinal expansion. Where engine and boiler room are in line and steam mains are of considerable length, a ∩ or Ω-shaped bend should be inserted between the engine and boiler-room. Anchor-plates should then be fixed in the centre of the engine-room main and in centre of boiler-room main.

Expansion bends are best arranged to project horizontally from the line of pipe, in order to allow a free passage for water. If they stand vertically upwards they prevent water flowing, and if hung downwards they form water pockets. In both cases they necessitate careful drainage.

When erecting new steam pipes, attention should be given to the thorough cleaning of the pipes, as otherwise fine particles of grit, driven by the steam into the faces of the valves, cut into the seats, causing considerable trouble and expense.

New pipes and drains should be well blown through to remove scale and dirt before traps are fixed.

Pipes may be best bent by filling with thoroughly dry sand and plugging the ends. They will then bend without buckling.

Copper pipes are bent by being filled with melted resin, or with solder, the pipe bent cold, and the filling then melted out.

The radius of bends in steel steam pipes measured to the centre line of the pipe should not be less than three and a half or preferably four times the diameter of the pipe.

While main steam pipes should have flanged joints, small pipes, if the pressures are moderate, say up to 80 lb. per sq. in., may be screwed into fittings or branch pieces. The screw threads should be tapered.

For dimensions of pipe flanges, and bolting, see pages 108–110, Section V (I), Vol. I.

Steam Pipes for High Pressure and Temperature.

An excessively thick pipe is not necessarily a strong pipe; it is very stiff, and therefore subject to heavy expansion stresses.

For 0·17 carbon steel the following are suitable maximum working stresses: at 900° F., 4,500 lb. per sq. in.; at 850°, 5,500 lb.; at 800°, 6,500 lb.; at 750°, 7,500 lb.; at 700°, 8,400 lb. These are below the stresses that will produce a creep rate of one-hundredth of one millionth inch per inch per hour, or one tenth of 1 per cent. during a working life of 100,000 hours.

The maker of weldless, hot-finish tubes requires a tolerance of 12½ per cent. Pipes must be specified 12½ per cent. thicker than the minimum required, and may come out 12½ per cent. thicker than that specification—i.e., a pipe required to be not less than ⅞ in. must be ordered 1 in., and accepted if 1⅛ in. This increase does not affect the strength, for as stress due to expansion goes up, that due to internal pressure comes down, but it does increase the thrust on the fixed points to which the pipes connect.

Under modern conditions the only satisfactory method of joining pipes is by welding. The best joint has the pipes butt welded, and a heavy sleeve passed over the joint and welded to both pipes, but it is not easy to erect and weld on site and can only be broken with difficulty. The usual method is to combine flanges and welding. Collars are formed on the ends of the pipes, integral with the pipes, and are backed by heavy loose flanges, which are bolted together and take all mechanical and expansion stresses. The joint is made steam tight by a weld round the junction of the collars. This type of joint can be made to connect to valves and fittings and can be easily broken. The bolts are of nickel-chrome-molybdenum steel, of ultimate stress (cold) 69 tons, screwed with five threads all the way, and with nuts at each end.

The tolerance up to and including 12 in. nominal bore is plus/minus 12½ per cent. as stated, but over 12 in. bore the tolerance is plus/minus 15 per cent.

Creased and Corrugated Iron Bends. (*Aiton.*)

With thick tubes, the inner wall does not appreciably thicken in bending, almost the entire difference in length of the inner and outer walls of the bend has to be accommodated by thinning of the outer wall. To avoid excessive thinning, the radius of a bend must be large, but this reduces the flexibility of the pipe. This difficulty can be overcome by forming creases on the inner wall which vanish at about two-thirds of the circumference. Bends can be made in a thick creased pipe to a smaller radius than in a plain pipe, giving at the same time less thinning of the outer wall and more flexibility. Another method of achieving the same object is to corrugate the pipes.

The bends have a series of corrugations which in themselves strengthen the tube, especially as the process of manufacture thickens the wall of the tube. The corrugated pipe is formed into a bend in the usual way, and there can be no thinning of the wall in bending as all that happens is that the corrugations are somewhat opened out on the outside of the bend and are closed up on the inside.

These corrugated pipes can be bent to a much smaller radius than plain pipe—in extreme cases they can be bent to the same radius as an elbow. In all cases, to stand corrugating, the tubes must be of a good soft quality of steel and must be well made, free from variations of thickness; they are usually solid drawn tubes, hot finish, 24 tons to 28 tons tensile, 25 per cent. elongation in 8 ins.

The comparative flexibility of different types of pipe bend is shown by the following examples of bends, 10-in. internal diameter, ⅞-in. thick, at 700 lb. per sq. in. pressure, 900° F., the bends measuring 45 ins. from centre line of one leg to flange of other leg.

Description of Pipe.	Radius of Curved Portion.	Thrust in Lb.	Stress Lb. per Sq. In.
Plain	30 in.	14,980	13,640
Creased at bend . . .	10 in.	6,070	7,250
Corrugated at bend . .	10 in.	4,400	6,140
Creased at bend and corrugated on straights for 10 in. . . .	10 in.	3,480	3,420
Corrugated at bend and on straights .	10 in.	2,345	2,505

An expansion bend of the lyre type made from plain tube 10 ins. bore and 138 ins. high from the centre of the main to the centre of the pipe at the crown of the bend, allows for a movement of 2¼ ins. out or in; a bend made from corrugated pipe of the same bore can be made 55 ins. high, and gives 2¼ ins. out or in, and a similar bend 72 ins. high gives 3⅜ ins. out or in, or 50 per cent. more than the much larger plain bend.

The severest test of the behaviour of the bends under heavy vibration has been on steam hammers. A 5-ton hammer when first started to work had connected to it a plain 5-in. steel bend and subsequently a copper bend, but this did not last more than a month, and the steam joints had to be remade, on an average, once a day. A 5-in. corrugated bend 24 ins. × 20 ins. made from stock solid-drawn hot-finish tube, $\frac{7}{32}$ in. thick, was fitted, and after working for two years was microscopically examined and showed no signs of deterioration.

Roughly speaking, the pitch of the corrugations is 2¼ ins. and each curve of the corrugation 1¼ ins. The steam friction is about double that of a plain bend.

There are no records of a corrugated pipe ever having failed in service.

See 'Strength and Flexibility of Corrugated Piping' by R. L. Dennison, John Hopkins University, *The Engineer*, 20th Sept. 1935, et seq.

Lloyd's Rules for Steam Pipes.

The following is abstracted and abridged from Lloyd's Rules:

Pipes made from electro-deposition of copper on a mandril are not to be used for steam.

Copper pipes are to be properly annealed.

Copper pipes for pressures over 75 lbs. per square inch are to be solid drawn.

Pipes for pressures above 180 lbs. per square inch are not to be of copper if the diameter exceeds 5 ins.

Copper pipes are not to be used for superheated steam.

Permissible working pressures (W.P.) are as follows:

where,

t = thickness in hundredths of an inch; d = diameter of pipe in inches.

For brazed copper	$\text{W.P.} = \frac{45(t-3)}{d}$
For solid-drawn copper	$\text{W.P.} = \frac{60(t-3)}{d}$
For solid-drawn cold-finished steel . .	$\text{W.P.} = \frac{120(t-10)}{d}$
For solid-drawn hot-finished steel . . .	$\text{W.P.} = \frac{120(t-12)}{d}$
For lap-welded iron or steel with or without covering straps	$\text{W.P.} = \frac{90(t-12)}{d}$

Pipe Hangers.

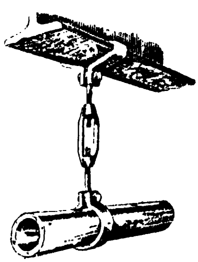

Fig. 15.

DIMENSIONS AND SAFE LOADS FOR PIPE-HANGER PARTS.

Pipe, Size. Ins.	Pipe, Thick. Ins.	Band, Width. Ins.	Clamp, Thick. Ins.	Clamp. Width. Ins.	Rod, Diam. Ins.	Bolts, Band. Ins.	Bolts, Clamp. Ins.	Safe Load. Lbs.	Length ½ Band. Ins.
2	1/4	1 1/4	5/16	1 1/4	1/2	1/2 × 2	1/2 × 2 1/4	1,160	6
2 1/2	1/4	1 1/4	5/16	1 1/4	1/2	1/2 × 2	1/2 × 2 1/4	1,160	7
3	1/4	1 1/4	5/16	1 1/2	1/2	1/2 × 2	1/2 × 2 1/4	1,160	8
3 1/2	1/4	1 1/2	3/8	1 1/2	1/2	1/2 × 2	1/2 × 2 1/4	1,160	9
4	1/4	1 1/2	3/8	1 1/2	5/8	1/2 × 2	5/8 × 2 1/2	1,870	10 1/2
4 1/2	1/4	1 1/2	3/8	1 1/2	5/8	1/2 × 2	5/8 × 2 1/2	1,870	11
5	1/4	2	3/8	2	5/8	1/2 × 2	5/8 × 2 1/2	1,870	13
6	1/4	2	1/2	2	5/8	5/8 × 2 1/4	5/8 × 2 1/2	1,870	14
7	3/8	2	1/2	2	3/4	5/8 × 2 1/4	3/4 × 3	2,830	16
8	3/8	2		2 1/4	3/4	3/4 × 2 3/4	7/8 × 3 1/4	2,830	18
9	3/8	2	1/2	2 1/2	7/8	3/4 × 2 3/4	7/8 × 3 1/4	3,940	20
10	1/2	2	1/2	2 1/2	7/8	3/4 × 3	7/8 × 3 1/4	3,940	22
12	1/2	2 1/4	1/2	2 1/2	1	7/8 × 3 1/4	1 × 3 1/2	5,180	25
14	1/2	2 1/4	1/2	2 1/2	1 1/8	1 × 3 1/2	1 1/8 × 3 3/4	6,510	27
15	1/2	2 1/2	1/2	2 1/2	1 1/8	1 × 3 1/2	1 1/8 × 3 3/4	6,510	30
16	1/2	3	5/8	3	1 1/8	1 1/8 × 3 3/4	1 1/4 × 4	6,510	33
18	5/8	3	5/8	3	1 1/4	1 1/8 × 4	1 1/4 × 4	8,410	35
20	5/8	3	3/4	3	1 1/4	1 1/4 × 4	1 1/4 × 4 1/2	8,410	37
22	3/4	3	3/4	3	1 1/4	1 1/4 × 4 1/2	1 1/4 × 4 1/2	8,410	39
24	3/4	3	3/4	3	1 1/4	1 1/4 × 4 1/2	1 1/4 × 4 1/2	8,410	42

(*Power.*)

Exhaust Pipes.

These may be of cast iron and should be free from blow holes. The flanges should be faced and jointed with soft rubber rings. Exhaust pipes should be painted while under vacuum, preferably with several coats of red lead paint well brushed in until all the pores are filled.

Joints for Steam Pipes.

Flanged steam pipes for low or moderate pressures are jointed with jointing pastes or special insertions, such as asbestos millboard.

For high pressures, corrugated metal joint rings (of brass, for example), inside the bolt circle, are used. They should be coated with jointing paste, of which there are special brands on the market, or with pipe cement, graphite and boiled oil. The joint should be bolted up before the paste has set.

To prevent joints adhering to flanges, they should be painted with black lead, mixed to a thin paste with water. Joints so treated can be used over again. Covering the bolt threads with graphite and tallow facilitates the removal of the nuts when joints are to be broken.

For very high pressures, a successful joint-ring is one of soft steel, fitting inside the bolt circle and thickened for a radial width of about ⅝ in., the inner diameter of the thickened ring being equal to the pipe bore. This thickened ring has narrow circular grooves with sharp edges which bear on the flanges or the ends of the pipes when the joint is bolted up.

Size of Steam Pipes.

The diameter of a steam pipe is usually in practice chosen to give a certain steam velocity in the pipe, 100 feet per sec. being a customary velocity for steam at ordinary pressures.

Many engineers use this speed for both saturated and superheated steam. Others vary the velocity as in the following table:

Diameter of Pipe.	Velocity, Feet per sec. Saturated.	Superheated.
Up to 3 ins.	75	100
3½ ins. to 9 ins.	90	120
10 ins. and over	90	140

For superheated steam in long, straight pipe lines 180 ft. per sec. may be used.

It should be remembered that the higher the velocity the smaller is the pipe diameter, and hence the external radiating surface is less.

Pressure Drop in Pipe Lines.

The formula (known as the Babcock formula) usually employed to relate the flow, pressure drop, and pipe diameter is

$$W = 87{\cdot}5\sqrt{\frac{D\,(p_1 - p_2)\,d^5}{L\left(1 + \frac{3{\cdot}6}{d}\right)}},$$

where W = weight in pounds per minute; d = diameter in inches; D = density or weight per cubic foot; p_1 = initial pressure; p_2 = pressure at end of pipe; L = length in feet.

This formula corresponds to $p_1 - p_2 = 0{\cdot}0001306\,\dfrac{W^2L\left(1 + \frac{3{\cdot}6}{d}\right)}{Dd^5}$ and also to $h = f\,\dfrac{v^2}{2g}\cdot\dfrac{4L}{d_1}$

where h = loss of head in feet, d_1 = diameter in feet, f = coefficient of friction of the steam in the pipe, f being equal to $0{\cdot}0027\left(1 + \frac{3}{10d_1}\right)$; this coefficient is due to R. C. Carpenter, and is accepted by Unwin.

The following table gives, approximately, the weight of steam per minute which will flow from various initial pressures, with one pound loss of pressure, through straight, smooth pipes, each having a length of 240 times its own diameter. The table is calculated from the above formula.

TABLE OF FLOW OF STEAM THROUGH PIPES.

Initial Pressure by Gauge. Lbs. per Sq. In.	Internal Diameter of Pipe in Inches. Length = 240 diameters.													
	½	1	1½	2	2½	3	4	5	6	8	10	12	15	18
	Weight of Steam per Minute in Pounds, with One Pound Loss of Pressure.													
1	0·92	1·79	4·80	9 4	15·6	23·8	45·5	75·0	112	209	337	497	799	1160
10	1·14	2·20	5·90	11·5	19·2	29·4	56·0	92·2	137	258	415	611	982	1430
20	1·33	2·58	6·91	13·5	22·7	34·4	65·6	108	161	302	486	716	1150	1680
30	1·49	2·89	7·75	15·2	25·3	39·5	73·5	122	180	338	545	808	1290	1880
40	1·64	3·18	8·54	16·7	27·8	42·5	81·0	133	198	372	600	885	1420	2070
50	1·77	3·44	9·23	18·1	30·1	46·0	87·6	144	215	403	650	956	1540	2240
60	1·89	3·67	9·85	19·3	32·1	49·0	93·5	154	229	429	694	1020	1640	2390
70	2·01	3·91	10·45	20·5	34·1	52·1	99·5	164	244	457	736	1080	1740	2540
80	2·12	4·11	11·0	21·6	35·9	54·9	105	172	266	480	775	1140	1830	2670
90	2·22	4·31	11·6	22·6	37·7	57·6	110	181	269	504	813	1200	1930	2810
100	2·32	4·50	12·1	23·6	39·3	60·0	114	188	281	526	850	1250	2010	2920
120	2·50	4·85	13·0	25·4	42·4	64·6	123	203	302	567	914	1340	2160	3150
150	2·74	5·33	14·3	28·0	46·5	71·0	136	223	332	623	1004	1480	2380	3480
200	3·11	6·04	16·2	31·7	52·9	80·5	154	253	377	705	1140	1670	2700	3940

(*Morley.*)

The foregoing table is based on saturated steam. For any other loss of pressure multiply by the square root of the given loss. For any other length of pipe *divide* 240 *by the given length expressed in diameters, and multiply the figures in the table by the square root of this quotient*, which will give the flow for 1 lb. loss of pressure. Conversely, dividing the given length in diameters by 240 will give the loss of pressure for the flow given in the table.

For superheated steam multiply the values in the table by $\sqrt{\dfrac{\text{density of superheated steam}}{\text{density of saturated steam}}}$

—or *approximately* $\dfrac{1}{\sqrt{1 + {\cdot}0015\,t_s}}$, where t_s = degrees of superheat.

The loss of head due to getting up the velocity, to the friction of the steam entering the pipe, and passing elbows and valves, will reduce the flow given in the table. The resistance at the opening, and that at a globe valve, are each about the same as that for a length of pipe

equal to 114 diameters divided by a number represented by 1 + (3·6 ÷ diameter). For the sizes of pipes given in the table these corresponding lengths are, approximately:—

·¾	1	1½	2	2½	3	4	5	6	8	10	12	15	18
20	25	34	41	47	52	60	66	71	79	84	88	92	95

The resistance at an elbow is equal to two-thirds that of a globe valve. These equivalents—for opening, for elbows, and for valves—must be added in each instance to the actual length of pipe. Thus a 4-in. pipe, 120 diameters (40 ft.) long, with a globe valve and three elbows, would be equivalent to 120 + 60 + 60 + (3 × 40) = 360 diameters long; and 360 ÷ 240 = 1½. It would therefore have 1½ lbs. loss of pressure at the flow given in the table, or deliver (1 ÷ $\sqrt{1\cdot 5}$ = ·816), 81·6 per cent. of the steam with the same (1 lb.) loss of pressure.

In place of the calculations on the table given in the preceding paragraphs, the following nomographic chart (based on the formula, p. 64) may be used, and provides a very convenient and sufficiently accurate method of determining pipe sizes, pressure drops, etc.

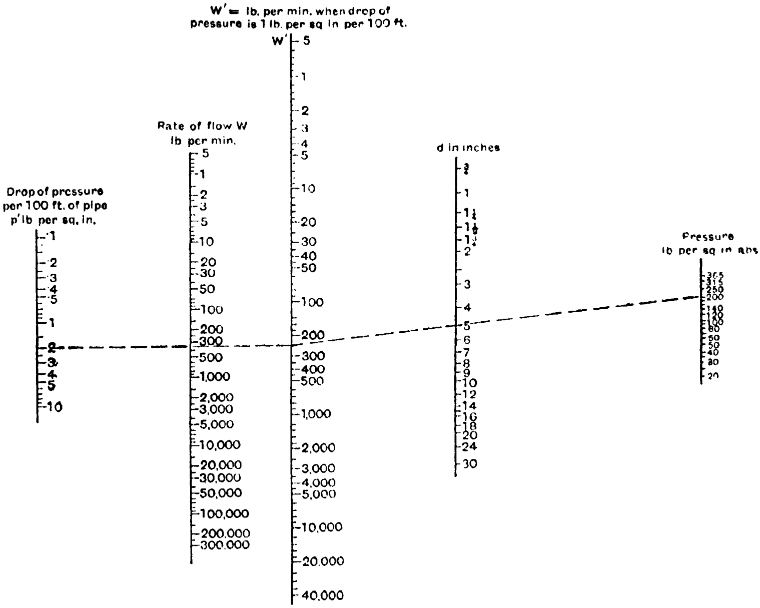

FIG. 16.

Method of use of chart:

(a) To find diameter for given quantity and pressure drop, join W and p' and produce to cut scale of W', join point so found to pressure scale at right, read required diameter at intersection with scale of d.

(b) To find quantity for given pipe and pressure drop, reverse above process, finding first W' and then W.

(c) To find pressure drop for given pipe and flow, join points on scales of pressure and d, produce to find W', join W' to W and produce to cut scale of p' at required value.

Flow of Air and Steam in Pipes.

The conclusions of subsequent investigations made by McAdams and Sherwood are that for air the best formula is:

$$P_1 - P_2 = \frac{0{\cdot}13 f L Q^2 P^2}{P_a D^5 T}$$

$$\text{or } Q_1 = 2{\cdot}77 \sqrt{\frac{D^5 (P_1 - P_2)\, T\, P_a}{f L P_1^2}}$$

and for steam

$$P_1 - P_2 = 0{\cdot}0484 \frac{f L G^2}{D^5 w_a}$$

$$\text{or } G = 4{\cdot}54 \sqrt{\frac{D^5 w_a (P_1 - P_2)}{f L}}$$

where

- P = absolute pressure in lbs. per sq. in.;
- P_1, P_2 = initial and final pressures in lbs. per sq. in.;
- P_a = average pressure, lbs. per sq. in. abs.;
- Q_1, Q = cub. ft. air per minute at P_1, P, respectively;
- D = inside diameter of pipe in inches;
- L = equivalent frictional length of pipe in feet = length of straight pipe + $2{\cdot}5\ D \times$ number of elbows;
- T = air temperature, ° Fah., abs.;
- G = lbs. steam per minute;
- w_a = lbs. per cub. ft. of fluid flowing at average temperature and pressure;
- f = friction coefficient,

in accordance with the values in the following table:

Values of $\frac{G}{D}$ for Steam, or $\frac{QP}{DT}$ for Air.	Steam.	Air.			
		Iron and Steel Pipes.		Copper and Brass Pipes.	
		300°	70°	300°	70°
0·10	—	0·0113	0·0104	0·0094	0·0088
0·20	—	0·0098	0·0090	0·0080	0·0074
0·40	0·0097	0·0085	0·0079	0·0067	0·0062
0·60	0·0086	0·0078	0·0072	0·0061	0·0056
0·80	0·0081	0·0074	0·0068	0·0056	0·0052
1·00	0·0077	0·0070	0·0065	0·0053	0·0049
1·50	0·0070	0·0065	0·0060	0·0048	0·0045
2·00	0·0066	0·0062	0·0059	0·0045	0·0043
4·00	0·0060	0·0059	0·0058	0·0041	0·0038
10·00	0·0059	·0058	0·0058	—	—
20	0·0056	0·0058	0·0058	—	—
30	0·0056	0·0058	0·0058	—	—
40	0·0055	—	—	—	—

Loss of Heat from Uncovered Steam Pipes.

The large amount of heat lost from bare steam pipes, and the consequent waste of coal, is frequently not fully appreciated. One square foot of uncovered pipe surface, the pipe containing steam at 100 lbs. per sq. in., will dissipate about 3 B.Th.U. per hour per degree of temperature difference between the steam and the atmosphere, or about 850 B.Th.U. per hour at ordinary

atmospheric temperatures. The steam condensed per hour due to this square foot of bare pipe will be 0·96 lb. In a year of 3,000 working hours the steam wasted will amount to 2,880 lbs., corresponding to a coal consumption of about 410 lbs.

A single 9-ft. length of 6-in. pipe, if uncovered, will waste by heat loss about 2·5 tons of coal per year.

80 to 90 per cent. of this loss may be avoided by the use of suitable coverings, the expense of which is amply counterbalanced by the saving in coal.

It is important that flanges, valves, etc., as well as the body of the pipe lengths, should be covered.

The loss of heat per hour per sq. ft. of surface, in the case of bare pipes, depends principally upon the difference of temperature between the pipe—*i.e.* practically, that of the steam in the pipe—and the surrounding atmosphere, but also is influenced to some extent by the diameter of the pipe, its position—whether horizontal or vertical—the presence of air currents about the pipe, the nature of the pipe surface, and the velocity of the steam in the pipe.

Complete data on all these factors are lacking.

It is usually assumed that the loss is proportional to the difference of temperature, but careful experiments show that the rate of loss, B.Th.U. per sq. ft. per degree F., increases at the higher temperature differences. Thus from 1·95 at 50° difference, it becomes 2·4 at 150°, 3·26 at 300°, 4 at 400°, and 5·26 at 460°. (*McMillan.*)

There is little experimental data on the effect of pipe diameter, but it appears that the loss per sq. ft. decreases as the diameter increases, and this is in accordance with theory.

The following table shows the heat loss from pipes covered with various thicknesses of 85 per cent. magnesia composition and the approximate loss from uncovered pipes is given for comparison.

LOSS OF HEAT FROM STEAM PIPES, WITH STEAM 300° F. ABOVE ATMOSPHERIC TEMPERATURE.
LOSS IN B.TH.U. PER HOUR PER SQUARE FOOT OF PIPE SURFACE.

	Pipe Diameter.					
	1 in.	2 ins.	4 ins.	6 ins.	8 ins.	12 ins.
Bare pipe	1100	1070	1010	945	920	880
Magnesia covering 0·5 in. thick . .	340	280	250	240	235	230
,, 1·0 in. ,, . .	260	200	170	155	148	140
,, 1·5 ins. ,, . .	220	165	145	120	115	105
,, 2·0 ,, ,, . .	195	140	110	100	92	87
,, 3·0 ,, ,, . .	170	120	89	75	71	66
,, 4·0 ,, ,, . .	—	110	75	65	59	52
,, 5·0 ,, ,, . .	—	—	66	58	52	46

(*McMillan.*)

The 'efficiency' of the covering in preventing loss of heat is taken as the ratio (heat loss from bare pipe − loss from covered pipe) ÷ heat loss from bare pipe. The table shows that this 'efficiency' varies with the thickness and the pipe diameter, hence to compare different covering materials, the comparison must be made under the same conditions of thickness, etc., if the comparison is to show the relative insulating properties.

The coverings may also be compared on the basis of cost per foot run—*i.e.* taking a greater thickness in the case of the cheaper materials.

The relative properties of various pipe coverings are shown in the following table, the thickness being about 1·4 ins., diameter of test pipe 1 in., steam pressures in pipe 50, 100, and 200 lbs per sq. in.

RELATIVE VALUE OF PIPE COVERINGS.

Material.	Weight per foot run, lbs.	Insulating Efficiency per cent. At 50 lbs. per sq. in.	At 100 lbs. per sq. in.	At 200 lbs. per sq. in.
Sectional cork	1·0	85	86	87
Slag wool	2·6	83	84	—
Cellular asbestos	2·4	81	82	83
Silk waste	2·3	85	85·5	86
Hair felt and asbestos	2·7	83	84	85
Sectional magnesia	1·6	79	79	—
Plastic magnesia	2·2	77	77	—
Diatomite (plastic)	2·2	79	79	—
Kisol and magnesia (plastic)	2·9	77	78	79
Kisol and hair felt (plastic)	3·0	75	77	78
Slag wool and composition (plastic)	1·5	75	76	—
Mica (plastic)	2·6	73	75	—
Asbestos composition (plastic)	3·9	69	71	72
Thinner Coverings.				
Hair felt 0·5 in. thick	—	79	—	—
Magnesia-filled rope 0·7 in. thick	0·8	61	64	—
Asbestos rope 0·7 in. thick	1·2	47	53	—

(*Bradley.*)

BOXES FOR PIPE FLANGES.

In estimating the loss of heat from covered pipe lines, the effect of the boxes covering the flanges in adding to the total area of radiating surface must be allowed for.

For typical flange boxes, assuming one box at each 10 feet of pipe line, the additional area may be taken as about 7 per cent. *i.e.* to find the heat radiated, calculate the pipe surface in square feet, as if there were no flanges, add 7 per cent., and multiply by the appropriate figure from the table on page 67.

NOTES ON COVERINGS FOR PIPES AND BOILERS.

As well as having good heat insulating properties, it is important that pipe coverings be durable. A material which easily disintegrates under vibration will not long remain an efficient insulator, nor will a material which cannot stand high temperature. For example, hair felt is a good insulator, but will not stand contact with pipes carrying high pressure or superheated steam.

Tests with locomotive-type boilers, first uncovered and then covered with 2 ins. of 85 per cent. magnesia, showed in the second case 2·5 per cent. additional evaporation for 13·8 per cent. less coal, or a total coal saving of 16 per cent.

Recent tests at the National Physical Laboratory have shown that 85 per cent. magnesia (that is, 85 per cent. hydrated magnesium carbonate and 15 per cent. asbestos) begins to calcine at 450° F., and that at 600° F. the decomposition is considerable. At the high steam temperatures now coming into use in large power stations (pressures up to 375 lbs. per sq. in. and temperatures up to 700° F.) it is necessary to place next the pipe a material unaffected by heat. About ½ in. of slag wool or of special composition may be used, then a layer of 85 per cent. magnesia or other good non-conductor, and then an outer layer, ½ in. thick, of hard-setting composition to bind the whole together.

Aluminium Foil for Heat Insulation.

Thin aluminium foil crumpled so as to form irregular ridges and include air spaces has been successfully applied as a heat-insulating medium for pipes or for flat surfaces.

In spite of the high conductivity of aluminium the flow of heat through the crumpled foil is low owing to the smallness of the contact areas. The bright surface also reduces radiation loss. The foil is crumpled and wrapped round the pipe or other object to be insulated in layers until a

sufficient thickness is obtained. The covering is held in position by spacing wedges and a thin sheet steel covering attached by spot welding.

It is claimed that the heat loss through the foil is considerably less than that through an equal thickness of 85 per cent. magnesia.

Notes on Lagging.

The holding of slag wool to pipes or boilers (to which asbestos composition has been applied) is best effected by means of wire-bound split laths, so spaced that the finishing overlay of cement may not only be keyed in between them but may also afford a means of locating any leakages, in addition to preserving a favourable density estimated at from 10 to 12 lbs. per cubic foot.

A cheaper but decidedly less effective way of holding slag wool together is by means of wire netting in the meshes of which an overlay of plastic cement is impacted. When, therefore, disintegration takes place and the wool settles down by its own gravity, the upper surface is reduced to a mere shell of wire and cement by which the bulk is held in suspension.

Glass Silk, a fairly recent introduction, has been ascertained by the N.P.L. to have very high insulating properties.

Hair-felt is one of the finest non-conductors known, but is quickly consumed by direct contact with heated surfaces.

STEAM SEPARATORS.

The function of a steam separator, or as it would be more correctly called a water-separator is to extract the water particles carried in suspension by wet steam. Apparatus of this kind served a very useful purpose before the use of superheated steam became general, but are now much less commonly employed. They act by causing a change of direction in the flow of the steam, either giving it a definite whirling path, when the water particles are thrown out by centrifugal force, or by causing the steam to pass round baffles to which the water adheres. To be at all efficient, a steam separator must have ample volume, so that the steam passes through it comparatively slowly and the water has time to settle out and is not picked up again by the rush of the steam past the edge of any baffles. Separators should be well lagged, and the larger they are the better.

STEAM TRAPS.

Steam traps are indispensable adjuncts to apparatus from which water of condensation has to be drained off with regularity and certainty, and without loss of steam.

Steam traps fall into two main classes; (*a*) Float or Bucket traps which are operated by the difference in density between water and steam, and (*b*) Expansion traps which depend on a difference of temperature. The varieties of each kind are almost innumerable, and great care should be taken in selecting the design which will be best suited to the duty to be performed.

In the simple float trap, a hollow metallic float on the end of a pivotted lever rises as the water accumulates in the trap, and opens a discharge valve until the water has fallen to its normal level. Such traps can be designed so that the whole of the working parts can be inspected or removed without interference with the pipe connections. The conditions that the float must be strong enough to resist collapse under the steam pressure, and yet light enough to have the necessary buoyancy and power to open the discharge valve, impose a limit on the valve size and discharging capacity of traps of this type. The discharge of such traps should theoretically be continuous. In practice it is generally intermittent at small loads, which is an advantage as it is less severe on the valve and seating.

When the amount of water to be handled varies greatly, or when the trap may have to work in a tilted position as on shipboard, the float trap is provided with a trip mechanism which ensures that it either discharges at full flow or not at all. This intermittent action enables the trap to be heard working, which is a good feature when it is not a nuisance. The trip mechanism detracts, somewhat of course, from the simplicity of the trap.

In traps of the ordinary bucket type, the discharge valve is kept shut by the upward pressure due to the buoyancy of a hinged open bucket around which the water gradually rises. In time the water overflows into the bucket, causing the latter to sink and thereby to open the valve which is situated at the end of a fixed pipe projecting downwards nearly to the bottom of the bucket. The latter is therefore emptied by the steam pressure and its buoyancy restored. The discharge is necessarily intermittent. This kind of trap is long-lived and reliable. It has no float to collapse or to leak, and can be designed for high pressures.

The inverted bucket trap is equally simple in construction. In this trap the bucket, which is upside down, is usually suspended from a hinge at the side. It is completely immersed, and the condensate enters it through an upturned pipe projecting into its interior. Air can escape through

a small hole in the top of the bucket, so that the trap can never become air-locked. This gives it an advantage over the open bucket type. Should the hole, however, get stopped up, the trap will be put out of action. Furthermore, if the trap gets hot enough to evaporate the water it contains, it will blow steam full bore and will not be able to re-set itself. Traps of this kind should always be left bare on top, and it may be desirable to connect them to the apparatus they serve by means of a length of unlagged pipe when the steam is very hot. Another precaution often advisable is a non-return valve to prevent loss of water from the trap.

Inverted bucket traps can be made in very small sizes. The bucket of these small traps is not hinged, but operates the valve directly by its rise and fall.

Expansion traps rely for their action on the fact that water of condensation can be, and practically always is, colder than the steam from which it was derived. The discharge valves of such traps are controlled by the expansion and contraction of some substance according to whether it is in contact with steam or water. This principle ensures that the trap shall always be full open when cold, so that it is not liable to be injured or put out of action by frost, and it will discharge all air from the system at starting up.

In the simplest form of expansion trap, a straight copper tube fixed at one end, expands and keeps the discharge valve closed so long as it is subjected to steam temperature. If water is present the tube contracts and opens the valve. Traps of this kind are somewhat lengthy, though this is often no drawback. The length of the trap can be much reduced by employing the expansion of a liquid such as oil sealed in a corrugated bellows piece to operate the valve. In both metallic and liquid expansion traps it is necessary to limit the closing pressure by some kind of spring device, otherwise the valve or its seating might be injured when the trap was exposed to steam at a higher temperature than normal.

As an alternative to oil, a volatile liquid may be used in the bellows. This permits of the construction of a very small and cheap trap which can also serve usefully as an automatic air-vent. Such traps, however, must not be used with superheated steam, nor in situations where they are subject to water hammer.

Traps draining any apparatus working with steam at more than atmospheric pressure can be used to discharge the condensate to an elevated tank whence it can flow to the inlet of the feed-pumps. This is a practically necessary arrangement for the pump when the condensate is at all hot. When traps are used for raising the discharged water to a higher level, a non-return valve should be placed at the outlet of the trap. The practical limit of lift can be taken as 2 ft. per pound of gauge pressure of the steam at the trap. Special forms of trap are manufactured by Messrs. Hopkinson Ltd. and other firms to deliver the water of condensation directly into the boiler.

General Notes on Steam-Trap Practice.

Steam traps should always be placed at the lowest point of the apparatus to be drained, and care must be taken that the water of condensation has a free flow to them, without the possibility of air-locks. When draining a steam main it is important to ensure that the connection to the trap does not project into the main and thus prevent the last drop of water being taken away. It is good practice to drain from a pocket welded to the underside of the main. A length of bare pipe on the upstream side of the trap will facilitate the action of the latter by cooling the condensate, but of course, only at the expense of a loss of heat.

It is good practice to every steam trap with a by-pass, but only on the condition that the by-pass is never improperly used. Its presence enables the trap to be examined or adjusted while the plant is working, and it also contributes to safety at starting up if opened for a short time to clear all water and air out of the system. On the other hand, unless it is properly closed at normal times, it is so wasteful a fitting that some engineers prefer to dispense with it altogether.

When a trap discharges into a sealed drain, it is convenient to have a three-way cock at the trap outlet, so that the trap can be adjusted with the cock open to atmosphere. Also turning the cock to this position enables the fact that the trap is working properly to be verified at any time.

The appearance of steam at the outlet of a trap is not necessarily a proof that the discharge valve is leaking, for whenever water under pressure and at a temperature greater than 212° F. is released to the atmosphere, some of the water will automatically flash into steam.

All traps should be situated where they are readily accessible for routine inspection and servicing, and no trap of the float or bucket type must be placed where it is liable to become frozen.

In general the greatest trouble with steam traps is due to dirt, scale, grit or some other foreign substance getting into them. Before they are installed the pipe work should be blown through with steam, and every care taken that it is thoroughly clean. Whenever there is any danger of

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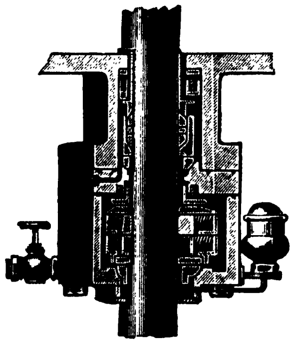

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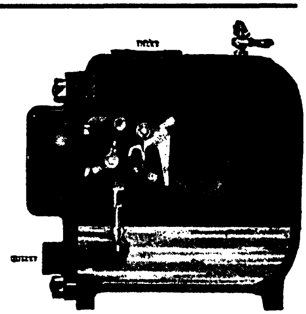

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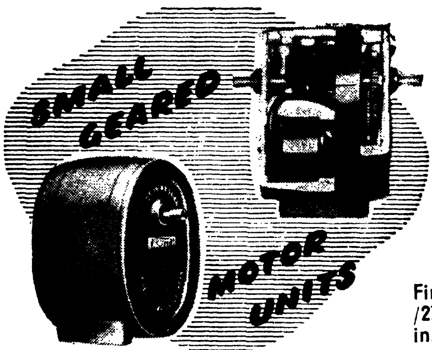

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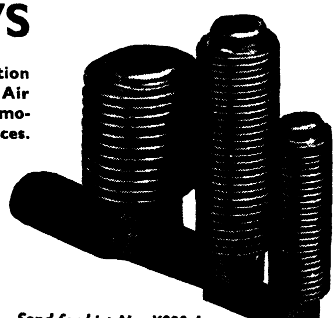

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dirt being carried into a trap of the float or expansion type, a strainer of wire gauze must always be fitted ahead of the trap. With bucket traps a strainer is less necessary. Suitable strainers are supplied by the trap makers, and they must be regularly inspected and cleaned.

In purchasing traps, the makers should be informed of the maximum and minimum pressures at which the trap will have to work, and the rate at which it will be required to discharge. The size of the pipe connections to a trap is no guarantee of its discharging capacity. A good trap is well worth what it may cost, for traps of cheap and shoddy design are a perpetual nuisance and usually a source of serious waste.

See also Descriptive Section XXVII, Part II.

SECTION XXVII

PART III

RECIPROCATING STEAM ENGINES

EXPANSION OF STEAM — HORSE-POWER—STEAM CONSUMPTION—INDICATOR AND INDICATOR DIAGRAMS—ENGINE TYPES FOR VARIOUS DUTIES — ENGINE DESIGN—FRAMES — CYLINDERS — BEARINGS — FLYWHEELS—GOVERNORS—SLIDE VALVE GEARS—LINK MOTIONS

(Revised by R. H. Parsons, M.I.Mech.E.)

Expansion of Steam.

It is found in practice that in engines using saturated steam the expansion line follows closely the hyperbolic law :—

absolute pressure × volume = a constant.

Hence it is usual to compute indicator diagrams on the assumption that the expansion is hyperbolic. With highly superheated steam the pressure falls more rapidly than the hyperbolic curve indicates.

Referring to fig. 1, the theoretical indicator diagram is A B C D O, there being no clearance and a perfect vacuum. Steam is cut off at B and exhausted at C.

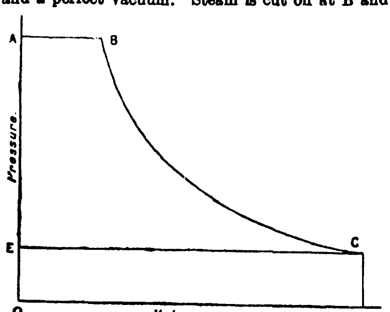


FIG. 1.

p = **absolute pressure in lbs. per sq. in.;**

u = **volume in cubic feet;**

R = **ratio of expansion =** $\frac{OD}{AB}$;

W = **area of diagram ;**

P = **mean effective pressure in lbs. per sq. in. (M.E.P**

pu = **a constant.**

then $W = ABOE + EODO$

$= p_2 u_2 \text{ hyp. log } R + p_2 u_2.$

$P = p_1 \left(\frac{\text{hyp. log } R + 1}{R} \right) = p_1 K.$

TABLE OF MEAN PRESSURE CONSTANTS K.

The table also gives the points of cut-off (nominal), based on assumed cylinder ratios, L.P. to H.P., in the case of multiple-expansion engines.

R.	K.	Nominal Cut-off (clearance neglected) as proportion of Stroke.			
		Single Cylinder, including Uniflow	In H.P. Cylinder of Multiple-Expansion Engine.		
			Compound. Cylinder ratio = 4.	Triple. Cylinder ratio = 6.	Quadruple. Cylinder ratio = 8.
1·25	0·979	0·8			
1·4	0·955	0·72			
1·6	0·919	0·62			
1·8	0·882	0·55			
2·0	0·847	0·50			
2·5	0·767	0·40			
3·0	0·700	0·33			
3·5	0·644	0·29			
4·0	0·597	0·25			
4·5	0·556	0·22			
5·0	0·522	0·20			
6·0	0·465	0·17	0·67		
7·0	0·421	0·14	0·57		
8·0	0·385	0·12	0·50		
9·0	0·355	0·11	0·44		
10·0	0·330	0·10	0·40	0·60	
11·0	0·309	0·091	0·36	0·55	
12·0	0·290	0·083	0·33	0·50	
14·0	0·260	0·072	0·29	0·43	0·57
16·0	0·236	0·062	0·25	0·38	0·50
18·0	0·216	0·055	0·22	0·33	0·44
20·0	0·200	...	0·20	0·30	0·40
25·0	0·169	...	0·16	0·24	0·32
30·0	0·147	...	...	0·20	0·27
35·0	0·130	...	...	0·17	0·23
40·0	0·117	...	...	...	0·20

(*Morley.*)

The referred M.E.P. of compound engines is the equivalent M.E.P. if the entire expansion were to take place in the low-pressure cylinder.

The nominal ratio of expansion is the stroke volume of the L.P. cylinder ÷ volume swept by H.P. piston up to point of cut-off, or

$$R = \frac{V_L}{a\,V_H} = \frac{\text{cylinder ratio}}{a}$$

where V_H, V_L are stroke volumes of H.P. and L.P. cylinders, and a is proportion of stroke at which cut-off takes place in the H.P. cylinder.

If O_H and O_L are the clearance volumes at each end of the H.P. and L.P. cylinders, the real ratio of expansion is

$$\frac{V_L + O_L}{a\,(V_H + O_H)}$$

To determine the probable actual mean pressure (referred to the low-pressure cylinder in the case of a multiple-expansion engine) multiply the initial pressure (absolute) by K, deduct the back pressure, and multiply by a suitable 'diagram factor.'

Diagram factors depend on the type of engine, type of valves and valve gear, jacketing, superheating, speed, etc. Approximate typical values are as follows:—

TABLE OF DIAGRAM FACTORS.

Triple-expansion	·65–·75
Compound condensing, superheated	·65–·75
" " saturated	·7– ·8
" non-condensing	·8–·85
Simple condensing, trip gears	·9–·95
" " slide valves	·85– ·9
" non-condensing	·9–·95

Note.—In calculating the M.E.P. the clearance is ignored. If the clearance is allowed for, the ratio of expansion for a given cut-off is lower and the theoretical M.E.P. higher, necessitating a lower diagram factor, particularly for simple and compound engines. Thus, some authorities give from 0·7 to 0·9 for simple engines.

Low revolutions, large clearance, jackets, reheaters, and low compression increase the diagram factor. These special conditions must be allowed for, as in slow-speed pumping engines.

Usual Expansions in Various Types of Engines.

Single Condensing Engines	from 2 to 5 times
Two-cylinder Compound Engines	" 10 " 20 "
Triple-expansion Engines	" 12 " 25 "
Quadruple-expansion Engines	" 16 " 30 "
Uniflow Engines	" 7 " 12 "

INDICATED HORSE-POWER OF STEAM ENGINES.

P = Mean effective pressure, in pounds per square inch;
A = Area of cylinder in square inches;
L = Stroke in feet;
N = Revolutions per minute;

then I.H.P. (for a double-acting engine) $= \frac{2\,PLAN}{33{,}000}$

The I.H.P. of a compound or multiple expansion engine is the sum of the powers of the individual cylinders, each calculated as above. For such engines, the 'referred M.E.P.' or 'M.E.P. referred to the L.P. cylinder' is frequently spoken of and used in designing. It means that M.E.P. which, if it acted in a single cylinder of the same dimensions as the L.P. cylinder of the compound engine and working at the same speed, would produce the same power.

The '*referred M.E.P.*' is thus equal to M.E.P. of H.P. cylinder $\times \frac{\text{H.P. cylinder area}}{\text{L.P. cylinder area}}$

$+$ M.E.P. of I.P. cylinder $\times \frac{\text{I.P. cylinder area}}{\text{L.P. cylinder area}}$ $+$ M.E.P. of L.P.

For a two-cylinder compound engine the middle term disappears.

Brake Horse-Power (B.H.P.).

'Brake' or 'Effective' or 'Actual' horse-power is the horse-power delivered by the engine to whatever it is driving. If the engine shaft is coupled to the driven machine—such as a generator for electric power, or a rolling mill, or whatever it may be—the B.H.P. is the power transmitted at the coupling. For a rope drive it is the power at the engine rope pulley. In all cases the B.H.P. is the indicated H.P. less the power absorbed in the running of the engine itself. This latter power—the frictional H.P.—is about 10 per cent. in good modern engines, 8 per cent. in those of the highest class, but may be as much as 20 per cent. in small engines of crude design.

Mechanical Efficiency.

The 'mechanical efficiency' is the ratio of B.H.P. to I.H.P. For high-class engines of, say, 100 H.P. and upwards, typical values of the mechanical efficiency at various loads are as follows:—

Load	Full	¾	½	¼
Mechanical efficiency, per cent.	92	90·5	88	80

Theoretical Steam Consumption.

The theoretical steam consumption of an engine is the weight of steam per h.p. hour which would be required by an ideally perfect engine working on the Rankine cycle with the given live steam and exhaust conditions. This can be most readily calculated by the following formula when, as is practically always the case, the exhaust steam is not in a superheated condition.

$$\text{Theor. lbs. of steam per h.p. hour} = \frac{2{,}545}{H_1 - H_2 + T_2(\phi_2 - \phi_1)}$$

where H_1 = B.Th.U. per lb. of live steam,
H_2 = ,, ,, ,, saturated steam at exhaust pressure.
T_2 = Absolute temp. in deg. Fahr. of saturated steam at exhaust pressure.
ϕ_1 = Entropy per lb. of live steam.
ϕ_2 = ,, ,, ,, saturated steam at exhaust pressure.

When the steam is superheated at exhaust, the formula becomes simply—

$$\text{Theor. lbs. of steam per h.p. hour} = \frac{2545}{H_1 - H_3}$$

where H_3 = B.Th.U. per lb. of steam at exhaust pressure and temperature.

Values H and ϕ are to be found in Callendar's Steam Tables. (See also p. 3 *et seq.*)

EFFICIENCY OF ENGINES.

The thermal efficiency—of the engine alone, exclusive of boiler plant—is the ratio of the work done per lb. of steam (in B.Th.U.) to the heat required to generate 1 lb. of steam, at the pressure and temperature, at the engine stop valve from feed water at the temperature of the engine exhaust.

$$\text{Thermal efficiency } e = \frac{\text{I.H.P.} \times 42 \cdot 42}{w(H_1 - h_3)} = \frac{\text{I.H.P.} \times 2545}{W(H_1 - h_3)},$$

where,

w = steam used, lbs. per minute; W = do., lbs. per hour;
H = total heat at pressure and temperature at boiler side of engine stop valve;
h_3 = water heat at temperature of exhaust.

It is customary to express the performance of engines in steam consumption per h.p. hour, but since the heat required to generate the steam increases with pressure and temperature the steam-consumption figure alone is not a complete criterion of the merit of the engine.

EFFICIENCY RATIO.

The ratio of the thermal efficiency of the actual engine to that of the Rankine cycle—or of the steam consumption of the Rankine cycle to the actual steam consumption (both having the same pressure, superheat, and vacuum or back pressure)—is the 'efficiency ratio.'

The efficiency ratio varies with the pressure, superheat, and vacuum, as well as with the merit of the engine as regards freedom from condensation, leakage, and other losses.

Nevertheless the 'efficiency ratio' is useful in comparing engine test results and in predicting steam consumptions.

Economy of High-pressure Working.

The consumption of steam in pounds per horse-power per hour decreases as the pressure rises. The actual steam consumptions vary according to the size and type of engine.

Steam Saving by Superheating.

Superheating improves the efficiency ratio, so that in practice the saving due to superheating is much greater than that calculated for the Rankine cycle.

Superheating reduces the steam consumption of engines approximately at the rate of 1 per cent. for each 10° F. of superheat, but the rate is rather greater for the lower grades of superheat and less for higher grades.

The practical steam and fuel economy due to the use of superheated steam are about as follows, all superheats being measured at the engine stop valves :—

Approximate Percentage Reduction in Steam Consumption by Superheating, for Compound and Uniflow Engines.

Superheat, deg. Fahr.	50	100	150	200
Common Steam and Exhaust Ports	12	20	26	30
Separate " " " "	10	16	21	25
Uniflow, about	8	12	17	21

(*Foster.*)

Superheat reduces the steam consumption most in simple engines.
For superheat in steam turbines see Part IV, page 140.
The advantages of superheating in general are chiefly :—

1. **Greater thermal efficiency of the engine, due to reduced condensation loss.**
2. **If the boiler plant is not provided with very ample heating surface the addition of a superheater increases the boiler plant efficiency and capacity apart from the engine effect.**
3. **As compared with saturated steam, higher steam velocities in the pipes may be used with superheated steam, and consequently the pipes may be smaller. For this reason, the percentage loss of heat in the pipe system is lower, in spite of the higher temperature.**
4. **Reduction in size of auxiliary plant owing to smaller weight of feed water and coal.**

Degree of Superheat.

In modern practice the following are usual :—

General engine practice: superheat at boilers 150° F., or, say, 120° to 130° at engine. Up to 650° F. total temperature, say 260° F. of superheat at 200 lbs. pressure, can be used if the engine is designed for it.

In general, with flat slide or Corliss valves on the high-pressure cylinder (about 550° F. total temperature) must not be exceeded.

For turbine practice, see Part IV. page 143.

Characteristics of Superheat.

Condensed steam is a slight lubricant. The absence of this must be met by adequate lubrication when using superheated steam. The high temperature requires a special oil of high flashpoint. Pipe friction losses are reduced by superheating.

In reciprocating engines a later cut-off is required to develop the same horse-power. Hence the maximum power which can be developed is slightly reduced by superheating.

Moderate superheat, about 50° or 100° F., is desirable for evaporating and similar heating purposes, especially with long pipe mains, but very high superheats are undesirable, as the steam should have little, if any, superheat by the time it reaches the heating surfaces of the evaporators.

Throttling slightly raises the superheat in steam; expansion, as in a turbine nozzle, very greatly reduces it.

Except for very moderate superheat, cast-iron and copper must not be used for pipes and valves in contact with the steam.

STEAM JACKETS.

Steam jackets are rarely adopted to-day except for slow-running engines. There is no doubt that a well-arranged jacket leads to considerable economy when the engine is supplied with saturated steam. The economy is much less marked with superheated steam, especially as jackets are usually arranged. Having regard to the extra first cost and complication, jackets are not to be recommended with superheated steam unless the conditions are exceptional.

The usual arrangement of steam supply is defective; the steam is supplied to the jacket and a trap fitted to drain away the condensed steam, but there is no through flow. Usually, on intermediate and low-pressure cylinders the jacket steam pressure is reduced to a few pounds above the steam chest pressure; except to relieve the cylinder barrels and liners of mechanical stress, there is no justification for such an arrangement; from the point of view of steam economy it is ridiculous and largely nullifies the value of the jackets. Wherever possible, the jacket steam should have a through-flow. With saturated steam the engine steam should pass through the jackets and then through a water separator before entering the cylinders.

When jackets are used with steam superheated 100° F. or more, they are unnecessary on the high-pressure cylinder.

Small and slow-running engines benefit most from jackets.

Reheaters, *i.e.* appliances for heating the steam in the receivers between the cylinders, do not appear to be useful with saturated steam.

With good initial superheat, say over 150° F., reheaters give good results.

Steam Consumption

Typical steam consumptions, with the corresponding efficiency ratios, are shown in the following table:—

TYPICAL STEAM CONSUMPTIONS AND EFFICIENCY RATIOS.

Type of Engine.	Pressure at Stop Valve. Lbs. per Sq. In. (Gauge.)	Superheat °F.	Condensing. Steam. Lbs. per I.H.P. Hour.	Condensing. Efficiency Ratio.	Non-Condensing. Steam. Lbs. per I.H.P. Hour.	Non-Condensing. Efficiency Ratio.
Simple	160	—	17 to 17·5	51 to 49	21 to 22	67 to 64
	160	150	12 to 13	65 to 61	16 to 17	78 to 74
	120	—	18·5 to 19·5	49 to 47	24 to 25	65 to 62
	120	150	14·5 to 15	57 to 55	19 to 20	73 to 70
	80	—	22 to 23	45 to 43	30 to 33	62 to 56
	80	150	16 to 17	56 to 53	24 to 26	69 to 64
Compound	200	—	12·5 to 13	66 to 64	—	—
	200	150	10 to 10·5	74 to 71	—	—
	160	—	13 to 13·5	66 to 64	17·5 to 18·5	80 to 75
	160	150	10·5 to 11	74 to 71	14·5 to 15	86 to 83
	120	—	14 to 15	65 to 61	19·5 to 20·5	79 to 75
	120	150	11·5 to 12·5	72 to 67	16 to 17	86 to 83
	80	—	—	—	25 to 26	74 to 71
	80	150	—	—	20·5 to 21·5	81 to 77
Triple	200	—	11·5 to 12	72 to 69	—	—
	200	150	10 to 10·5	75 to 71	—	—
	180	—	12 to 12·5	70 to 67	—	—
	180	150	10·2 to 11	75 to 70	—	—
Uniflow	200	—	11·8 to 12·3	70 to 67	—	—
	200	150	10 to 10·5	75 to 71	—	—
	160	—	12·5 to 13	69 to 66	19 to 20	74 to 70
	160	150	10·5 to 11	75 to 71	16 to 17	78 to 74
	120	—	13·5 to 14·5	68 to 63	21·5 to 22·5	73 to 69
	120	150	11·5 to 12·5	72 to 67	18 to 19	78 to 73

The particulars in the table refer to high-class modern engines with Corliss or drop valves and of about 600 i.h.p. or upwards working at normal full load.

In the case of condensing engines, the vacuum has been taken as 26 ins.

The non-condensing uniflow values are subject to considerable variations.

The figures given being for engines of the highest class, for good average practice add from 5 to 10 per cent. For badly clothed or defective engines add 20 per cent. or more. For small engines add from 5 per cent. upwards according to size. The superheat is measured at the engine stop valve. The loss of temperature between superheater and engine may be from 20 to 100 degrees, according to steam-pipe conditions. Simple (alternating-flow single-cylinder) engines vary a good deal according to the cut-off. In general, a uniflow has a rather lower steam consumption than a good compound engine.

Duty of Engines.

The duty of an engine is rarely spoken of except for pumping engines, and is then expressed in terms of the work done in lifting the water per 1,120 lbs. of steam used, or some other basis. Sometimes 1,000 lbs. of steam is used, especially in America; and sometimes coal or heat units is the basis.

Example.—Pumping engine consumes 14 lbs. of steam and 1·8 lbs. of coal per water, or pump h.p. hour. The total heat added per lb. of steam is 1,125 B.Th.U.

Duty per 1,120 lbs. steam is $\frac{1,120}{14} \times 33,000 \times 60$ . . . = 158,400,000 ft. lbs.

Duty per 112 lbs. coal is $\frac{112}{1 \cdot 8} \times 33,000 \times 60$ . . . . = 123,200,000 „

Duty per 1,000,000 B.Th.U. in steam is $\frac{1,000,000}{1,125} \times \frac{158,400,000}{1,120} = 125,700,000$ „

The steam consumption of high-class slow-speed triple-expansion pumping engines is very low.

The duty of Winding Engines is rarely measured, but is sometimes stated in terms of steam used per shaft h.p. hour, or per h.p. hour in the coal hoisted. This steam consumption increases greatly when the cages are not fully loaded and the number of winds is below maximum. Under favourable conditions compound non-condensing engines use from 30 to 50 lbs. of steam per shaft h.p. (in the coal raised) per hour. For high pressure engines and unfavourable conditions the consumption increases to about 100 or 120 lbs., and occasionally more.

Tests of a cross-compound non-condensing engine at Sherwood colliery gave a steam consumption of 40 lbs. per S.H.P. hour. The engine cylinders were 32 ins. and 53 ins., stroke 66 ins., pressure 125 lbs. per sq. in., superheat 65° F.

For the steam consumption of turbines *see* Part IV, Steam Turbines, p. 143.

THE INDICATOR DIAGRAM

General Definitions.

The Atmospheric Line, AB, fig. 2, is a line drawn by the pencil of the indicator when the connections with the engine are closed and both sides of the piston are open to the atmosphere. This line represents on the card the pressure of the atmosphere, or zero gauge pressure.

*The Line of **Absolute Zero Pressure***, OX, is a reference line usually drawn about 14·7 lbs. by scale below the atmospheric line. It represents a perfect vacuum, or absence of all pressure.

At any considerable height above sea level the line OX becomes nearer the line AB.

The Clearance Line, OY, is another reference line drawn at a distance from the end of the diagram equal to the same percentage of its length as the clearance volume is of the piston displacement. The distance between the clearance line and the end of the diagram **represents the volume of the clearance and of the ports and passages at the end of the cylinder.**

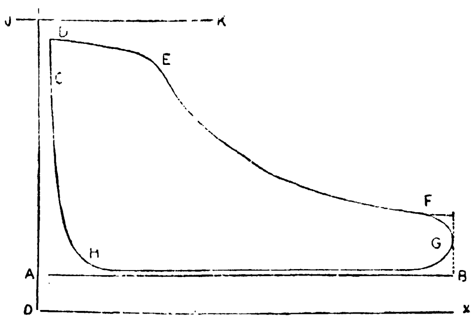


Fig. 2.

The Line of Boiler Pressure, JK, is drawn parallel to the atmospheric line, and at a **distance from it by scale equal to the boiler pressure shown by the gauge. The difference in pounds between it and DE shows the loss of pressure due to the steam pipe and the ports and passages in the engine.**

***The Admission Line* is in two parts. CD, which show the rise of pressure due to the admission of steam to the cylinder by opening the steam valve (if the steam is admitted quickly when the engine is about on the dead centre this line will be nearly vertical), and**

DE, which is drawn when the steam valve is open and steam is being admitted to the cylinder.

***The Point of Cut-off*, E, is the point where the admission of steam is stopped by the closing of the valve. It is difficult to determine the exact point at which the cut-off takes place. It is usually located where the outline of the diagram changes its curvature from convex to concave.**

***The Expansion Curve*, EF, shows the fall in pressure as the steam in the cylinder expands and does work.**

The Point of Release, F, shows when the exhaust valve opens.

FG represents the change in pressure that takes place when the exhaust valve opens; exhaust continues to the point H.

The Back Pressure Line, GH, shows the pressure against which the piston acts during its return stroke. On diagrams taken from non-condensing engines it is either coincident with or above the atmospheric line, as in fig. 2. On cards taken from a condensing engine, however, it is found below the atmospheric line, and at a distance greater or less according to the vacuum obtained in the cylinder.

The Point of Compression, H, is the point where the exhaust valve closes. It cannot be located very definitely, as the change in pressure is at first due to the gradual closing of the valve.

The Compression Curve, HO, shows the rise in pressure due to the compression of the steam remaining in the cylinder after the exhaust valve has closed.

The Mean Effective Pressure (M.E.P.) is the mean net pressure urging the piston forward.

The Initial Pressure is the pressure acting on the piston at the beginning of the stroke.

The Terminal Pressure is the pressure above the line of perfect vacuum that would exist at the end of the stroke if the steam had not been released earlier. It is found by continuing the expansion curve to the end of the diagram, as in fig. 2.

Defects in Diagrams.

A very common and serious defect in diagrams arises from the stretching of the indicator cord. If two diagrams taken from the same engine at the same time show noticeably unequal lengths the cords are probably inaccurate. As a rule the stretch is greatest when the drum is pulled open. Thus one end of the diagram is widened and the other narrowed, giving incorrect M.E.P. and points of valve action.

The cord used should be as short and as inextensible as possible.

The reducing gear should give an accurate copy of the motion of the engine crosshead, and the cord from the reducing gear should run in a direction parallel to the line of motion of the crosshead.

If the diagram is very much to one side of the card, and if the near end is quite straight and vertical, the drum is probably hitting its stop at one end of its travel. Alter the length of the indicator cord.

If the expansion line is stepped in rough terraces the indicator piston is probably sticking and the calculated M.E.P. will be too high.

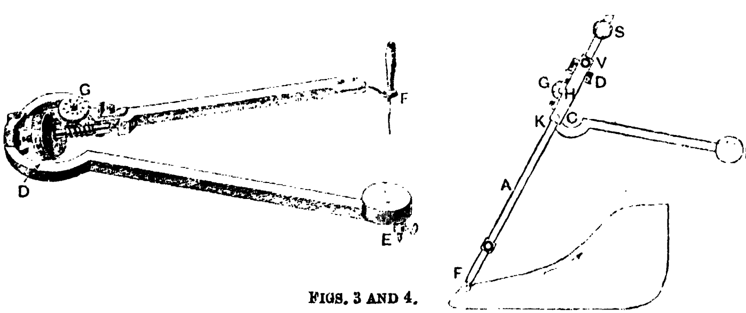
If the expansion is wavy the spring is too light and should be replaced by a stiffer one.

Mean Effective Pressure.

This is best determined by finding the area of the indicator diagram; dividing this by the length, so obtaining the mean height; then, multiplying the mean height by the pressure scale the mean effective pressure is the result.

The best method of measuring the area of the diagram is to use a planimeter, which if carefully handled gives results accurate to hundredths of a square inch.

The most commonly used planimeter is of the Amsler type, and is shown by figs. 3 and 4.



FIGS. 3 AND 4.

It consists of two arms hinged together; one is constrained to swing round a fixed pivot, the other carries a tracing point which is made to follow the outline of the diagram. This arm also carries a roller, with its axis in the centre line of the arm, and provided with a scale and vernier

by which its rotary movements can be accurately measured. As the tracing point moves round the diagram the roller partly slides and partly rolls over the paper or board on which the diagram is drawn or to which it is pinned. The readings of the scale before and after each exact circuit of the diagram are noted; their difference measures the area of the diagram on a scale determined by the proportions of the instrument.

The roller should be pivoted so as to rotate freely, but without endshake, and the surface on which it moves should be flat and unglazed. The position of the pivot point should be chosen so that the tracing point can reach freely all parts of the outline. The pivot point has a needle which can then be pressed into the paper or board.

It is advisable to check the planimeter occasionally by running it round a simple figure, such as a square or circle the area of which can be easily calculated independently.

To Interpret the Indicator Diagram.

Besides the calculation of horse-power and the rough estimate of steam consumption, from the diagram, other important information can be learned from a study of the diagram, as follows:—

The diagram will show the opening and closing of the valves, as explained on p. 79, fig. 2. If the point of cut-off is fixed, as with most slide and piston valve engines, the depth of the steam admission line DE below the boiler pressure line YK is a measure of the loss by throttling. If this is invariably considerable, it may pay to shift the eccentric so as to give an earlier cut off. With a simple slide or piston valve this would make the exhaust open earlier and the compression commence earlier. These, and particularly the latter, must be watched, as too early compression sends the compression pressure above the steam chest pressure. This is not good; although the compression line should rise nearly to the admission pressure. In the case of a reversing engine having a Stephenson, Joy, or similar link motion, the point of cut-off is varied by shifting the equivalent eccentric; and thus at early cut-off the exhaust and compression are both early, but at late cut-off the exhaust and compression both become late. Hence when studying such diagrams these changes of exhaust and compression with cut-off must always be kept in mind. The points of valve action can also be altered by changing the lap. Thus if the exhaust edge of the valve is chipped the exhaust takes place earlier and the compression later. Similarly chipping some of the lap off the steam edge of the valve makes the cut-off later and lead earlier.

If the admission line DE falls rapidly, the valve is too small or the valve travel too small.

In a multiple-cylinder engine the diagrams give the distribution of power and forces among the several cylinders. As a rule these should be roughly equal for the several cranks, but need not be equal for the two cylinders of a tandem engine. Indeed, in the latter case, the h.p. cylinder may with advantage take rather more than half the load if the steam is superheated.

Valve setting should always be checked by actual indicator cards after an engine is first started to work.

It is found in practice that the expansion line (fig. 2, p. 79) is closely hyperbolic when using saturated steam, the two axes of the hyperbola being the absolute zero line OX and a vertical touching the diagram at CD. Hyperbolic curves can be purchased printed on transparent paper for placing over indicator diagrams. If the expansion line rises above the hyperbola as the pressure falls it indicates leakage past the steam valve into the cylinder. Similarly, if the expansion line falls more rapidly than the hyperbola it indicates either leakage past the exhaust valve out of the cylinder, excessive condensation of steam in the cylinder, or the use of initially superheated steam. If superheated steam is used, the vertical axis of the hyperbola should be about the clearance line OY in fig. 2; but the exact position depends on the degree of superheat.

The hyperbola is easily drawn more or less completely as desired when it is remembered that the absolute pressure multiplied by the volume has the same value at all points on the same hyperbolic curve.

ENGINE TYPES FOR VARIOUS DUTIES.

Each type of engine has a field, not sharply defined, in which it shows to best advantage.

Elaborate and expensive types are least desirable when coal is very cheap, the working hours few, labour unskilled, or the working conditions very rough.

Non-condensing engines consume from 20 to 50 per cent. more steam and coal than condensing engines.

Unless floor space is limited or very high speeds are desired, a horizontal engine is generally more satisfactory than a vertical. It is more accessible for running and overhauling.

Steam turbines are relatively uneconomical in very small units. In the neighbourhood of 1,000 to 1,500 horse-power they surpass the best reciprocating engines, and for very large units—unless for some special purpose, as driving a reversing rolling-mill—are without serious rival. To show to advantage a turbine must be run condensing, with a good vacuum.

Where economy is at all important the ordinary simple (single-expansion) engine is at a marked disadvantage, which, however, is least with steam superheated 120° or 150°F. For most purposes the compound is the standard type, although the uniflow (a special form of single-expansion engine) is widely adopted. Triple and quadruple expansion engines are more economical than compound, but are more costly and complex. They are therefore chiefly employed on

ships, for towns pumping, flour mills, and other situations where coal consumption is specially important. Their superiority over the compound engine is reduced if the steam is superheated.

The class of engine is frequently designated by the valve gear. Thus the plain slide valve gives the cheapest engine and also, other things being equal, the least economical.

The piston valve is really a circular slide valve, and gives an engine of similar characteristics. For the highest economy and class of engine Corliss or drop valves are desirable. The Corliss valve is really a circular slide valve of small diameter which rotates to and fro through a small angle in its valve chamber. The drop valve may be of the double-beat seating type or of the double-beat piston type. As a rule Corliss and drop valves are operated by a trip valve gear or other form of valve mechanism enabling the cut-off to be automatically varied to suit the load. Slide and piston valves, on the other hand, are not so suitable for cut-off governing and are not usually so arranged, but rely on throttling the steam to meet changes of load. High-speed engines almost invariably have piston or slide valves, because trip and other cut-off gears are unsuitable at high speeds of revolution.

Cut-off, or expansion, governing gives the best economy for variable loads, except at very light loads, when throttling is superior. The difference is least with highly superheated steam, which tends to equalise the steam consumptions at different loads (measured per horse-power, of course).

The superior economy of Corliss and drop valve engines is often believed to be due to the superior qualities of trip valve gears. Actually this is a very minor factor. The chief gain arises from the fact that the inlet and exhaust ports and valves are usually quite separate.

The following list shows the engine types frequently adopted for various duties :—

Duty.	*Type of Engine.*
Agriculture, occasional service	Simple, slide or piston, N.C.
,, regular service	Compound, slide or uniflow, N.C.
Cement works	Compound, Corliss or drop, uniflow, turbine, S.
Electric generating, small	Compound, high-speed vertical, turbine, S.
,, ,, large	Turbine, S.
Factories, cotton mills, etc.	Compound, Corliss or drop, uniflow, geared turbine, S.
Flour mills	Compound or triple, Corliss or drop, uniflow, S.
Paper mills	Compound, Corliss or drop; uniflow, turbine, also with steam extraction, S.
Portable	Same as agriculture.
Pumping, centrifugal	Compound, usually high-speed vertical, also uniflow if at moderate speed, S.
,, hydraulic and town water	Triple three-crank vertical, slow-speed, Corliss or drop, S.
,, well pumps	Compound or triple, Corliss or drop, S.
Rolling mills, not reversing	Horizontal compound, Corliss or drop, uniflow, S.
,, ,, reversing	Two- or three-crank, piston valves, simple or tandem compound, often N.C., S.
Winding	Two-crank simple or compound (tandem usually), N.C., S.

Note.—N.C. indicates non-condensing. S. indicates superheated steam.

Back Pressure and Extraction Engines.

Increasing attention is now being paid to the saving of fuel, which is rendered possible by the combination of power supply and heating services, the heating being effected by using exhaust steam or steam taken from the engine at an intermediate pressure. Factories using steam for heating in their own industrial processes offer special opportunities in this direction, though schemes have also been successfully carried out in which the heating of adjoining offices and houses has been effected in addition to the heating required in the factory processes.

For example, in a factory requiring 200 i.h.p. for driving machinery, and 3,200 lbs. of steam per hour at about 15 lbs. per sq. in. gauge pressure for heating purposes, the methods of independent and combined supply may be compared as follows. In the independent method an economical condensing engine would be used, taking, say, 11 lbs. steam per i.h.p. hour, or 11 × 200 = 2,200 lbs. steam per hour. The total steam is thus 2,200 + 3,400 = 5,600 lbs. per hour.

In the combined system, a back pressure engine would be used, exhausting at the pressure required for the heating steam. It would require, say, 18 lbs. steam per i.h.p. hour, or 3,600 lbs. per hour. Allowing 10 per cent. for loss by condensation, 3,240 lbs. exhaust steam per hour are available for heating. Thus the steam to be generated for all purposes is 3,600 lbs., at

against 5,600 lbs. in the former case, and even though the whole of this steam has to be generated at high pressure and (preferably) superheated, the fuel consumption at the boiler is reduced in nearly the same proportion. The saving in fuel is thus $\frac{5,600 - 3,600}{5,600} \times 100 = 35 \cdot 6$ per cent. approximately.

By means of such systems the large proportion of heat carried away and lost in the condensing water of even highly economical power plants is greatly reduced; cases are on record in which the heat utilised as work in the engine, together with heat made available for heating purposes, amounts to 77 per cent. of the heat in the steam delivered by the boiler.

In the 'intermediate steam extraction' system, which is adapted for fluctuating power and heating requirements, a compound engine is used, the heating steam is taken from the receiver between the cylinders, and only a part of the steam used in the h.p. cylinder passes to the l.p. cylinder, and thence to the condenser. By means of a pressure regulator, the intermediate pressure is allowed to vary only between fixed limits. With an increased demand for heating steam, the pressure falls slightly, the pressure regulator makes the l.p. cut-off take place earlier, and, if the load on the engine remains unchanged, the speed governor makes the h.p. cut-off correspondingly later; thus more steam passes through the h.p. cylinder and less through the l.p. Provision is usually made for supplementing the 'extraction' steam, when necessary, by steam taken directly through a reducing valve from the live steam main; this additional supply is also controlled automatically. The l.p. cylinder may with advantage be of the 'uniflow' type.

On account of the variations in the distribution of the power between the two cylinders with varying demands for extracting steam, the extraction engine usually has its cylinders arranged in tandem.

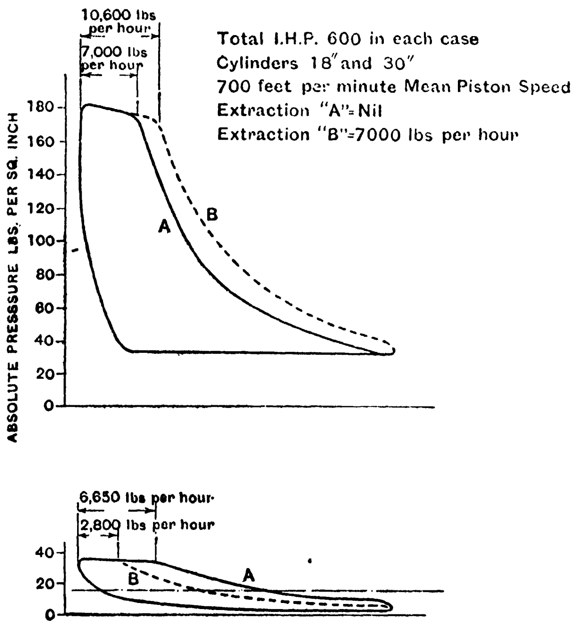


FIG. 5.

This is not, however, absolutely essential, and, particularly if the extraction steam is comparatively small in quantity, the engine may be of the cross compound arrangement.

Typical indicator diagrams for an extraction engine are shown in fig. 5.

The steam quantities in each cylinder with and without extraction are marked on the diagrams, the total load being the same in both cases.

Oil in Exhaust Steam.

The objection is often raised that steam exhausted from, or extracted from, a reciprocating engine is unsuitable for use in contact with textile fabrics on account of its contamination with lubricating oil.

Vertical engines may be operated with bronze piston rings and no cylinder oil in order to avoid this objection.

The risk of trouble due to oil is, however, much exaggerated. The experience obtained in using exhaust steam from oil-lubricated cylinders in bleaching and dyeing works where it comes into contact with all sorts of fabrics and all sorts of dyes justifies the conclusion that no trouble due to oil need be feared. Not the slightest effect of oil on the fabrics has been detected in many years of working under such conditions.

An oil separator should, of course, be provided, although it will not remove absolutely the whole of the oil.

ENGINE DESIGN.

The type of engine is first settled from a consideration of duty, steam consumption, price of fuel, space available, working conditions, water supply, first cost, etc.

Cylinder proportions, stroke and speed of revolution are then fixed. Typical British cylinder ratios and mean effective pressures are given in the tables below. On the European Continent the cylinder ratios adopted are usually considerably below the tabular values. That is, a larger high-pressure cylinder is adopted. This makes for economy with superheated steam, but for a two-crank engine tends towards unequal distribution of load on the two cranks which is not altogether desirable. This objection does not apply in the case of tandem engines.

TYPICAL MEAN EFFECTIVE PRESSURES AT NORMAL FULL LOAD IN LBS. PER SQ. IN.

Boiler Pressure—Gauge	80	100	120	140	160	180	200
Simple Condensing, about	30	34	37	—	—	—	—
„ Non-Cond. „	35	40	43	—	—	—	—
Compound Condensing	25	27	28	29	30	31	32
„ Non-Condensing	26	30	34	37	40	—	—
Triple Condensing	—	—	—	27	29	30	30·5
Uniflow Condensing	30	33	35	37	38	39	40
„ Non-Condensing, about	26	30	34	37	40	—	—

Note.—The above are often exceeded 5 or 10 per cent.

MEAN PISTON SPEEDS, FT. PER MIN.

Type of Engine.	Piston Speed.
Horizontal, Slide valve	240–500
Horizontal Corliss, and drop valve	500–750
High speed	500–1,000
Pumping, direct drive	120–180
Locomotive	1,000–1,400

USUAL BRITISH CYLINDER RATIOS. HORIZONTAL. CUT-OFF GOVERNING.

Boiler Pressure, lbs. Gauge	100	120	140	160	180	200
Initial Piston Press., absolute	110	130	150	170	190	210
Compound Non-Condensing	2·3–2·5	2·5–2·7	2·8–3·0	3·0–3·2	—	—
Compound Condensing	3·0–3·3	3·2–3·5	3·7–3·8	4·0	4·3–4·6	4·5–4·7
Triple Condensing:						
L.P. to H.P.	—	—	5·2–5·6	5·6–6·0	5·9–6·3	6·0–6·6
I.P. to H.P.	—	—	2 3–2·36	2·36–2·45	2·43–2·5	2·45–2·67

Having fixed the cylinder sizes the probable steam pressures on the pistons are put down, and from these the total piston loads are obtained. From these, by allowing suitable stresses and bearing pressures, the principal sizes are fixed.

Usual stresses in high-class engines are as follows :—

USUAL ENGINE STRESSES.

Crank and crosshead pins, up to	9,000 lbs. sq. in.
Crankshaft journals	7,000–9,000 „
Piston rod, net tension at cotter hole	5,000–6,000 „
Piston rod, gross tension (direct plus bending in body)	9,000 „
Connecting rod bolts	4,000–6.000 lbs. sq. in.
Main studs and bolts	5,000 „
Lesser „ „	4,000–6,000 „
Connecting rod body, small end	3,000–6,000 „
Shear on cotters	3,000–5,000 „
Bending on cotters	up to 7,000 „
General forgings	4,000–8,000 „

Cast Iron in Tension.

Simple frames, direct pull	700–900 lbs. sq. in.
Frames, with some bending	500–800 „
Complex frames, with bending	500–600 „
Flywheel arms (C.F.)	1,500–2,500 lbs. sq. in.
Cylinder barrels, new	1,200–2,000 „
„ „ usual	1,500 „

For engines, such as rolling mill engines, subject to shocks, the stresses calculated on the steady load should be less—e.g., a 9-in. crank-pin of good material failed, although the stress calculated from steady load conditions was 5,650 lb. per sq. in.

(*From an Insurance Co. Report*, 1930–31.)

It should be noted that in calculating the above frame tensions only the direct pull is included. If bending is calculated separately and added on, very high figures are sometimes reached.

HORIZONTAL ENGINE BEARING PRESSURES.

Bearing.	Lbs. per Sq. In.	Bearing.	Lbs. per Sq. In.
Crosshead pins	1,200–1,300	Second motion bearings	100–200
Crank pins	800– 950	Crosshead slides	40– 50
Crankshaft journals (steam load)	100– 150	Intermediate piston rod slides	20– 40
Crankshaft journals (combined loads)	150– 200		

USUAL BEARING PRESSURES FOR VERTICAL HIGH-SPEED ENGINES.

Bearing.	Lbs. per Sq. In.	Bearing.	Lbs. per Sq. In.
Main bearings	200– 300	Crosshead slippers	40–50
Crank pins	400– 550	Eccentric straps	60–70
Crosshead pins	1,000–1,200		

For journals with steady loading running at high speeds, see p. 134.

ENGINE DESIGN.

As an example, take the case of a cotton mill engine of 1,500 i.h.p. normal load, the engine to be of the cross-compound condensing type ; boiler pressure 160 lbs.

The cylinder ratio we take to be 4 nominal, piston rods being ignored. We aim at equal loading on the two cranks so that of the total drop in pressure from stop valve to condenser one-fifth will take place in the low-pressure (l.p.) cylinder. Allowing 12 lbs. vacuum and losses of 5 lbs. and 8 lbs. at the steam pipe and intermediate receiver respectively, the total effective pressure drop is 159 lbs., of which 32 takes place in the l.p. cyl. and 127 in the h.p. If the engine has a stroke of 5 ft. and runs at 66 r.p.m. (higher speeds are coming into favour), the piston speed is 660 ft. per min., and the area of the l.p. cylinder for a m.e.p. of 30 lbs.

$$= \frac{1{,}500 \times 33{,}000}{660 \times 30} = 2{,}500 \text{ sq. ins.}$$

This is taken as the gross piston area, giving a diameter of 56·5 inches. The actual m.e.p. would therefore be about 31 lbs. The h.p. cylinder should then be 28·25 ins. diam., but odd dimensions are unsatisfactory, and the cylinders will be fixed at 28 and 56 inches, giving a nominal m.e.p. of 30·5 lbs., which is rather below modern practice, which tends rather to higher m.e.p. than given in the table above.

Ignoring the piston rods, the gross load on the l.p. piston is 32 × 2,500 = 80,000 lbs. The load on the h.p. piston is the same. Owing to the effect of the rods this value is rather high, but the difference may be ignored, and 5 per cent. added to cover variations in the load distribution, making the total piston load 84,000 lbs. Taking bearing pressures of 1,250 and 850 lbs. per sq. in. on the crosshead pin and crank pin, the projected areas of these become 67·2 and 98·8 sq. ins. respectively. We may thus fix the pin bearings (diam. × length) as 6·75 × 10 and 10 × 10 ins.

We now turn to the size of the crankshaft neck. The bending 'arm' from centre of crank pin to centre of main crankshaft neck, we estimate at about 26 ins., so that the bending moment is M = 84,000 × 26 = 2,180,000 inch lbs. The twisting moment is

$$T = 84{,}000 \times 30 = 2{,}520{,}000 \text{ inch lbs.}$$

The equivalent bending moment is

$$M_e = \tfrac{1}{2}\left[M + \sqrt{M^2 + T^2}\right] = 2{,}755{,}000.$$

Adopting a stress of 8,000 lbs. per sq. in., the modulus of the neck section is

$$Z = \frac{2{,}755{,}000}{8{,}000} = 344.$$

For a solid neck of diam. d, $Z = 0{\cdot}0982d^3$. Hence $d = 15{\cdot}2$ ins., say 15·5 ins. The length of the neck must be determined to give a suitable bearing pressure from dead weight and steam load.

The piston rod diam. is next fixed, and partly from it and partly directly from the load the proportions of the connecting rod. The piston rod is proportioned so that the stress round the cotter hole at the crosshead is about 5,000 to 5,500 lbs., whilst the total direct tension plus tension due to bending (assuming a tail support to the rod) does not exceed about 9,000 lbs. The net section of the frame will lie between about 100 and 170 sq. ins. according to type, the highest stresses being allowed where there is least bending.

The size of the crosshead slipper is determined as follows:—If the connecting rod is 5 cranks long, centre to centre, divide the total piston load by 5. The result, 16,800 lbs. in this case, is approximately equal to the maximum load on the slide due to piston pressure. To this should be added the dead weight in the case of a horizontal or inclined engine, giving, say, about 18,000 lbs. in all. Taking a pressure of 45 lbs. per square inch (less if the pressure is steady and not highly variable), gives a slipper area of 400 sq. ins.

FRAMES.

Bedplates and other frames are either of strong cast iron or of fabricated steel. They should be so formed as to reduce bending to a minimum consistent with convenience for the engine attendant and reasonable cost. To secure this they should be as nearly as possible symmetrical on both sides, and also above and below the centre line. This does not mean strict geometrical symmetry, as this is very rarely attainable except as regards the two sides; but it implies an approximately equal distribution of stress. Circular bores and turned joints make for truth in alignment and often for cheapness in manufacture. Ample bearing surface on the foundations must also be allowed.

CYLINDERS.

Cylinders should be hard and strong. Apart from questions of strength, for information on which see the table of stresses on page 85, the following important points should not be overlooked:—

(1) The clearance volume should be as small as reasonably possible.

(2) The clearance surface should be reduced to a minimum.

(3) The exhaust and inlet ports are better separated.

(4) The ports should be short, direct, and not too large. Large ports involve increased clearance surface.

(5) The covers which, however, are closely influenced by the piston details, should be free from recesses and curves.

(6) Where there are jackets the steam should be arranged to have a decided through flow, the more rapid the better.

(7) Proper relief valves and drainage should be provided.

(8) Metal bosses and projections connecting to the frames or atmosphere facilitate heat radiation, and should be kept small.

The port sizes are based on a steam velocity of 100 to 125 feet per second for inlet ports, and 90 to 120 feet per second for exhaust ports when using saturated steam. With superheated steam these speeds are increased to 130 or even 150 feet per second.

Area of Steam Passages and Valves.

The area is usually calculated from the mean piston speed. Thus if the pipe or valve area is one-tenth the piston area and the piston speed is 600 ft. min. or 10 ft. sec. the steam speed through the pipe or valve is 100 ft. sec.

Usual values for steam and exhaust valves, ports, and pipes:—

Saturated steam	100 ft. per sec.
Superheated steam	125 ,,

For drop-seated valves add about 50 per cent. at high speeds, and up to 100 and 150 per cent. for uniflow engines.

For steam turbines base on actual weight of steam, but assumed saturated, and allow 90 ft. per sec. This is equal to about 120 ft. sec. for steam superheated about 150° Fah. at the turbine. A rough rule is 1000 lbs. of steam per hour at rated load per sq. in. of pipe area.

For the main steam range, unless this is very short, base on actual volume, assumed saturated, flowing at about 90 ft. per sec.; rather less, say, 75 to 80 ft. per sec., for reciprocating engines or pipes below 4 ins. in diameter.

BEARINGS.

With very rare exceptions oil is dragged into a bearing by the relative movement of the two so-called rubbing surfaces by a kind of wiping or dragging action. To prevent loss of lubrication the oil must not be allowed to escape from the bearing except at the proper place. Ordinary crosshead pin bearings work mostly by suction. The slight pulsation of the pin in the bearing draws in oil on alternate strokes. On the return of the pressure the oil has not time to escape.

Oil grooves should be looked upon as local supply reservoirs.

Bearing clearances should not increase so rapidly as the diameter of the journal. They vary according to the class of work.

The use of white metal bearings (see page 1242 Vol. I) is a safety precaution in case of overheating. It also has constructive merits.

Usual bearing pressures are given on page 85.

For the Michell bearing see page 302.

CRANK CONNECTING-ROD MECHANISM.

If r = crank radius; l = length of connecting rod; n = R.P.M.; θ = crank angle (measured from inner dead centre); x = per cent. stroke of piston (measured from end of stroke remote from crank),

then the following relationships are nearly exact:

$$x = \frac{1 - \cos\theta}{2} + \frac{r}{8\,l}\left(1 - \cos 2\,\theta\right);$$

$$\text{piston velocity} = 2\,\pi\,n r\left(\sin\theta + \frac{r}{2\,l}\sin 2\,\theta\right) = 2\,\pi\,n\,r \times k_1$$

$$\text{piston acceleration} = 4\,\pi^2\,n^2\,r\left(\cos\theta + \frac{r}{l}\cos 2\,\theta\right) = 4\,\pi^2\,n^2\,r \times k_2.$$

TABLE OF CRANK ANGLES AND FACTORS FOR PISTON VELOCITY AND ACCELERATION.

θ = angle of crank for out-stroke.
Angle for in-stroke = 180 − θ, measured from outer dead centre.

Stroke per cent.		$\frac{l}{r}$ = infinity.			$\frac{l}{r} = 5$			$\frac{l}{r} = 4$		
Out.	In.	θ	k_1	k_2	θ	k_1	k_2	θ	k_1	k_2
0	100	0	0	1·00	0	0	1·20	0	0	1·25
1	99	11·5	0·20	0·98	10·5	0·22	1·17	10·3	0·22	1·22
2	98	16·3	0·28	0·96	14·8	0·305	1·14	14·5	0·31	1·19
4	96	23·1	0·39	0·92	21·1	0·43	1·08	20·7	0·435	1·12
6	94	28·3	0·475	0·88	26·0	0·515	1·02	25·4	0·525	1·06
8	92	32·9	0·54	0·84	30·1	0·555	0·965	29·5	0·60	1·00
10	90	36·9	0·60	0·80	33·8	0·65	0·905	33·1	0·66	0·94
15	85	45·6	0·715	0·70	41·9	0·765	0·765	41·1	0·78	0·785
20	80	53·1	0·80	0·60	48·9	0·855	0·63	48·0	0·87	0·64
25	75	60·0	0·865	0·50	55·4	0·915	0·495	54·3	0·93	0·50
30	70	66·4	0·915	0·40	61·5	0·96	0·37	60·4	0·975	0·365
35	65	72·5	0·955	0·30	67·3	0·995	0·245	66·1	1·01	0·235
40	60	78·5	0·98	0·20	73·0	1·01	0·125	71·8	1·02	0·11
45	55	84·3	0·995	0·10	78·7	1·02	0·01	77·3	1·03	−0·007
50	50	90·0	1·00	0·0	84·3	1·015	−0·10	82·9	1·02	0·12
55	45	95·7	0·995	−0·10	90·0	1·00	−0·20	88·6	1·01	0·225
60	40	101·5	0·98	−0·20	95·8	0·975	−0·295	94·3	0·98	0·32
65	35	107·5	0·955	−0·30	101·8	0·94	−0·39	100·3	0·94	0·41
70	30	113·6	0·915	−0·40	108·0	0·89	−0·47	106·6	0·89	0·495
75	25	120·0	0·865	−0·50	114·7	0·83	−0·55	113·2	0·83	0·565
80	20	126·9	0·80	−0·60	121·9	0·76	−0·62	120·5	0·75	0·63
85	15	134·4	0·715	−0·70	129·9	0·67	−0·675	128·6	0·66	0·68
90	0	143·1	0·60	−0·80	139·2	0·555	−0·73	138·1	0·545	0·715
92	8	147·1	0·54	−0·84	143·6	0·495	−0·745	142·5	0·485	0·73
94	6	151·6	0·475	−0·88	148·5	0·435	−0·76	147·6	0·425	0·735
96	4	156·9	0·39	−0·92	154·4	0·355	−0·775	153·5	0·345	0·745
98	2	163·7	0·28	−0·96	161·9	0·25	−0·79	161·3	0·245	0·748
99	1	168·5	0·20	−0·98	167·2	0·175	−0·795	166·7	0·175	−0·749
100	0	180·0	0	−1·00	180·0	0	−0·80	180·0	0	−0·75

The positions of piston and crank for zero piston acceleration are as follows:

	$\frac{l}{r} = \infty$	$\frac{l}{r} = 6$	$\frac{l}{r} = 5$	$\frac{l}{r} = 4$	$\frac{l}{r} = 3$	$\frac{l}{r} = 2$
Stroke per cent.	50	46·1	45·4	44·4	43·2	41·7
Crank angle (degrees)	90	80·8	79·1	76·7	73·2	67·7

Crank Effort and Turning Moment.

The crank effort is the force exerted by the connecting rod on the crank pin in the direction of motion of the pin.

The turning moment (at a particular crank) is the crank effort multiplied by the crank radius.

The total turning moment of an engine is the sum of the moments at all the cranks.

To find the crank effort: From the indicator diagram (the assumed probable diagram in the case of design) and the cylinder area, the total force exerted on each side of the piston by the steam is found for any point of the stroke, and hence also the nett steam force on the piston.

To this must be added, with due regard to algebraic sign, the inertia force, *i.e.* the force required to accelerate (or retard) the reciprocating masses. If the engine is vertical, the weight of the moving parts must also be allowed for.

Thus the force P, say, acting at the crosshead in the direction of motion is found.

To find the inertia force at any position of the piston, it is usually sufficient to find the inertia at the ends of the stroke and to find the position at which the inertia is zero, thus fixing three points on the diagram, fig. 6. A fair curve drawn through these points gives the inertia line RRR.

At the inner dead centre, the acceleration $a = 4\pi^2 n^2 r\left(1 + \frac{r}{l}\right)$; at the outer dead centre $a = 4\pi^2 n^2 r\left(1 - \frac{r}{l}\right)$, and $a = 0$ at a position depending on the ratio $\frac{l}{r}$ (see table, p. 88).

The inertia force = weight of reciprocating parts $\times \frac{a}{g}$.

Fig. 6 shows the form of a typical diagram of forces. AB is the stroke, A the inner and B the outer end. PPP represents the resultant force of the steam on the piston during the out-stroke and QQQ that during the in-stroke. RRR is the inertia line. Ordinates measured from RRR represent the forces at the crossheads in the direction of motion (For a vertical engine, the line RRR is lowered by an amount representing the weight of the parts.)

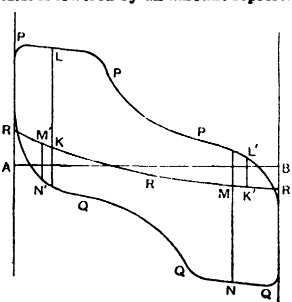


Fig. 6.

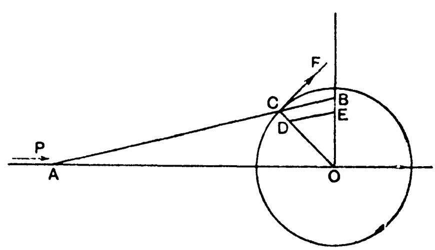


Fig. 7.

E.g. for the out-stroke, ordinates such as KL or K'L', and for the in-stroke, such as MN or M'N', represent the crosshead forces P.

Fig. 7 shows the crank-connecting rod mechanism diagrammatically. If AC (the connecting rod) is produced to meet the perpendicular from O at B, and if OD is measured along OC (the crank) to represent the crosshead force P, and DE drawn parallel to CB, then OE represents the crank effort F.

The ratio OB : OC or OE : OD also represents the ratio of the velocity of the piston to that of the crank pin. Thus the table of piston positions and velocity ratios k_1, may conveniently be used to find the crank effort in place of the graphical method of fig. 7.

That is: Find P as in fig. 7, find k_1 from the table, then the crank effort = force P $\times k_1$.

Starting Torque of Two-Crank Engines.

Engines which have to start against load, such as locomotive engines, winding engines, etc., have usually two cranks at right angles. They should be large enough to start from any crank position with full steam pressure in the cylinders. The starting torque has minimum values at four points in the revolution occurring just after one crank has passed the position of cut-off. At these points, when the steam stop-valve or throttle-valve is opened, only one of the cylinders receives steam and its crank may be in a very unfavourable position.

The four positions of minimum torque are shown in Fig. 8. In position (1) crank A has just passed the cut-off position on the out-stroke; in (2), B has just passed cut-off on the in-stroke; in (3), A, and in (4), B have just passed cut-off. The worst position is (1).

The ratio of the minimum starting torque to the torque produced by the steam load in one cylinder acting at full crank radius, or as it is often termed 'the equivalent crank ratio,' depends

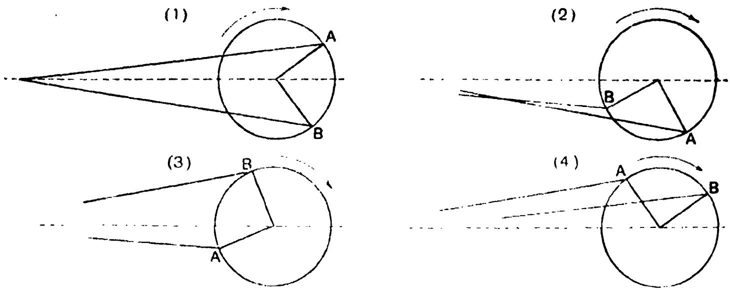


FIG. 8.

on the point of cut-off. For an engine in which the connecting-rod crank ratio is 5, its values are shown in the following table:

Cut-off per cent.	Position 1.	Position 2.	Position 3.	Position 4.
70	0·25	0·39	0·56	0·37
75	0·34	0·475	0·66	0·49
80	0·44	0·56	0·755	0·62
85	0·54	0·645	0·845	0·74
90	0·66	0·74	0·92	0·855
95	0·79	0·84	0·99	0·96

For estimating stresses in the shaft, etc., the maximum torque must also be known.

The combined torque due to two cranks at late cut-off reaches maximum values at four points in the revolution. Two of these points are when the cranks are both above the centre line and at 45°, and when both are below at 45°. The ratio of the torque to that due to one cylinder at full crank radius is then 1·414. Another maximum position is when crank A is at 135° from the inner and B at 45° from the outer dead centre, the ratio being then 1·214. The fourth position is with A at 135° from the outer and B at 45° from the inner dead centre, and this gives the largest ratio of all, namely 1·614.

If the cut-off is sufficiently early, the steam valve of one cylinder will close before the above positions are reached, and the steam pressure in that cylinder will have fallen due to partial expansion; the torque will thus be reduced somewhat as compared with late cut-off.

FLYWHEELS.

D = diameter, measured to centre of gravity of section of rim, in ft.

W_r = weight of rim, in lbs.

W_a = weight of arms and boss, in lbs.

W_e = 'Equivalent weight' at centre of rim section = approximately ($W_r + \frac{1}{3} W_a$)

N = R.P.M.

E = stored energy in ft. lbs.

$= 0\cdot0000425\ W_e D^2 N^2$

or $= 0\cdot00017 W_e R^2 N^2$, where R = radius to centre of rim.

For a specified limiting speed fluctuation, the equivalent weight of the flywheel depends on the irregularity of turning moment of the engine.

A = work done per revolution in ft. lbs. = area of turning moment diagram.

a = max. excess or deficit of area of turning moment curve over curve of resisting moment (which is usually taken as uniform).

k = coefficient of energy fluctuation $= \dfrac{a}{A}$.

NOTE.—Sometimes k is taken as excess energy ÷ work done per stroke instead of work per revolution, and some reference books therefore have values of k twice those given below.

K = flywheel coefficient = $\dfrac{\text{stored energy in flywheel}}{\text{work done per revolution}}$.

X = coefficient of speed fluctuation, or irregularity.

Then, if suffixes 1, 0, 2 refer to maximum, mean and minimum respectively,

$$X = \frac{N_1 - N_2}{N_0} = \frac{k}{2K} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$K = \frac{k}{2X} = \frac{a}{2AX} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Let Y = angular deviation from position of uniform turning,

„ C = number of complete turning moment cycles per revolution,

then $$Y_1 - Y_2 = 28 \cdot 65 \frac{X}{C} \text{ in geometrical degrees} \quad . \quad . \quad . \quad . \quad (3)$$

For engines driving electrical alternators,

let F = frequency of (electrical) cycles per sec.

p = number of poles in alternator field magnets,

then $$Y_1 - Y_2 = 14 \cdot 32 \frac{pX}{C} \text{ in electrical degrees} \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$= 1719 \frac{FX}{NC} \text{ in electrical degrees} \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The formulæ (3), (4), and (5) are based on sine curves for the turning moment, and are rarely strictly correct as regards the angular deviation $Y_1 - Y_2$, but formulæ (1) and (2) are always correct if the value of a is properly determined.

Considerable uncertainty arises as to the value of a in the case of multi-crank engines.

It is usual to specify that for alternators the angular displacement must not exceed 2½ electrical degrees. In that case, we obtain from equation (4) the relationship

$$\text{coefficient of irregularity } X = \frac{1}{5 \cdot 73} \frac{C}{p} \quad . \quad . \quad . \quad . \quad (6)$$

In the B.E.A.M.A. standardisation rules for electrical machinery, the rule is given

$$\text{cyclic speed irregularity} = \frac{k}{6p} \quad . \quad . \quad . \quad (7)$$

where

k = engine impulses per revolution ; p = number of poles.

This formula differs from (6) only in replacing 5·73 by 6, but, as already pointed out, these formulæ depend upon the form assumed for the turning moment curve, and variations are therefore not unlikely and do not indicate any error in principle.

Certain flywheel data are given in the three following tables. It will be understood that these are typical values, and that considerable variations occur in practice.

Typical Values of Energy Coefficients, k.

Engine.	Cranks.	Double or Single Acting.	Cycles per rev. C.	k	Crank Angles.
Steam engines :					
Simple	1	Double	2	0·12 –0·16	
Tandem	1	„	2	0·11 –0·14	
Cross-compound	2	„	4	0·06 –0·09	90
Triple	3	„	6	0·025–0·03	120
Uniflow	1	„	2	0·11 –0·13	
Gas engines, 4-cycle :					
Tandem, two cylinders	1	Double	2	0·13 –(0·2)	A
„ four „	2	„	4	0·06 –(0·1)	90 A
One cylinder	1	„	0·5		
One cylinder	1	Single	0·5	0·2 –2·4	
Two cylinders, tandem	1	„	1	0·9 –1·1	360
Two cylinders, twin	2	„	1	0·9 –1·0	360
Two cylinders, opposed	1	„	0·5	1·3 –1·5	180
Three cylinders	3	„	1·5	0·5 –0·6	120
Four cylinders	4	„	2	0·1 –0·13	180
Six cylinders (tandem) . . .	3	„	3	0·06 –0·08	120

$k = \dfrac{\text{maximum excess energy}}{\text{work done per revolution}}$. A indicates figures in brackets, adopted to cover misfiring.

TYPICAL CYCLIC IRREGULARITIES X AND FLYWHEEL COEFFICIENTS K.

Driving.	Connection.	Engine.	No. of Cranks.	X	K
Alternators in parallel . . .	Direct	Large	—	$\frac{1}{250}$	—
" "	"	High-speed	—	$\frac{1}{250}-\frac{1}{300}$	—
D.C. dynamos	"	Steam	—	$\frac{1}{100}-\frac{1}{125}$	—
"	"	Large gas	—	$\frac{1}{150}$	—
"	"	Small gas	—	$\frac{1}{80}-\frac{1}{125}$	—
"	"	Diesel	—	$\frac{1}{100}-\frac{1}{125}$	—
"	Ropes	Various	—	$\frac{1}{80}-\frac{1}{125}$	—
Best factory	"	Steam	1	$\frac{1}{155}$	12
" "	"	"	2	$\frac{1}{200}-\frac{1}{340}$	7–12
General factory	"	"	1	$\frac{1}{125}-\frac{1}{155}$	8–10
" "	"	"	2	$\frac{1}{170}-\frac{1}{250}$	6– 9
Pumping, reciprocating . . .	Direct	"	1	—	·15–·5
Pumping, usual	"	Triple vert.	3	—	3

VALUES OF FLYWHEEL COEFFICIENTS K FOR VARIOUS VALUES OF X AND *k*.

Engine.	Cranks	*k*	X $\frac{1}{75}$	$\frac{1}{100}$	$\frac{1}{125}$	$\frac{1}{150}$	$\frac{1}{200}$	$\frac{1}{250}$
Steam:								
Double acting	1	0·13	4·9	6·5	8·1	9·8	13·0	16·3
" "	2	0·07	2·7	3·5	4·4	5·3	7·0	8·8
" "	3	0·03	1·2	1·5	1·9	2·3	3·0	3·8
Gas: double acting tandem	1	(0·2)	7·5	10·0	12·5	15·0	20·0	25·0
" " " "	2	(0·1)	3·8	5·0	6·3	7·5	10·0	12·5
Gas: 4-cycle single-acting:								
One cylinder	1	2·3	86	115	144	173	230	288
Two cylinders tandem . .	1	1·0	37·5	50·0	62·5	75·0	100·0	125·0
Two cylinders opposed . .	1	1·4	52·5	70·0	87·5	105·0	140·0	175·0
Three cylinders	3	0·55	20·6	27·5	34·4	41·2	55·0	68·8
Four cylinders	4	0·13	4·9	6·5	8·1	9·8	13·0	16·3
Six cylinders	3	0·07	2·7	3·5	4·4	5·3	7·0	8·8

NOTE.—The values of *k* are based on the assumption of a uniform engine load, and do not apply accurately to engines driving pumps, blowers, or even electric alternators. (*Foster.*)

For engines driving alternators, the flywheel effect should be such that the displacement Y_1-Y_0 does not exceed 2·5 electrical degrees, nor should X as a rule exceed $\frac{1}{250}$. Where several units have to work in parallel the flywheel effect must also be such as to avoid resonance, i.e. the natural period of oscillation of the rotor about the position of uniform turning must not coincide with the period of any periodic impulses due to the engine. The determination of the oscillation period requires knowledge of the electrical characteristics of the alternators.

Alternator builders commonly state the flywheel quantity of their alternators in terms of GD^2, which is short for weight multiplied by the square of the diameter of gyration. If the units employed are kilogrammes and metres, the corresponding GD^2 in lbs. and feet can be obtained by multiplying by 23·7.

$$\text{Stored energy} = GD^2 \times \left(\frac{\pi N}{60}\right)^2 \times \frac{1}{2g} = \frac{GD^2 \times N^2}{23{,}500}.$$

If G, D are in lbs. and feet, the stored energy is in ft.-lbs.

The stored energy may be expressed in terms of any radius (or diameter) if the weight corresponding to that radius is stated. Thus, the flywheel effect is sometimes stated as T tons (or lbs.) at 1 ft. radius. The GD^2 is then T × 4 in tons (or lbs.) × feet². The actual diameter need not be known, which is a convenience of this method of expression.

The proportions of flywheels vary, but approximately for a typical engine flywheel the rim weight is about two-thirds of the total weight; the diameter of gyration is about 0·77 × diameter of rim; and the equivalent weight if assumed concentrated at the extreme diameter is about 0·60 × total weight. Thus, if the equivalent weight at the extreme diameter is calculated, the total weight is obtained by multiplying it by $\frac{1}{0·6}$, i.e. by 1·66.

Frequently the calculation is based upon the equivalent weight W_e reduced to the radius at the centre of gravity of the rim section. The actual rim weight is about $0 \cdot 9\ W_e$.

Example.—Tandem compound engine of 1,200 i.h.p. normal load at 94 r.p.m., driving an alternator of 50 cycles. The deviation must not exceed two electrical degrees on either side of the mean position.

From the tables we see that k is about 0·13, and $C = 2$.

In formula (5) we have

$$C = 2,\ Y_1 - Y_0 = 2,\ F = 50,\ N = 94$$

Hence $$X = 0{\cdot}00437 = \frac{1}{229}.$$

Inserting the value of X in formula (2) we obtain $K = 14{\cdot}9$.

The work done per revolution is

$$A = \frac{1{,}200 \times 33{,}000}{94} = 421{,}000 \text{ ft.-lbs.}$$

The stored energy in the flywheel is KA or 6,280,000 ft.-lbs.

If the flywheel is 18 ft. diam., the rim velocity is 88·6 ft. per sec., and the equivalent weight at this diameter is, therefore, 51,500 lbs. The actual flywheel weight should be about 66 per cent. in excess of this, say 86,000 lbs. This makes no allowance for the flywheel effect of the alternator rotor, which is usually considerable and need not be ignored.

The diameter measured to the centre of gravity of the rim section in the example may be about 16 ft., the rim being about 2 ft. in radial thickness.

$$W_e \text{ is then } 51{,}500 \times 18^2 \div 16^2 = 65{,}000 \text{ lbs.}$$

and the rim weight about $65{,}000 \times 0 \cdot 9 = 58{,}500$ lbs., which is a little more than two-thirds of the total weight.

SPEED OF FLYWHEELS.

The rim of a flywheel is subjected to stress on account of centrifugal force, and this stress is independent of the cross-section. For a particular material it depends only on the linear speed of the rim.

For cast-iron wheels the safe rim speed is customarily taken as 6,000 ft. per minute. For a given r.p.m. this fixes the wheel diameter.

Where higher speeds are desired with consequently lighter wheels for the same amount of stored energy, cast steel may be used, or the wheel may be made with discs of steel plate.

A steel plate wheel with wire-wound rim has been run at 15,000 ft. per minute.

GOVERNORS.

It is the function of the governor to adjust the quantity of steam to the load on the engine. The sensitiveness of the governor should depend on the nature of the load, but in all cases the governor should be as free from friction (which also applies to all the valve mechanism operated by the governor) as possible. Friction tends to coarse governing and hunting. On the other hand, a very sensitive governor, especially if of great power on an engine with insufficient flywheel, tends to produce hunting, which should be corrected by an oil dashpot on the governor mechanism. In governor design attention must be paid to the levers and connections so as to give a reasonably equal ratio of governor movement to speed variation at all loads.

The plain Watt fly-ball governor is rarely used, being deficient in power. When loaded with a central weight (Porter governor) it is satisfactory on typical factory engines, especially if carried on ball-bearings and knife edges throughout. The centre weight makes it somewhat sluggish in following changes of speed, and springs replace the weight in many modern governors. The springs need not be vertical.

If the governor is mounted on the crankshaft it may become bulky owing to its slow speed, but has some merits where eccentrics or valve rods have to be moved directly. Shaft governors, owing to their bulk, tend to be sluggish and to lag behind the engine when the speed changes. This inertia effect has been utilised in the inertia shaft governor, in which it is the falling behind of the governor weights due to inertia and not their outward movement under centrifugal force which operates the valve gear.

Relay Governors.

In relay governors the movement of the valve gear is effected by a motor, usually taking the form of a cylinder and piston operated by oil pressure. The movement of the piston is controlled by a small sensitive governor, usually of the centrifugal type. This governor has only to move the oil valve and can be relatively small, whilst the force exerted by the relay piston may be anything desired. The oil is supplied by a reciprocating or, more usually, a rotary pump. A simple

drum or toothed-gear pump is entirely satisfactory. Any pressure can be obtained, but 50 or 60 lbs. sq. in. is a convenient value. In calculating the pump capacity allow for the maximum reasonable movement of the relay piston, and for 25 to 50 per cent. slip at the pump according to the speed, size, and pressure of the pump. These governors are common on large turbines and growing in favour for large oil, gas, and steam engines. Unlike the ordinary centrifugal governor insufficient power shows itself chiefly by slow action and not very much by a large difference in the rising and falling speeds. In short, the relay governor is, under these conditions, slow but accurate.

Governors are best driven by gears, but chains may be used and even belts or ropes if the last two are in duplicate or triplicate to minimise trouble in case of one belt or rope breaking.

Governors are best arranged to knock off at over speed, but this function may be arranged for by a simple separate governor.

Governors are often termed 'expansion' or 'throttle' governors, but these expressions refer to the valve gear rather than the governor. Expansion control is the more economical in most cases, but more difficult to arrange for except in the case of engines having Corliss or drop valves and running at moderate speeds only.

SLIDE VALVE GEAR.

(*W. C. Popplewell, M.Sc., A.M.I.C.E.*)

The movements of an ordinary eccentric driven slide valve are shown by A, fig. 10, S and S_1 are ports leading from the steam chest to the back and front of the piston respectively; E is the port leading to exhaust. The valve itself is shown sliding on the port face. As shown by A, the valve is in mid-position; *ef* is the outside lap, and *gh* the inside lap. As the valve is moved upwards by the eccentric, it reaches a position B where *e* has come opposite *b*, and the steam port S_1 is opening to the steam chest; the amount by which the valve is open at the beginning of the piston stroke is the 'lead.' In position C the steam port is fully open; the valve then returns to D, thus shutting off the supply of steam to the cylinder. This point is 'cut off.' The movement continues until *h* reaches *c*, as at E, when communication is effected between S_1 and E. This point is called 'exhaust.' After opening S_1 fully to exhaust, the valve returns to F, where the exhaust is closed and the point 'compression' reached. This completes the cycle for one end of the valve.

The movements of slide and other eccentric driven valves are analysed by such diagrams as the 'Valve Ellipse' (II, fig. 9), 'Zeuner's Valve Diagram' (fig. 12), and several others designed for the same use. Of these the Valve Ellipse can be made to take account of obliquity of both connecting rod and valve rod, and so yield accurate results. Zeuner's Diagram makes no allowance for obliquity, and in consequence is only approximately accurate, but for purposes of design, where existing data are not available, the Zeuner is the more useful, as it enables the principal dimensions to be approximately fixed. In such a case final adjustments may be completed by using the Valve Ellipse.

Valve Ellipse.

To set this out, begin by drawing the crank circle as at I, fig. 9, and dividing it into a number of equal angles (in this case 8), the crank-pin points being 0, 1, 2, 3, 4, 5, 6, 7. Take the connecting-rod length 1. 1_1, and set it off successively from the crank-pin positions, marking off points on the stroke line 0_1, 1_1, 2_1, 3_1, 4_1. These are positions of the piston corresponding to 0, 1, 2, 3, 4, of the crank-pin, and are seen at II. A similar construction is followed out for the eccentric circle at III (fig. 11). This is turned through a right angle as compared with I, and the 'angle of advance' is marked $\theta°$. The distances of the valve from its mid-position, corresponding to the successive positions of the piston, are shown below the eccentric circle. These are set up as ordinates from the horizontal points on II, and a smooth curve drawn through the points so found. Thus from 1_1 on the line of piston travel at II is plotted an ordinate which represents the distance of the valve from its central position when the piston is at 1_1; this is the distance from ZZ to the line marked 1 at III. Next draw a horizontal line at distance l above 0_1, 4_1, in II, this is the 'lap' line; also a second line at a distance p beyond the lap line, which is the 'port opening.' Similarly below 0_1, 4_1, are two further lines at l_1, 'inside lap,' and p_1, 'exhaust opening.' The points where the ellipse cuts these lines represent the chief incidents in the cycle. They are B (admission), where the valve is just opening to steam; C (full port); D (cut-off); E (release); F (compression).

In the case of the back plate of a gear of the Meyer type, separate ellipses must be drawn for the edge of the plate, and the points where these cut the main ellipse noted.

ZEUNER DIAGRAM.

A few of the principal cases are given in what follows. The results are not strictly accurate, because they are vitiated by the angularity of the eccentric rod. But with reasonable proportions a design will be sufficiently near what the diagram shows. Fuller information may be found in Zeuner's treatise, or in the small book of Cowling Welch, who reduced Zeuner's methods to practice.

SLIDE VALVE GEAR.

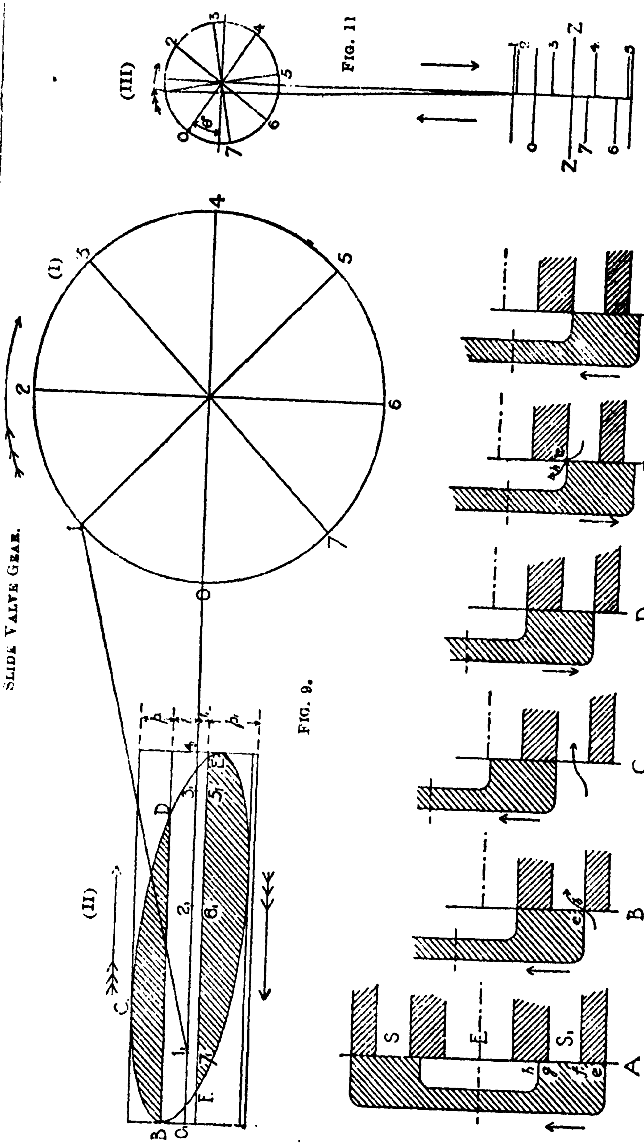


FIG. 9.

FIG. 10.

FIG. 11

Reversing gears which do not employ two eccentrics usually derive motion from some point on the connecting rod. Such are the Joy gear and others, for all of which special advantages are claimed under certain conditions, though as valve movements it is doubtful if any give better results than plain link gears. The newer motions often involve considerable sliding movement of a block in an inclined link, this link being pivoted so as to reverse its inclination, whereby the valve, driven by the block which slides up and down in the inclined link, has its movement reversed or varied according to the degree of inclination either way of the link. Such valve movements are best studied by the aid of models, with variable dimensions made from flat bars of cedar or baywood.

Simple Slide Valve.

Fig. 12 shows Zeuner's diagram of slide valve motions, simple case.

Draw NM and PR at right angles; NM = piston stroke.

OP = OR = valve travel each side of centre point; OA = crank when steam is admitted; OB = crank at cut-off.

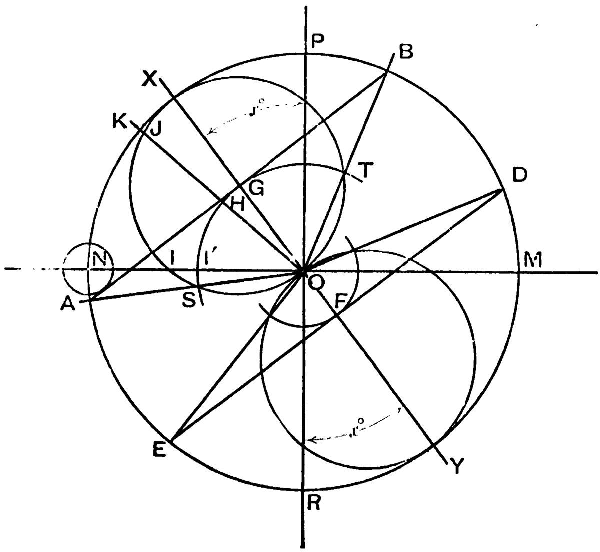


Fig. 12.

Bisect AB. Join OX, OY perpendicular to AB.

$x° + 90°$ = angular advance of eccentric; SGT = outside lap circle; cuts 'valve' circle at ST; OS = position of valve from centre when admitting steam; OT = position of valve from centre when cutting off steam; OF = inside lap; OD = crank when exhaust commences; OE = crank when exhaust closes.

The intercept of any radius from O, cut off by valve circles, shows distance of valve from its midstroke for that crank angle. Thus when crank is at OK valve has moved a distance OJ from its centre and steam port is open a width HJ.

DY = YE. AX = BX. Lead = small circle N = 2I.I.

Double Back-Plate Cut-off.

Figs. 13 and 14 show Zenner's expansion valve diagram with Meyer valve.

Draw AB, XY at right angles, OD at angular advance $\phi°$ = half travel of main valve; OR = crank at admission: ON = lap circle. Draw OP slightly in advance of latest opening of main valve as shown by point *t*. This gives a miminum cover *ts*. Join PD, and draw OE parallel to it and DE parallel to OP. Then OE = half travel and position of the cut-off eccentric which will cut off steam when the crank, OR, reaches the position OP. It need not cut off later than this, for then the main valve has already acted.

The radius Oa or any other vector of the cut-off valve circle shows the movement of the cut-off valve from its middle position, OE being the half throw of the cut-off eccentric, and $\theta°$ its angular advance. Any radius vector of the auxiliary circle OP will show the relative displacements of the main and cut-off valves from each other. At the crank position OP the cut-off plates are drawn close together. Draw the large circle APDM through P. Then for any position OV of

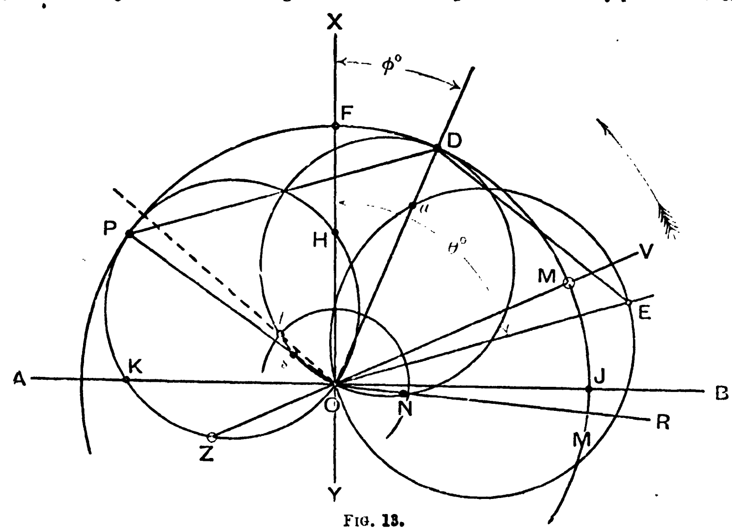


FIG. 13.

desired cut-off the length of the cut-off valve must be increased by separating the two cut-off plates of the right and left screw to the amount MZ. To cut off at OX of the crank would demand the movement FH, while to cut off at the dead point OB the movement of the plates would be JK. The length over the steam ports in the back of the main valve will be equal to the sum of the length of the cut-off plates plus twice OP. The edges of the cut-off plates when fully extended have a maximum distance from the main valve centre of JK (or MZ if OM is the earliest desired

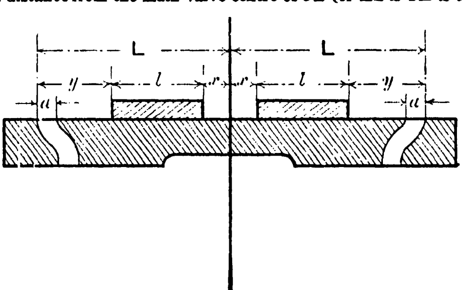


FIG. 14.

cut-off). But at no moment must the inside edges of the cut-off plates uncover the inside edges of the ports in the main valve. These ports have a width a. At the latest cut-off, viz at OP, the cut-off plates will be drawn together so that x = nil.

To find movement of cut-off plates from centre for any point of cut-off, as with crank at OV, the plates must be screwed out a distance x = MZ. If cut-off is at OF position of crank, then x = FH.

Taking the position of earliest cut-off at OV, to find the proportions L and l of valves which will be a maximum :—

Calling FE $= y = L - l - x$ for earliest cut-off, the maximum relative movement of the valve centres by the eccentrics is OP. Since at the position OV the valve centres are the opposite way removed from cut-off by the amount OZ and that OP is the maximum relative movement, we have $L - l = OP$ when $x =$nil and x must be $OP + OZ = MZ$ for earliest cut-off. When the crank reaches OP the cut-off valve is moved out already a distance $x = MZ$ and it is moved a further distance OP by the eccentric, so that the outer edge of the cut-off plate is a distance OP+OP+OZ from the middle of the main valve. And when so far the inner edge of the cut-off plate must not overrun the inner edge of the back port a. Therefore $L - a = 2OP + OZ$ ∴ $L = OP + OP + OZ + a$ and $L - l = OP$ ∴ $l = OP + OZ + a$. Since the main valve has cut-off at the position Ot there is safety lap of ts on the main port below, even if the inner edge of the cut-off plate then reaches the port a.

In the figure the main valve travels $4\frac{1}{2}$ ins. and the cut-off valve $5\frac{1}{2}$ ins. The cut-off plates are each moved out by screws at earliest cut-off $x = 3\frac{1}{4}$ ins., and by eccentric movement $2\frac{1}{4}$ ins. relative to main valve, or a total of $5\frac{1}{2}$ ins. The main valve has a length $L = 5\frac{3}{4} + a$ and the cut-off plate (or $L - OP = l$) a length of $3\frac{1}{2}$ ins. $+ a$.

If $a = 1$ in. then $L = 6\frac{3}{4}$ ins.; $l = 4\frac{1}{2}$; $x = 2\frac{1}{4}$ ins. maximum plus $3\frac{1}{4}$ screw-out.

If early cut-off is desired in the general working of the engine with quick action the auxiliary circle OP should be near to 90° from the cut-off position of the crank. Thus at the position OD the two valves are moving at high relative speed. It is usual to give the greater travel to the cut-off valve.

LOCOMOTIVE LINK MOTIONS.

These may be investigated also by the Zeuner method. A link motion is simply built up with two eccentrics, either of which may be made to bear fully and directly on the valve spindle and cause an engine to run forwards or backwards. This is accomplished by connecting the two eccentric rods by a link in which the end of the valve spindle is carried. At the extreme end position the spindle is wholly driven by one eccentric, while at intermediate positions the two eccentrics combine to give a gradually shorter movement to the valve.

There are three general link gears. 'Stephenson's,' in which the link is raised or lowered to give the desired position of the spindle block or link block; and the link is concave to the axle and has a radius equal to the length of the eccentric rods.

'Gooch's,' with link convex to the axle and not movable up and down; movement of the block being effected by a radius rod and the link having this radius of curvature.

'Allen's,' in which the link is straight and both link and radius rod are raised and lowered in opposite directions.

All three forms are made with open or crossed rods, the former giving greater lead of the valve near the middle position of the link block, and the latter giving less lead towards the middle position. In Gooch's motion the lead is of course always constant.

STEPHENSON LINK MOTION.

Open Rod Design.

In fig. 15, AE=link. BD=link block in forward and backward gear. OJ=virtual single centrics to give requisite travel to B or D in full gear. Join OB, cutting valve circle on OJ at K.

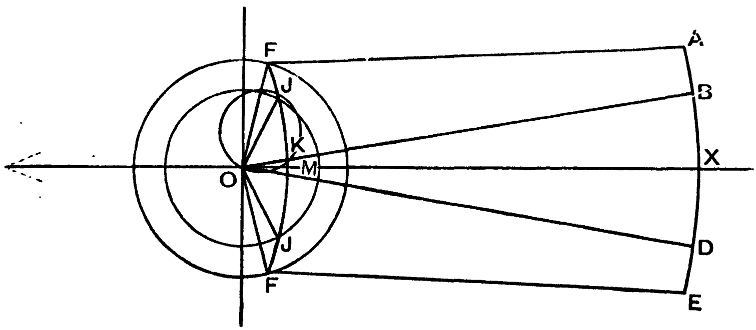


FIG. 15.

Join JM through K. Then the arc struck from c through JMJ will contain the ends of all the eccentric radii or valve circle diameters for all positions of the slider B or D. Dividing this arc

in the same ratio as B divides the link AE will give the ends of the valve circles for each point of the link. And the real eccentrics will have the position and half travel OF, OF just as the ends of the link lie beyond B and D. If the link end-pin comes opposite the extreme positions BD of the block then OJ become the real eccentrics in position and in half throw, i.e. XB : XA : : MJ : MF. Rods are said to be *open* when they are as FA, FE when the eccentric arms point towards the link. With open rods the valve lead increases with the expansion.

Crossed Rod Design.

In fig. 16, AE=link. BD=link block in full forward or backward gear. OJ=virtual single eccentric that would give requisite valve travel in full gear. Join OD, cutting upper valve circle on OJ at L. Join JN through L. N is on the line OX. Then the arc struck from O through JNJ contains the ends of all the valve circles for every position of the slide block, the arc JNJ being divided for each diameter in the same ratio as the slide block divides the link AE. Thus OG, OG become the actual eccentric arms, and NJ : NG : : XB : XA. Rods are said to be *crossed* when they are as GA, GE, the eccentric arms pointing towards the link. With

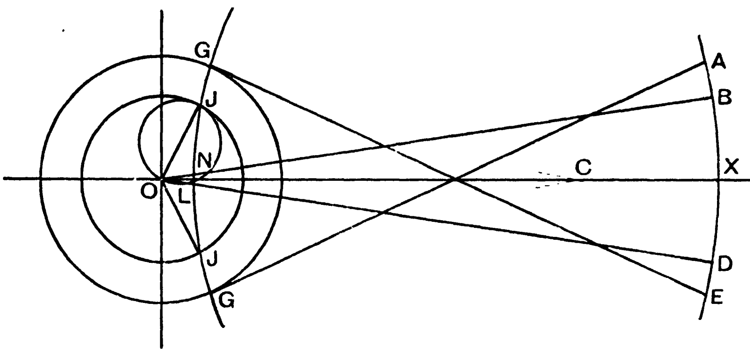


FIG. 16.

crossed rods the lead of the valve decreases with increased expansion. Specially note that OD cutting the valve circle at L is employed in this construction, whereas OB is the line used in the open-rod diagram.

GOOCH'S LINK MOTION.

In this motion, fig. 17, the 'curve' of valve circle centres is a straight line *aa* distant from O an amount $f = \frac{r}{2}\left(\sin \delta^\circ + \frac{c}{l} \cos \delta^\circ\right)$ for the open rod design and $\frac{r}{2}\left(\sin \delta^\circ - \frac{c}{l} \cos \delta^\circ\right)$ for the crossed rod design, where

r = eccentricity of eccentric or OE ;

m is the point from which the link is hung and is a fixed point ;

the versed sine of the link arc is $\frac{c^2}{2l_1}$;

m is distant from O a length $S = -\frac{c^2}{2l_1}$;

n is the point from which the radius rod is hung.

$On = l + l_1 - l_0 - \frac{c^2(l + l_1)}{2l\,l_1}$.

The point n should move in a parabola whose parameter $= 2l_1$ or a circular arc of radius l_1.

The co-ordinates of any point on the curve of centres are as per fig. 18, where

for open rods, $OB = \frac{r}{2}\left(\sin\delta^\circ + \frac{c}{l}\cos\delta^\circ\right)$;

for crossed rods, $OB = \frac{r}{2}\left(\sin\delta^\circ - \frac{c}{l}\cos\delta^\circ\right)$.

$BC = \frac{ru}{2c}\left(\cos\delta^\circ \pm \frac{c}{l}\cos\delta^\circ\right)$, according as the rods are open for − or crossed for + signs, where c = half length of the link and u = the variable position distance of the slider block from the middle of the link.

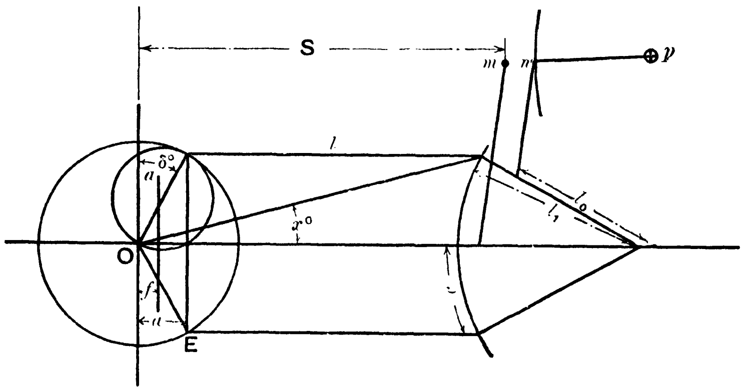


FIG. 17.

The lead is constant at all grades of expansion. The link is so hung that its middle point moves to and fro horizontally by the combined effect of the arc of swing and the cant due to slope of the link, the suspension link being pinned at the centre of the chord of the link.

The radius rod l_0 is so hung that its to-and-fro motion produces a minimum of sliding of the link block in the link. The extreme positions of n being known the actual arc in which it moves

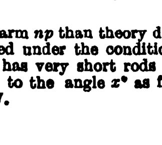


FIG. 18.

is arranged in practice with a shorter arm np than theory demands, the best compromise possible being accepted which can be arranged under the conditions. The Gooch gear, for any given distance between axle and cylinder, has very short rods compared with the Stephenson gear. Also the angle δ° in fig. 18 is equal to the angle x° as found in fig. 12, p. 96, for the simple slide valve, plus the angle δ° in fig. 17.

ALLEN'S STRAIGHT LINK MOTION.

$$\frac{b}{a} = \frac{l_0}{l}\left(1 + \sqrt{1 + \frac{l}{l_1}}\right);\quad n = 1 + \left(\frac{l_1}{l_0}\ \frac{b}{a}\right).$$

Curve of centres is a parabola whose vertex is distant from $O = \frac{r}{2}\left(\sin\delta + \frac{c}{l}\cos\delta\right)$ for open

rods, and $\frac{r}{2}\left(\sin\ \delta - \frac{c}{l}\cos\ \delta\right)$ for crossed rods, and is concave towards O for open rods, and convex for crossed rods. Parameter for both open and cross rods $= \frac{nlr}{2c\cos\delta}\cos\delta \mp \left(\frac{c(n-1)}{nl}\sin\delta\right)^2$ for which a circular arc may be substituted of one half the parameter.

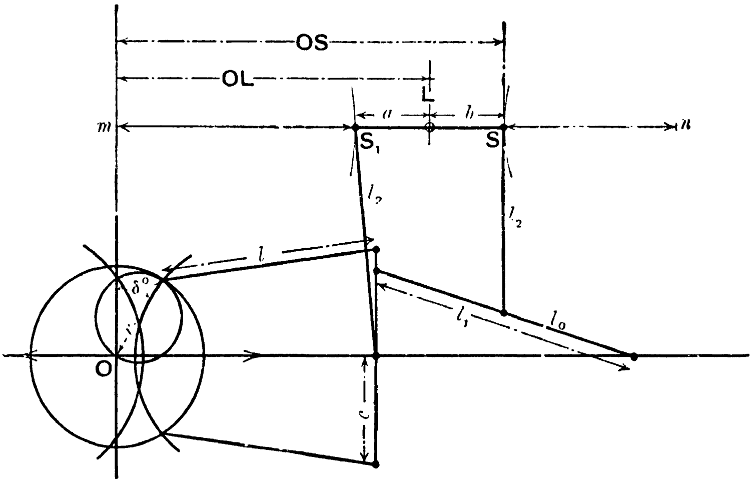


FIG. 19.

S and S_1 should move in parabolas of $2l_1$ and $2l$, or in arcs of l_1 and l radius.

As arranged in practice $a + b = (l_1 - l_0) +$ a little extra l_2; as long as possible.

$$Os_1 = \left(l - \frac{c^2}{2l}\right);\ OL = l + a - \frac{c^2}{2};\ ms_1 = l;\ ns = l_1;\ Os = l + l_1 - l_0 - \frac{c^2(l + l_1)}{2ll_1}.$$

Lead increases with expansion with open rods and decreases with crossed rods.

SECTION XXVII

PART IV

STEAM TURBINES.*

FLOW OF STEAM—DESIGN OF NOZZLES—GENERAL PRINCIPLES OF DESIGN—ESTIMATION OF STEAM CONSUMPTION—IMPULSE TURBINES—REACTION TURBINES—TURBINE LOSSES—RECENT DEVELOPMENTS IN STEAM CONDITIONS—TYPES OF TURBINES—TURBINE TEST CORRECTIONS—CENTRIFUGAL STRESSES IN DRUMS AND DISCS—CRITICAL SPEEDS—MATERIALS FOR TURBINE CONSTRUCTION—LUBRICATION—PROCEDURE IN STARTING UP LARGE TURBINES—B.S. SPECIFICATION FOR STEAM TURBINES—TURBINE SPEED REDUCTION GEARS—PARTICULARS OF REPRESENTATIVE MACHINES—NOTABLE INSTALLATIONS OF GEARED TURBINES.

(Contributed by Sir Henry Guy, D.Sc., F.R.S., Wh. Exh., M.I.C.E., M.I.Mech.E., and L. S. Robson, M.C., M.Sc.Tech., M.I.Mech.E.)

In this section the following symbols have been used in the indicated significance, except where otherwise stated :—

c = absolute velocity in ft. per sec.
g = constant of gravitation = 32·2 ft. per sec.^{2}
H.h. = heat drop in B.Th.U.
N = speed in r.p.m.
P = pressure in lbs. per sq. ft.
p = pressure in lbs. per sq. in.
t = temperature °F.
T = absolute temperature °F = $(460+t)$.
U = peripheral velocity in ft. per sec.
v = specific volume in cubic ft. per lb.
W = weight of steam in lbs. per sec.
w = relative velocity in ft. per sec.
λ = ratio of the specific heats of a gas.
ω = angular velocity in radians per sec.
J = Joule's Equivalent = 778 ft. lbs. per B.Th.U.

FLOW OF STEAM.

When steam flows through a simple orifice from a chamber (denoted by suffix 1) into a space (denoted by suffix 2) the velocity with which discharge takes place is always dependent on the steam conditions existing before the orifice, and sometimes dependent upon those existing on the discharge side.

If the pressure, P_1, in the chamber is maintained constant, while that beyond the orifice is gradually reduced from an initial value equal to P_1, the velocity and quantity of discharge gradually increase and depend on the pressure drop over the orifice, until a critical value is reached corresponding to a definite value of the ratio $\frac{P_2}{P_1}$.

* See also Marine Steam Turbines, Section XXIX.

Dr. Zeuner * has shown theoretically, and the result is borne out by experiment, that the critical drop is reached when

$$\frac{P_2}{P_1} = \left(\frac{2}{\lambda + 1}\right)^{\frac{\lambda}{\lambda - 1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

For saturated steam $\lambda = 1{\cdot}135$, and $\frac{P_2}{P_1} = 0{\cdot}577$.

For superheated steam $\lambda = 1{\cdot}3$, and $\frac{P_2}{P_1} = 0{\cdot}546$.

After this *Critical Pressure Drop* has been reached, the quantity and velocity of discharge remain constant for smaller values of P_2, the discharge being accompanied by strong acoustic vibrations.

Dr. de Laval was able to obtain—and used in his turbine—much higher velocities than the critical velocity, by adding a conically divergent nozzle to the orifice. In this way the vibrations are eliminated, and although the velocity of exit from the nozzle may be considerably greater than the critical velocity, the velocity at the minimum section or throat of the nozzle and the weight of steam discharged remain unchanged from the values which would exist in a simple orifice.

From this it will be apparent that there is a discontinuity in the phenomena accompanying discharge of steam through an orifice or nozzle, and that the equations existing between the various terms entering into the formulæ of steam flow must be considered in two groups.

(*a*) *When the Pressure Drop is less than the Critical Drop.*

i.e., when $P_2 > 0{\cdot}577\ P_1$, for saturated steam ; $P_2 > 0{\cdot}546\ P_1$, for superheated steam.

The theoretical velocity c_1 with which discharge takes place, can be calculated from the formula due to Wantzel and St. Venant,

$$c_1 = \sqrt{2g\frac{\lambda}{\lambda - 1}(P_1 v_1 - P_2 v_2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The actual velocity of discharge c_2 is less than c_1, on account of the friction and eddy losses in the orifice, and is given by

$$c_2 = \phi\, c_1,$$

$$\text{where } \phi = \text{velocity coefficient} = \frac{\text{Actual velocity of discharge}}{\text{Theoretical velocity of discharge}}.$$

The value of ϕ for various cases is given in Table I., page 108.

The weight of steam discharged per second is given by the relation

$$W = k\,S\frac{c_2}{v_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where,

v_2 = specific volume of the steam in the lower state, in cubic ft. per lb.
S = the sectional area in sq. ft. of the smallest section or throat of the orifice.
k = coefficient of discharge (see Table I.).

The expansion takes place with very great rapidity and for purposes of calculation it may be assumed to be adiabatic without serious error. The validity of this assumption has been questioned, but Dr. A. Loschge† has proved its approximate correctness by obtaining equal discharge through duplicate nozzles made of metal and of porcelain, and it need only be added that formulæ deduced from this basis agree very closely with experimental results.

For adiabatic expansion—

$$P_1 V_1^{\lambda} = P_2 V_2^{\lambda} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

from which, by combination with (2) and (3), we get

$$W = k\,S\sqrt{2g\frac{\lambda}{\lambda - 1}\frac{P_1}{v_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{\lambda}} - \left(\frac{P_2}{P_1}\right)^{\frac{\lambda+1}{\lambda}}\right]} \quad . \quad . \quad . \quad . \quad (5)$$

* *Technical Thermodynamics*, Dr. Zeuner.
† *Engineering*, October 17, 1913.

Alternative Method.

The following method of dealing with such problems, due to Dr. Zeuner, is often more simply manipulated than that already given, especially when used in conjunction with the Mollier or other entropy diagram.

If

H_1 = total heat of 1 lb. of the steam in the higher condition in B.Th.U.;

H_2 = total heat of 1 lb. of steam in the lower condition in B.Th.U.;

h_a = the heat drop available for adiabatic expansion = $H_1 - H_2$.

Then

$$e_1 = \sqrt{2g \cdot J \cdot h_a} \quad . \quad . \quad . \quad (6), \quad \text{and } e_2 = \phi \cdot e_1.$$

W can be calculated as before from (3); h_a can either be measured directly off a Mollier diagram or calculated from steam tables in the following manner :—

If $t°$ = degrees of superheat; L_1 = latent heat corresponding to P_1; T_1 = absolute temperature corresponding to P_1; h_1 = sensible heat corresponding to P_1; X_1 = dryness fraction of the steam in the higher state; T_2 = absolute temperature corresponding to P_2; h_2 = sensible heat corresponding to P_2; X_2 = dryness fraction of the steam in lower state; ϕ_1 = entropy of the liquid in the higher state; ϕ_2 = entropy of the liquid in the lower state;

then

$$X_2 = \frac{T_2}{L_2}\left[\frac{X_1 L_1}{T_1} + C_p \log_e \left(\frac{T_1 + t}{T_1}\right) + \phi_1 - \phi_2\right],$$

and

$$h_a = X_1 L_1 + C_p \cdot t \cdot + h_1 - X_2 L_2 - h_2.$$

In these formulæ the mean value of the specific heat of superheated steam at constant pressure C_p for the range of the superheat should be taken.

Nozzles which are designed for a smaller pressure drop than the critical are wholly convergent and must be sized according to the methods given in this section.

Simple orifices, the nozzles of Rateau or Zoelly turbines, and both the rotating and stationary blades of Parsons turbines, fall into this class.

(b) *When the Pressure Drop is Greater than the Critical Drop.*

i.e., when $P_2 < 0{\cdot}577P_1$, for saturated steam; $P_2 < 0{\cdot}546P_1$, for superheated steam.

In this case let the suffix m denote the conditions in the minimum section or 'throat' of the nozzle.

Then whatever the total pressure difference over the nozzle, that from P_1 to the throat pressure is always equal to the critical value, and the nozzle must be convergent to the throat.

By combining (1) with (2) and (4), the velocity in the throat is obtained.

$$e_t = \sqrt{g\lambda P_2 v_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7a$$

alternatively,

$$= \sqrt{2g \frac{\lambda}{\lambda + 1} P_1 v_1} \quad . \quad . \quad . \quad . \quad . \quad (7b)$$

if $\phi = 1{\cdot}0$.

(7) is identical with the expression for the velocity of sound in the conditions present in the throat, so that the velocity in the throat, or the critical velocity, is equal to sound velocity.

By substituting from (1) and (4) in (5)—

$$W = k S_m \sqrt{2g \frac{\lambda}{\lambda + 1}\left(\frac{P_1}{v_1}\right)\left(\frac{2}{\lambda + 1}\right)^{\frac{2}{\lambda - 1}}} \quad . \quad . \quad . \quad . \quad . \quad (8)$$

from which the discharge per second can be obtained, S_m being the area at the throat in square feet.

The nozzle from the throat to the mouth must be divergent, the area at the mouth being calculated from the relation

$$= S_m \cdot \frac{V_2}{V_m} \frac{c_m}{c_2} = S \left[\left(\frac{2}{\lambda+1}\right)^{\frac{1}{\lambda+1}} \left(\frac{P_1}{P_2}\right)^{\frac{1}{\lambda}} \sqrt{\frac{\frac{\lambda-1}{\lambda+1}}{1-\left(\frac{P_2}{P_1}\right)^{\frac{\lambda-1}{\lambda}}}} \right] \quad . \quad . \quad (9)$$

Expressions involving λ are strictly applicable only when the expansion takes place wholly in either the superheated or the saturated region, the corresponding value of λ being introduced.

The following expressions will be found to give approximately the same results as (9).

For Superheated Steam.

If $\quad P_1 = (1{\cdot}732 \text{ to } 20)\, P_2 \qquad S_2 = S_m\left(0{\cdot}7175 + 0{\cdot}163\, \frac{P_1}{P_2}\right)$

$\quad P_1 = (20 \text{ to } 100)\, P_2 \qquad S_2 = S_m\left(1{\cdot}5 + 0{\cdot}125\, \frac{P_1}{P_2}\right)$

For Saturated Steam.

If $\quad P_1 = (1{\cdot}832 \text{ to } 25)\, P_2 \qquad S_2 = S_m\left(0{\cdot}777 + 0{\cdot}1216\, \frac{P_1}{P_2}\right)$

$\quad P_1 = (25 \text{ to } 100)\, P_2 \qquad S_2 = S_m\left(1{\cdot}875 + 0{\cdot}0812\, \frac{P_1}{P_2}\right)$

Introducing the numerical value of λ (1·135 for saturated steam and 1·3 for superheated steam), and with $g = 32{\cdot}2$, formula (7) becomes—

For Saturated Steam.	*For Superheated Steam.*
$c_m = 70{\cdot}2\, \phi\, \sqrt{p_1 v_1}$	$c_m = 72{\cdot}38\, \phi\, \sqrt{p_1 v_1}$

In the same way formula (8) would produce two expressions for the discharge depending on the condition of the steam as regards superheat. Researches by Loschge* and Bendemann have however shown that the numerical coefficient of the formula for nozzles having a convergent entrance is the same both for saturated and superheated steam, and that this value is 2 per cent. higher than is obtained by inserting $\lambda = 1{\cdot}135$.

This is in accordance with the previously known fact that for saturated steam the discharge from such a nozzle exceeds the theoretical amount.

Just prior to the publication of these researches Mr. H. M. Martin* propounded the ingenious theory that with steam initially saturated an unstable condition exists in the throat, cooling taking place below the saturation temperature without condensation. The decrease in specific volume which results from this 'Under-Cooling' would account for the measured excess in discharge over that calculated on the assumption of adiabatic expansion.

Prof. Stodola has suggested that this under-cooled steam obeys the same law of expansion as superheated steam, so that the experimental fact that the discharge both for saturated and superheated steam is given by one formula is rational and in line with theory.

The formula for discharge deduced from Loschege's experiments is :—

$$W = 44{\cdot}11\, k\, S_m \sqrt{\frac{p_1}{v}}.$$

Fig. 1 shows the theoretical discharge of steam per second per square inch of throat area with critical pressure drop, and is calculated from Callendars' approximate formula :

$$W = 0{\cdot}3153 \sqrt{\frac{p^1}{v_1}}$$

* *Engineering*, October 17, 1913.

The following simple formulæ, the first due to Napier and the second to Rankine, are often useful when an error of a few per cent. is admissible.

For Dry Saturated Steam.

$$P_1 > \frac{5}{3}P_2,\ W = \frac{s \cdot p_1}{70}; \qquad P_1 < \frac{5}{3}P_2\ \ W = 0{\cdot}029\ s\ \sqrt{p_2(p_1 - p_2)}$$

where s = area of orifice in square inches;

Design of Nozzles.

The values of ϕ and k as determined by experiment are given in Table I. for orifices and nozzles of various kinds.

FLOW OF STEAM THROUGH NOZZLES.

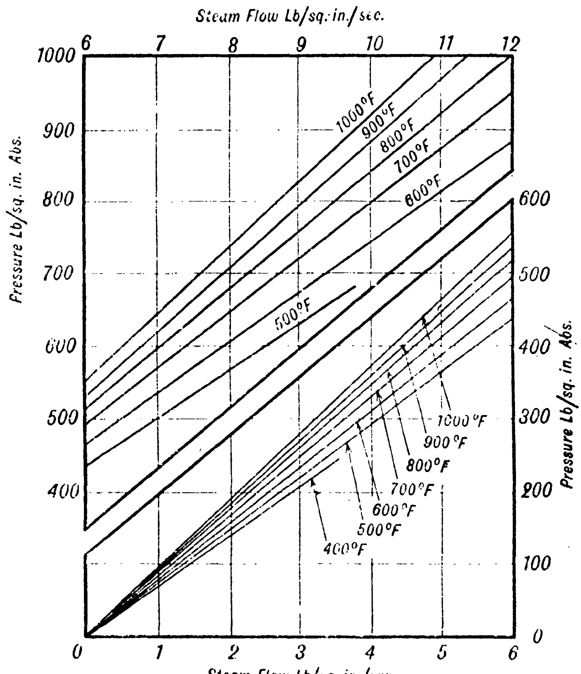


FIG. 1.

The coefficient of discharge k depends amongst other things upon the friction loss, the coefficient of contraction, and usually for a pressure drop below the critical, upon the ratio $\frac{P_2}{P_1}$.

Professor Rateau found that for nozzle C the value of k was slightly greater than unity when P_2 is less than $0{\cdot}5P_1$.

TABLE I.

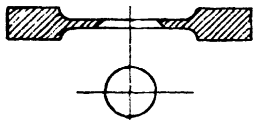
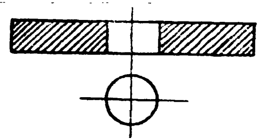
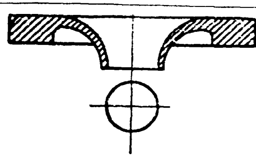
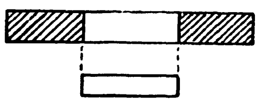
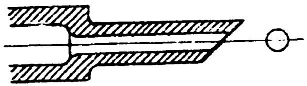
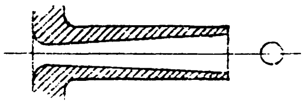
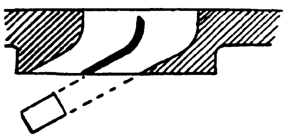
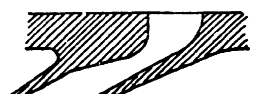
Type of Orifice.		Velocity Coefficient ϕ and Discharge Coefficient k.	Name of Experimenter.	Reference.
 Circular Hole in Thin Plate.	A	$\phi = 0{\cdot}86$	Rosenhain	Proc. Inst. C.E. vol. cxl.
		$\frac{P_2}{P_1} = 0{\cdot}95 \quad 0{\cdot}8 \quad 0{\cdot}6 \quad 0{\cdot}4$ $k = 0{\cdot}63 \quad 0{\cdot}68 \quad 0{\cdot}77 \quad 0{\cdot}83$	Rateau	'Flow of Steam' A. Rateau
 Cylindrical Hole.	B	$\frac{P_2}{P_1} = 0{\cdot}95 \quad 0{\cdot}8 \quad 0{\cdot}7 \quad 0{\cdot}6$ and smaller values $k = 0{\cdot}79 \quad 0{\cdot}83 \quad 0{\cdot}86 \quad 0{\cdot}88$	Gutermuth	Z. Ver. Deutsch. Ing. January 1904
 Straight Convergent Nozzle.	C	$c_e = 500 \quad 900 \quad 1{,}300$ ft. per sec. $\phi = 0{\cdot}936 \quad 0{\cdot}965 \quad 0{\cdot}993$ Saturated steam $\phi = 0{\cdot}954 \quad 0{\cdot}967 \quad 0{\cdot}98$ Superheated steam	Loschge	Z. Ver. Deutsch. Ing. January 11, 1913
		$\frac{P_2}{P_1} = 0{\cdot}95 \quad 0{\cdot}8 \quad 0{\cdot}7 \quad 0{\cdot}6$ smaller values $k = 0{\cdot}94 \quad 0{\cdot}967 \quad 0{\cdot}975 \quad 0{\cdot}99 \quad 1{\cdot}0$	Rateau	'Flow of Steam' A. Rateau
 Rectangular Orifice.	D	$\frac{P_2}{P_1} = 0{\cdot}95 \quad 0{\cdot}8 \quad 0{\cdot}7 \quad 0{\cdot}6$ smaller values $k = 0{\cdot}8 \quad 0{\cdot}88 \quad 0{\cdot}91 \quad 0{\cdot}93 \quad 0{\cdot}93$	Gutermuth	Z. Ver. Deutsch. Ing. January 1904

TABLE I.—*continued.*

Type of Orifice.		Velocity Coefficient ϕ and Discharge Coefficient k.	Name of Experimenter.	Reference.
 Convergent-cylindrical Nozzle. Circular Throat.	E	$c_1 =$ 200, 500, 1,000, 1,500 ft. per sec. $\phi =$ 0·923, 0·935, 0·95, 0·97	Briling	Z. Ver. Deutsch. Ing. February 1910
		$\frac{P_2}{P_1} =$ 0·95, 0·5 $k =$ 1·0, 1·0	Hirn ('Experiments on Air')	'Flow of Steam' A. Rateau
 Straight Convergent-divergent Nozzle. Circular Throat.	F	For a nozzle 3 ins. long $\phi = 0{\cdot}95$ " " 6 " " $\phi = 0{\cdot}915$	Sibley and Kemble	Journal Amer. Soc. M.E. November 1909
		$\frac{P_2}{P_1}$ Less than 0·89 $k = 0{\cdot}98$	Gutermuth	Z. Ver. Deutsch. Ing. January 1904
 Turbine Convergent Nozzle. Rectangular Throat.	G	$c_1 =$ 1,200, 1,400, 1,600 ft. per sec. $\phi =$ 0·86, 0·89, 0·91 $k = 1{\cdot}0$	Christlein	Z. für G. Turbinenwesen January 1912
 Turbine Divergent Nozzle. Circular or Rectangular Throat.	H	$c_1 =$ 1,700, 2,000, 2,200, 2,700, 2,300, 4,000 ft. per sec. $\phi =$ 0·92, 0·92, 0·93, 4·94, 0·95, 0·955	Christlein	Z. für G. Turbinenwesen January 1912
		$k = 1{\cdot}0$	Loschge	Z. Ver. Deutsch. Ing. January 1913.

In reference to nozzles G and H, which are of the types used in impulse turbines, it should be mentioned that although the value $k = 1 \cdot 0$ has been inserted in Table I., Dr. Ohristlein found that the actual discharge exceeds the theoretical by a few per cent. This effect had already been observed by Dr. Rosenhain for nozzles of form F having well-rounded entrances, and he concluded from his experiments that a loss in efficiency results, and was, therefore, led to recommend that the entrance should be only slightly rounded.

These facts are confirmed by Dr. A. Loschge's researches and explained by Mr. H. M. Martin's theory, which is referred to in the preceding paragraph.

The value of ϕ for nozzle G cannot be very satisfactorily determined from Dr. Ohristlein's experiments; for a given value of c_1, ϕ appears to increase with the pressure and to decrease with the superheat before the nozzle.

Unfortunately, the results of Dr. Rosenhain's experiments, like those of Messrs. Siberly and Kemble, are vitiated by the fact that the wetness of the steam during the experiments was not measured.

Nozzles of type B are useful for approximate measurements of the amount of steam flowing through a pipe line. It is only necessary to put a thick plate having a cylindrical hole in it between two of the pipe flanges. If the pressure is observed before the plate (P_1) and also in the cylindrical hole (P_2) by means of a hole drilled from the rim through the centre thickness of the plate, the flow can be calculated from formula (5).

Of course, the hole in the plate should be of such a diameter that the ratio $\frac{P_2}{P_1}$ is not less than $0 \cdot 6$.

Several steam meters operate on this principle.

The nozzles of turbines are usually of rectangular cross-section, and the steam in passing through them turns through an angle of about 70°, as indicated in figs. G and H, p. 109. It is important to bear this in mind when employing data relating to ϕ and k deduced from straight nozzles of circular cross-section—the type usually favoured by experimentalists.

For a given ratio of expansion the length of the nozzle depends on the angle of divergence. If this angle is too great the steam does not completely fill the cone, which results in considerable loss. If the angle is too small the length of the nozzles becomes very great, and many experiments indiczte that the friction loss depends on the length. An angle of divergence of 10° is very common.

The experiments of Dr. Rosenhain, and more recently of Messrs. Siberly and Kemble, show that there is no appreciable difference in efficiency for angles varying from 9° to 20°.

If the ratio $\frac{S_2}{S_m}$ is less than that theoretically required, expansion is completed beyond the nozzle. Rateau's experiments demonstrate that when this *under-expansion* is small, little loss results. On the other hand, if $\frac{S_2}{S_m}$ is greater than that theoretically required, the steam is expanded in the nozzle below the condition into which it discharges. Such *over-expansion* results in steam shock with great loss of efficiency.

When the divergent nozzles of a turbine are correctly designed for the normal load, over-expansion takes place at overloads, and under-expansion at partial loads, with nozzle cut-out governing.

The entrance of divergent nozzles should be smoothly rounded off, with a radius about equal to the throat diameter; larger radii are unnecessary and should be avoided.

Since the efficiency of a nozzle depends upon the smoothness of the surfaces with which the steam is in contact, the material of which it is constructed should be capable of withstanding erosion and oxidisation—nickel steel or a bronze highly polished is usually employed for the divergent nozzles of De Laval and Curtis turbines where high steam velocities are encountered, while cast in mild steel plates have given quite satisfactory results with the large nozzle areas and moderate velocities used in the convergent nozzles of Rateau and Zoelly turbines.

The state of knowledge on the efficiency of steam nozzles has been recently advanced by the researches carried out for the Nozzle Research Committee of the Institution of Mechanical Engineers by Dr. Stoney and Dr. Telford Petrie.

The apparatus used is of the impact plate type, largely designed by Mr. H. M. Martin. The first two reports dealt with tests taken on 1-in. wide reaction blading having a nominal outlet angle of 20°, and on impulse nozzles having an outlet angle of 20° with guide blades $0 \cdot 04$ in. thick.

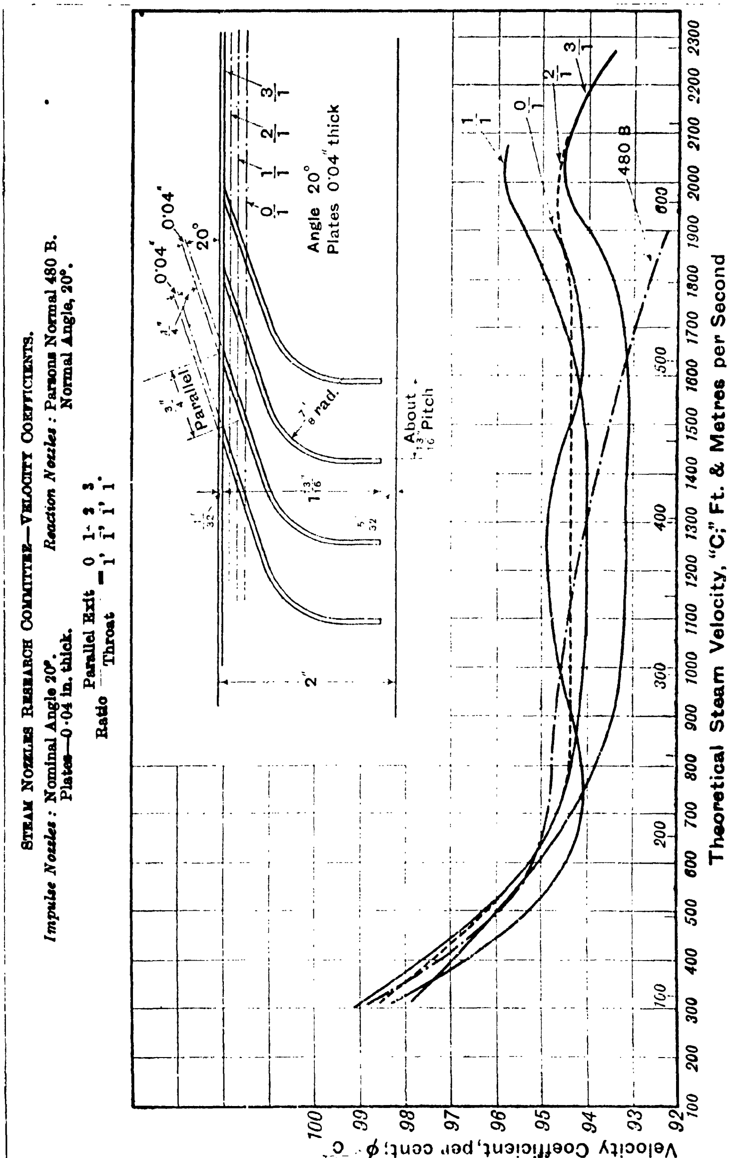
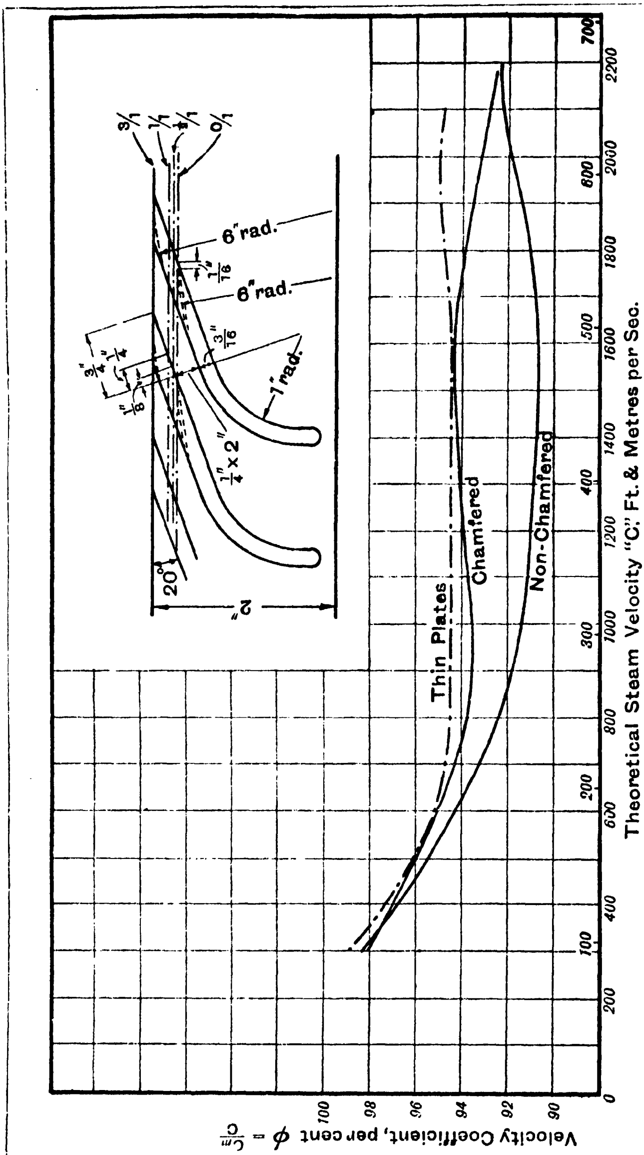

STEAM NOZZLES RESEARCH COMMITTEE—VELOCITY COEFFICIENTS.
Impulse Nozzles : Nominal Angle 20°.
Plates—0·04 in. thick.
Reaction Nozzles : Parsons Normal 480 B.
Normal Angle, 20°.
Ratio Parallel Exit / Throat = 0/1, 1/1, 2/1, 3/1
Angle 20°
Plates 0·04" thick
Parallel
About 1 3/16" Pitch
7/8" rad.
480 B
Velocity Coefficient, per cent; φ
Theoretical Steam Velocity, "C" Ft. & Metres per Second

FIG. 2.



Thin Plates
Chamfered
Non-Chamfered
6" rad.
6" rad.
1" rad.
20°
2"
Theoretical Steam Velocity "C" Ft. & Metres per Sec.
Velocity Coefficient, per cent
0
200
400
600
800
1000
1200
1400
1600
1800
2000
2200
100
200
300
400
500
600
700
100
98
96
94
92
90

FIG. 3.

The mean velocity coefficient (ϕ) values for the impulse nozzles obtained on those tests are shown on fig. 2, p. 111, for nozzles with throats of various proportions The corresponding curves for the reaction blading are shown by the dotted curve.

The notable feature of these tests is that they were carried out as large-scale experiments on commercial blades and nozzles.

The character of the curves, particularly for the impulse nozzles showing a rising value at low velocity, is particularly notable, as the region was not previously explored in published researches.

The third report of the Nozzle Research Committee of the Institution of Mechanical Engineers contains the results obtained with thick plate impulse nozzles summarised in diagram, fig. 3.

It will be observed that although the efficiency of the thick plate nozzles is materially lower than that of thin plate nozzles, the efficiency of the former can be improved to that of the latter by suitably chamfering the outlet.

In this report the experimental results on the efflux angle of the discharged steam suggest that its value is equal to $\sin^{-1}\frac{o}{p}$, and not to the geometrical angle (see fig. 4, and table II).

Experiments were also carried out on straight nozzles well radiused at the inlet and having ong parallel throats equal to about three diameters in length.

The efficiency of this nozzle was very high, giving a velocity coefficient θ of about 0·98 for 500 ft./sec., 0·97 for 300 ft./sec., and 0·98 for 1,700 ft./secs.

The experiments also lead to the conclusion that the efficiency of a nozzle is decreased as the superheat is increased.

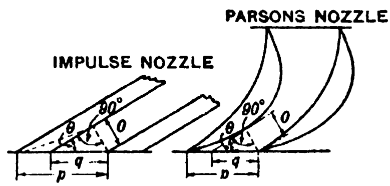


FIG. 4.—Efflux Angles.

TABLE II.

—	$\frac{o}{q}$	θ Degrees.	$\frac{o}{p}$	δ Degrees.	Angle of Exit. Degrees.	Difference Degrees.
20° thin plate	0·34	20	0·308	18·0	17·0	− 1·0
20° thick plate	0·34	20	0·190	11·0	14·0	+ 3·0
Ditto chamfered	0·34	20	0·190	11·0	10·5	− 0·5
12° thin plate	0·21	12	0·178	10·3	10·5	+ 0·2
12° thick plate	0·21	12	0·121	7·0	6·0	− 1·0
1 in. Parsons	0·61	37	0·350	20·5	18·5	− 2·0
¾ in. Parsons	0·61	37	0·350	20·5	18·5	− 2·0

GENERAL PRINCIPLES OF STEAM TURBINE DESIGN.

(1) *Preliminary Estimate of Efficiencies and Steam Consumption.*

This will be done very largely as a result of test results on similar machines and careful estimates of the effects of any new features in the turbine under consideration. A preliminary estimate may be made from the notes given on pp. 114 *et seq.*

(2) *Last Blade Dimensions.*

In estimating the steam consumption of the machine some allowance must be made for the throw away or leaving loss at the last row of blades and the exhaust loss. For a given permissible leaving loss, the area through the last row of blades will be fixed by the specific volume at the exhaust pressure specified and the total steam quantity passing to the condenser. As the relation between last blade height and mean diameter tends to a constant ratio (usually fixed by the difficulty of obtaining suitable steam passages through long blades at the root and the tip), the output of the machine will usually determine the length and mean diameter of the last blade. Similarly maximum output at any speed of rotation will be limited by the mechanical considerations involved in providing sufficient leaving area (depending on blade height and mean diameter), and the use of such devices as double flow or Baumann exhausts.

(3) *Number of Stages and Mean Diameters.*

Maximum efficiency for any type of blading can only be obtained by the employment of suitable ratios between steam speeds and peripheral speed of blades (see curves, p. 123). Small auxiliary turbines of moderate efficiency utilise the whole of the pressure drop available in a single stage running at a high speed of rotation and geared down to the speed required by the driven element. In larger machines where moderate speeds of rotation are employed, the heat drop is usually divided into several stages. Where efficiency is of the highest importance the heat drop available will usually be large, and the provision of a correspondingly large number of single row impulse or reaction stages is essential. The area through the early stages will be small, and as there are practical lower limits to the height of blading of accurate manufacture, the h.p. blade diameter will also be small, and the number of stages correspondingly increased (the heat drop capacity of any given stage depending on the product of speed rotation and mean diameter). With high stop-valve pressures, therefore, and low vacua, considerations of critical speed of shafts usually demand two or more cylinders, an arrangement which has many advantages with high inlet pressures and temperatures.*

Where efficiencies are of less importance, or questions of space or first cost demand it, a shorter, less expensive machine may be obtained by utilising a two-row impulse wheel, or a single-row impulse wheel of large diameter in the first stage, with partial admission. This arrangement has the advantage of reducing the pressure and temperature in the cylinder of the turbine, and with partial admission cut out governing may be employed.

(4) *Preliminary Layout.*

These particulars being decided, a preliminary layout of the turbine may now be made. Based on previous experience the general dimensions of the rotating elements may be estimated and the shaft critical speeds approximately determined, along with sizes of bearings, thrust blocks, couplings and pedestals, cylinder dimensions and materials, and so on. As this preliminary work will form the basis of, and be to some extent modified by, more detailed calculations, it is clear that a good deal of judgment is necessary in this stage of the design.

(5) *Determination of Blade Areas.*

From these preliminary designs a more detailed calculation may now be made of the areas, heights and angles of the blading, and of the steam conditions in the various stages of the machine.

For this purpose accurate steam charts or tables will be required, and the steam condition line can be approximated by provisional estimates of the internal efficiency of the various stages of the turbine. The final state point at the exhaust of the machine, depending on the internal efficiency of the whole turbine, may be estimated with some accuracy from a consideration of tests on existing machines, together with such data as are available as to the mechanical losses involved.

(6) *Mechanical Considerations.*

The stage pressures being determined, detailed calculations may now be made of the stresses and deflections of diaphragms. The bending stresses in moving blades, together with centrifugal stresses and angles required, will dictate dimensions of moving blades, from which the stresses in wheels may be calculated. The estimation of natural frequencies of vibration of moving blades and wheels can usually be made from previous tests, and the accuracy with which this may be done will depend on the degree of variation from blades and wheels already made and tested. Where this is necessary, the final tuning should be done by tests in the actual blades and wheels.

(7) *Efficiency Calculation.*

A closer approximation may now be made to the efficiency and steam consumption of the machine, by the detailed calculation of the various losses (see p. 1505 *et seq.*). Here, again, experience is required as the separate losses in a steam turbine are not readily measurable, and the closeness with which the calculated overall efficiency agrees with the actual efficiency eventually obtained in the turbine on test will depend on the judicious estimate of the many small individual losses involved.

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* See Guy, 'Tendencies in Steam Turbine Development,' *Proc. Inst. Mech. Eng.* 1929

ESTIMATION OF STEAM CONSUMPTION.

The 'Overall Efficiency' of a turbine is obtained by comparing the work actually done per lb. of steam, with that which would be performed by an ideal engine working on the Rankine cycle and having the same initial and final steam conditions as the turbine.

This latter amount of work is equal to the mechanical value of the change in total heat contents (h_a B.Th.U.) which takes place during adiabatic expansion from the steam conditions before the governor valve, to those obtaining in the turbine exhaust.

h_a is referred to as the 'adiabatic heat drop' or the 'available heat drop,' and its value for the steam conditions usual in practice is given in Table IV., p. 118.

TABLE III.—ADIABATIC HEAT DROP IN B.TH.U./LB. DRY SATURATED STEAM.

Vacuum 30 Ins. Barm.	Pressure Lbs. per Sq. In. Absolute. 14	15	16	17	18	19
27·0	144·41	148·90	153·11	157·08	160·82	164·35
27·25	149·33	153·79	157·99	161·95	165·68	169·19
27·5	154·67	159·12	163·30	167·24	170·96	174·46
27·75	160·51	164·94	169·10	173·03	176·73	180·21
28·0	166·96	171·37	175·51	179·42	183·10	186·57
28·25	174·20	178·58	182·70	186·59	190·25	193·70
28·5	182·45	186·81	190·90	194·76	198·40	201·83
28·75	191·97	196·30	200·36	204·19	207·81	211·22
29·0	203·51	207·80	211·82	215·62	219·21	222·59
29·25	217·92	222·15	226·17	229·95	233·44	236·73
29·50	237·96	242·10	246·07	249·79	253·23	256·48

If

d = steam consumption in lbs. per S.H.P. hour; η_T = overall efficiency of the turbine,

then

$$\eta_T = \frac{2{,}544}{d.\,h_a} \times 100 \text{ in per cent.}$$

If

d_1 = steam consumption in lbs. per kWh.; η_G = generator efficiency in per cent.,

then

$$\eta_T = \frac{3{,}412}{\eta_G d_1.h_a} \times 100 \text{ in per cent.}$$

The total steam consumption of the turbine is given by

$$W = \frac{d\,(\text{S.H.P.})}{3{,}600} \text{ lbs. per sec., or, } W = \frac{d_1\,(\text{kW.})}{3{,}600} \text{ lbs. per sec.}$$

While the efficiency calculated with h_a as already defined is the true index of the performance of a turbine when running, it is sometimes the practice—and it is one which has a special value to turbine designers—to calculate the efficiency by taking the heat drop h'_a as that available between the conditions existing after the governor valve, that is, before the nozzles, and the exhaust of the turbine.

This is equivalent to eliminating the consideration of the pressure drop through the governor valve, which is rightly regarded as not being an inherent loss to any given turbine.

Suppose, for example, that two turbines are sold as 1,000 kW. units to operate with certain steam conditions, and that one is actually capable of developing a load of 1,100 kW., while the second can only pull a load of 1,000 kW.

If these two turbines are tested on a load of 1,000 kW.. the latter will show the better steam consumption, because with the former the steam will be throttled at the governor to the greater extent. Of course, the first machine is in every way the better from the customer's point of view, provided the excess of power is not too great to be carried by the generator.

For estimating purposes steam consumptions may be obtained by means of figs. 5, 6, and 7. Fig. 5 shows turbine efficiencies at economical load for various sizes of machines with steam at 200 lbs./sq. g. 200° F. superheat, and with a vacuum at the exhaust flange of 29 ins. For other

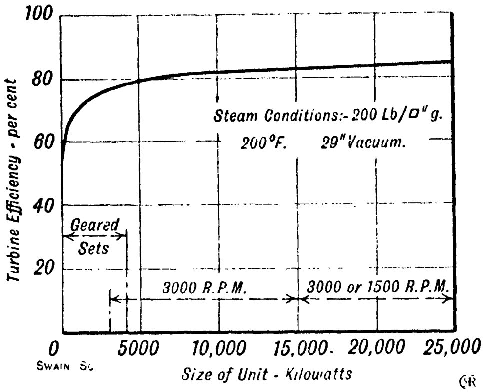


FIG. 5. TURBINE EFFICIENCY FOR H.P. TURBINES.

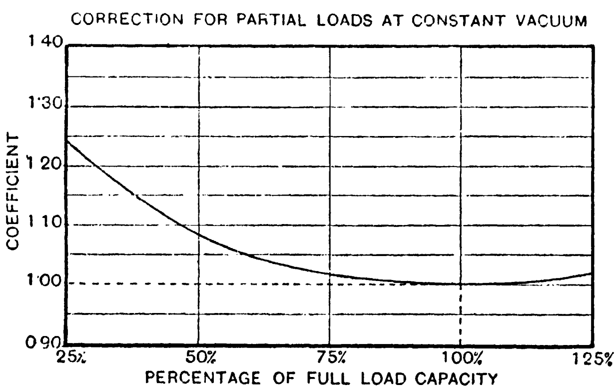


FIG. 6.

conditions the efficiency at economic load may be estimated by using the efficiency correction curves given in fig. 7.* When the economic load consumption is known, consumptions at other loads may be estimated from fig. 6.

* See Baumann, 'Some Recent Developments in Large Steam Turbine Practice,' *Proc. Inst. Elec. Eng.*, June 1921.

Efficiency Corrections for Turbines Designed for Various Conditions.

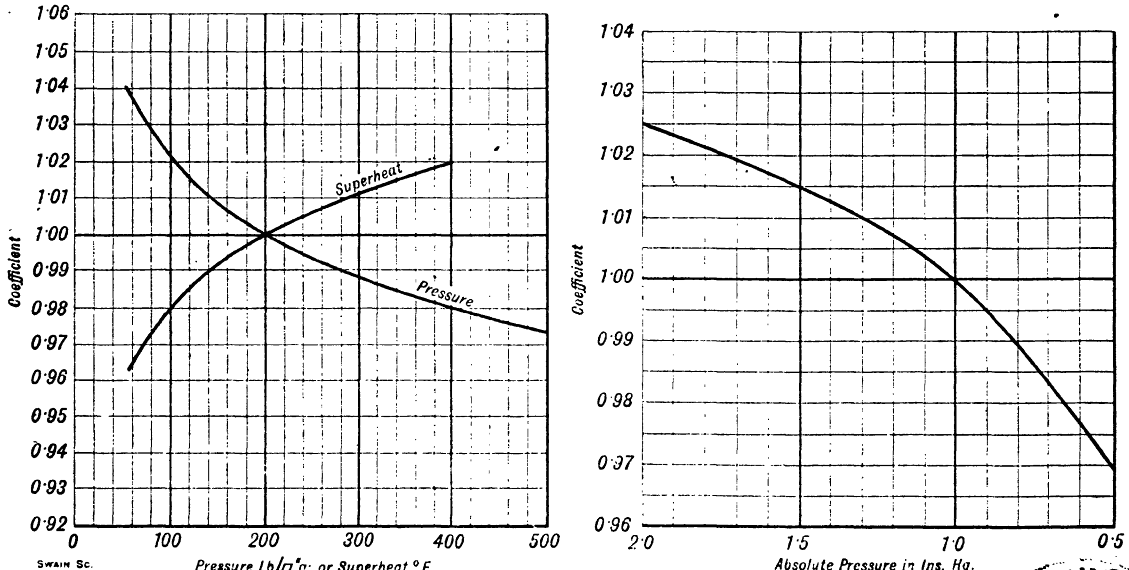


Fig. 7.

Standard Steam Conditions: 200 lbs. per sq. in. g. 200° F. 29 in. Vac.

For other steam conditions, multiply the efficiencies obtained from fig. 5 by the coefficients obtained from the curves.

TABLE IV.—ADIABATIC HEAT DROP IN B.TH.U./LB.

Vacuum. 30 ins. Barm.	Super-heat. °F.	Pressure Lbs. per Sq. In. Gauge.												
		2000	1500	1200	1100	1000	900	800	700	600	550	500	450	400
27·5	0	463·5	449·6	438·0	433·2	427·9	422·1	415·5	407·6	398·6	393·4	387·8	381·4	374·5
	100	504·7	487·1	473·0	467·3	461·2	454·2	446·5	437·6	427·2	421·2	414·9	407·7	399·9
	200	543·2	523·0	506·8	500·6	493·6	485·9	477·3	467·6	456·2	449·7	442·9	434·9	426·5
	300	580·4	557·8	540·2	533·3	525·9	517·5	508·2	497·8	485·7	478·7	471·4	463·1	454·1
28·0	0	472·8	459·1	447·6	442·9	437·7	431·9	425·4	417·6	408·6	403·5	398·0	391·7	384·8
	100	514·5	497·1	483·1	477·5	471·5	464·5	456·9	448·1	437·7	431·9	425·6	418·4	410·7
	200	553·5	533·5	517·4	511·2	504·3	496·7	488·1	478·5	467·2	460·8	454·0	446·1	437·7
	300	591·0	568·6	551·1	544·3	537·0	528·6	519·3	509·0	497·1	490·2	482·9	474·7	465·7
28·5	0	484·5	471·0	459·8	455·2	449·9	444·3	437·9	430·2	421·3	416·2	410·7	404·5	397·6
	100	526·8	509·7	495·9	490·4	484·3	477·5	470·0	461·2	451·1	445·3	439·0	431·9	424·2
	200	566·3	546·5	530·7	524·5	517·7	510·1	501·7	492·1	480·9	474·7	467·9	460·1	451·7
	300	604·6	582·2	564·8	558·1	550·8	542·6	533·5	523·2	511·3	504·5	497·2	489·0	480·2
28·75	0	491·7	478·4	467·3	462·7	457·5	451·9	445·5	438·0	429·0	424·0	418·6	412·4	405·6
	100	534·3	517·4	503·8	498·3	492·2	485·5	478·1	469·3	459·2	453·5	447·3	440·2	432·6
	200	574·2	554·6	538·8	532·7	526·0	518·4	510·1	500·5	489·4	483·2	476·5	478·7	460·4
	300	612·8	590·6	573·2	566·6	559·1	551·2	542·2	531·9	520·0	513·3	506·0	497·9	489·2
29·0	0	500·4	487·3	476·3	471·8	466·6	461·1	454·8	447·3	438·5	433·5	428·2	422·0	415·4
	100	543·7	526·8	513·3	507·9	501·9	495·2	487·8	479·1	469·1	463·4	457·3	450·2	442·7
	200	583·9	564·4	548·9	542·8	536·0	528·5	520·2	510·8	499·7	493·5	486·9	479·2	470·9
	300	622·6	600·7	583·5	576·8	569·7	561·6	552·5	542·4	530·7	524·5	516·8	508·7	500·1
29·25	0	511·5	498·5	487·8	483·3	478·3	472·8	466·5	459·1	450·5	445·6	440·3	434·2	427·5
	100	555·4	538·7	525·4	520·0	514·1	507·5	500·1	491·6	481·7	476·0	469·9	463·0	455·6
	200	596·1	576·8	561·4	555·4	548·7	541·3	533·1	523·7	512·8	506·6	500·1	492·4	484·4
	300	635·2	613·5	596·5	589·9	582·8	574·8	565·8	555·8	544·2	537·5	530·4	522·4	513·7
29·5	0	526·9	514·2	503·7	499·3	494·3	488·9	482·9	475·6	467·0	462·3	457·0	451·0	444·4
	100	571·6	555·2	542·0	536·7	531·0	524·5	517·1	508·8	499·0	493·4	487·4	480·6	473·2
	200	613·0	593·8	578·7	572·7	566·1	558·9	550·7	541·5	530·7	524·6	518·1	510·5	502·5
	300	652·5	631·2	614·3	607·8	600·9	592·8	584·0	574·1	562·6	556·0	549·0	541·1	532·5

TABLE IV.—*continued.*

Vacuum. 30 ins. Barm.	Super-heat. °F.	Pressure Lbs. per Sq. In. Gauge. 350	300	275	250	220	200	180	160	140	120	100	80	60
27·5	0	366·3	357·0	351·8	346·0	338·3	332·6	326·3	319·3	311·5	302·5	291·9	279·5	263·9
	100	390·9	380·6	374·8	368·5	360·1	353·9	347·0	339·4	330·9	321·2	310·0	296·6	280·0
	200	416·8	405·7	399·5	392·8	383·8	377·1	369·9	361·8	352·8	342·6	330·8	316·7	299·4
	300	443·9	432·2	425·6	418·6	409·2	402·2	394·6	386·2	376·8	366·2	353·9	339·3	321·3
28·0	0	376·7	367·5	362·3	356·5	349·0	343·3	337·1	330·1	322·4	313·4	303·1	290·7	275·3
	100	401·7	391·5	385·8	379·6	371·2	365·1	358·3	350·7	342·3	332·7	321·5	308·3	291·9
	200	428·1	417·1	410·9	404·2	395·3	388·7	381·5	373·5	364·6	354·5	342·8	328·8	311·6
	300	455·5	443·9	437·4	430·4	421·1	414·2	406·6	398·3	389·0	378·4	366·2	351·7	334·0
28·5	0	389·8	380·7	375·5	369·9	362·4	356·7	350·5	343·7	356·0	327·2	317·0	304·7	289·5
	100	415·4	405·3	399·6	393·5	385·2	379·1	372·4	365·0	356·6	347·1	336·1	323·0	306·8
	200	442·2	431·4	425·2	418·7	409·9	403·4	396·2	388·3	379·5	369·4	357·9	344·0	327·0
	300	470·2	458·7	452·2	445·3	436·0	429·2	421·8	413·5	404·3	393·8	381·7	367·4	349·8
28·75	0	397·8	388·8	383·6	378·1	370·6	365·1	358·9	352·1	344·5	335·8	325·6	313·4	298·3
	100	423·9	413·8	408·1	402·0	393·9	387·8	381·1	373·7	365·4	356·0	345·0	332·1	315·9
	200	451·0	440·2	434·1	427·6	418·8	412·3	405·2	397·4	388·6	378·7	367·1	353·4	336·6
	300	479·2	467·8	461·3	454·5	445·3	438·5	431·1	422·9	413·7	403·3	391·3	377·1	359·6
29·0	0	407·6	398·7	393·6	388·1	380·6	375·1	369·1	362·3	354·8	346·1	336·0	324·0	309·0
	100	434·1	424·2	418·6	412·5	404·4	398·3	391·7	384·4	376·1	366·8	355·9	343·0	327·0
	200	461·6	450·9	444·8	438·3	429·7	423·2	416·2	408·4	399·7	389·8	378·4	364·8	348·1
	300	490·1	478·8	472·4	465·6	456·5	449·7	442·4	434·2	425·1	414·8	402·9	388·8	371·5
29·25	0	419·9	411·2	406·1	400·6	393·3	387·8	382·0	375·3	367·7	359·1	349·1	337·2	322·4
	100	447·1	437·2	431·7	425·7	417·7	411·8	405·2	397·9	389·7	380·5	359·6	356·8	340·9
	200	475·0	464·4	458·4	452·0	443·4	437·1	430·0	422·3	413·8	403·9	392·6	379·2	362·5
	300	504·0	492·6	486·4	479·7	470·7	464·0	456·6	448·6	439·6	429·5	417·6	403·6	386·3
29·5	0	436·8	428·2	423·1	417·8	410·6	405·3	399·5	392·9	385·3	376·9	367·1	355·3	340·8
	100	464·7	455·1	449·6	443·6	435·7	429·8	423·4	416·2	408·2	399·1	388·3	375·7	360·1
	200	493·4	482·9	477·0	470·6	462·1	455·3	449·0	441·3	432·8	423·1	411·9	398·7	382·3
	300	522·9	511·7	505·5	498·8	489·8	483·2	476·0	468·2	459·3	449·1	437·5	423·7	406·9

The figures in the above tables have been abstracted from 'Heat Drop Tables' and 'Extended Heat Drop Tables' prepared by Prof. H. L. Callendar, F.R.S., and published by the British Electrical and Allied Industries Research Association.

IMPULSE TURBINES.

A pure impulse turbine is one in which the available heat energy of the steam is completely converted into kinetic energy in stationary nozzles.

The steam jet thus produced performs mechanical work by impinging on a rotating row of blades or successive rows of rotating and guide-blades placed immediately before the nozzles.

An impulse turbine in which the whole pressure drop available is utilised in a single set of nozzles is said to consist of one *pressure stage*. If the pressure drop is subdivided between two, three, or more successive sets of nozzles, it consists of two, three, or more pressure stages.

When the kinetic energy of the steam jet is utilised in one row of rotating blades, the turbine is said to have one *velocity stage* per pressure stage. If, after the first row of rotating blades, a guide blade is placed to reverse the direction of steam so as to impinge on a second row of rotating blades, the turbine is said to have two 'velocity stages,' and so on.

Since a minimum nozzle height will be fixed by practical considerations (not less than ¼ in. even in the smallest turbines), the arc over which the nozzles will extend depends upon the total nozzle mouth area required. In the high-pressure stages this may be less than the mean circumference of the blades. Such a turbine is said to be working with partial admission, and the arc over which the nozzles extend is termed the 'arc of admission.'

This arrangement lends itself especially well to conveniently and efficiently dealing with overloads, it being only necessary to increase the 'arc of admission' by admitting steam to additional sets of nozzles.

In the same way it is possible to divide the total number of nozzles into blocks controlled by separate cutout valves. Thus one block of nozzles can be designed to give all loads from 0 to ½ full load, while these together with a second block give ½ to ¾ and so on.

By this means the loss of pressure through throttling at the governor valve is reduced to a minimum, and substantial economies can be effected on partial loads. The nozzle cutout valves may be operated automatically or by hand. Nozzle cutout governing is not possible with reaction turbines.

(a) Single Pressure Stage Turbine with a Single Velocity Stage.

The nozzles of all impulse turbines can be designed by the methods already indicated, while the work done and the efficiency can be obtained by drawing the triangles of velocity a_1 and a'_1 (see fig. 8). All velocities are in feet per second.

Let

W = total weight of steam through nozzles in lbs. per sec.; c_1 = absolute velocity leaving nozzles; U = peripheral velocity at mean diameter of blades; w_1 = relative velocity at entry

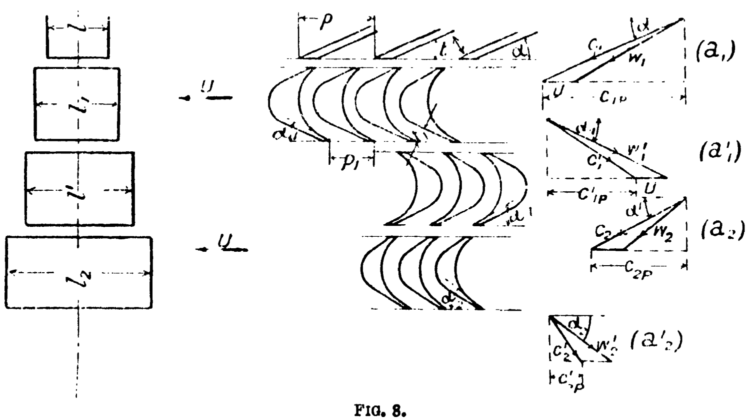


FIG. 8.

to moving blades; $w_1{}'$ = relative velocity at exit from moving blades; c_1 = absolute velocity leaving the moving blade; c_{1p}, $c_1{}'_p$ = the peripheral components of c_1 and $c_1{}'$, respectively.

Then

$$w_1{}' = \psi w_1,$$

where $(1 - \psi_1{}')$ is the loss of velocity due to friction in the blade (see fig. 12, page 124) ψ is called the *velocity coefficient* for the blades.

$$\left.\begin{array}{l}\text{The total work done, or}\\ \text{The indicated work}\end{array}\right\} = \frac{WU}{g}(c_{1p} + c_{1p}^{1}) \text{ ft.-lbs. per sec.}$$

$$\text{The blading efficiency } \eta_B = \frac{\text{(total work in ft.-lbs. per sec.)}}{W.\,J.\,g.\,h_a}$$

If

l = the required nozzle height; l_1 = the required blade height;

then

$$l_1 = l \cdot \frac{c_1}{w_1} \cdot \frac{t}{t_1} \cdot \frac{p_1}{p}$$

The de Laval turbine, which is of this type, is made in sizes from 5 to 500 h.p., running at 30,000 r.p.m. in the smallest size, and 10,000 r.p.m. in the largest outputs.

The peripheral speeds are as high as 1,300 ft. per second, and the steam velocity at exit from the nozzles as high as 4,000 ft. per second.

It can be shown that the efficiency varies with $\frac{U}{C_o}$, where Co is the velocity theoretically obtained if $\phi = 1{\cdot}0$.

For a maximum efficiency, $\frac{U}{C_o}$ should be about equal to 0·5 (see fig. 10, page 123).

(b) Single Pressure Stage. Two Velocity Stages.

In this case the work done in the first row of blades is obtained in exactly the same way as in case (a).

The guide-blade returns the steam jet towards the direction of U, the steam leaving it with an absolute velocity $c_2 = \psi_2 c_1^{1}$.

The relative velocity of exit from the second rotating blade is $w_2^{1} = \psi_2^{1} w_2$, knowing which, the velocity triangles a_2, a_2^{1} can be drawn. In a_2 and a_2^{1} the same symbols are employed as in a_1 and a_1^{1}, a suffix 2 denoting the guide-blade, while a dash denotes the second moving blade.

The total work per second is now

$$= \frac{WU}{g}\left\{(c_{1p} + c_{1p}^{1}) + (c_{2p} + c_{2p}^{1})\right\} \text{ ft.-lbs. per sec.}$$

WORK DONE IN 1ST AND 2ND ROWS OF VELOCITY WHEELS. 3,000 R.P.M.

Initial conditions :—200 lbs. per sq. in. g., 200° F. supf.; steam pressure in casing = per sq. in. g.; temperature in casing = 410° F.

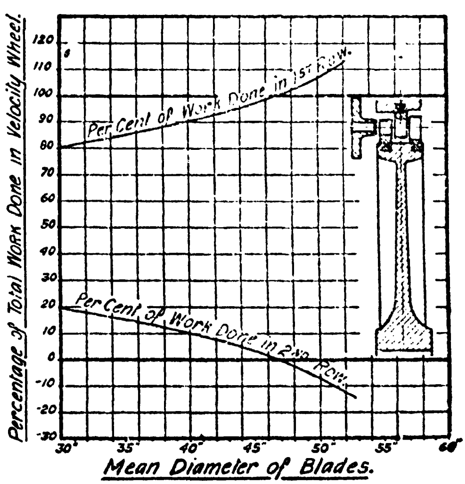


Fig. 9.

and

$$\eta_3 = \frac{\text{(Total work in ft.-lbs. per sec.)}}{W \cdot J \cdot g \cdot h_a}$$

$$\text{The height of the guide-blade } l' = l \cdot \frac{c_1}{c_2} \cdot \frac{t}{t^1} \cdot \frac{p^1}{p}$$

$$\text{The height of the second rotating blade } l_2 = l \cdot \frac{c_1}{w_2^1} \cdot \frac{t}{t_2} \cdot \frac{p_2}{p}$$

As the mean peripheral velocity of the blades increases the proportion of the work done in the first row increases. This effect is illustrated in fig. 9, from which it will be seen that for the condition stated the second row does no positive work if the mean diameter of the blades exceeds 47½ ins.; it can therefore be dropped with advantage. For this reason two-row wheels are rarely used in modern high-speed machines.

(c) Single Pressure Stage. More than two Velocity Stages.

The solution of this case can be obtained from (*b*) in exactly the same way as (*b*) was obtained from (*a*).

Single-wheel Curtis stages belong to classes (*b*) and (*c*) and are very much employed for small high-speed turbines, where efficiency is not the first consideration.

The peripheral speeds for Curtis wheels are as high as 600 ft. per second, and the steam velocity 2,500 ft. per second.

For a maximum efficiency $\frac{U}{C}$ should be about 0·23 for two rows of moving blades, and 0·15 for three rows (see fig. 10).

(d) Multi-Pressure Stage Turbine with Single Velocity Stages.

In order that reasonable efficiencies may be reached with practicable peripheral speeds, it becomes necessary to limit the heat drop utilised in each set of nozzles to some value considerably less than the total available between the conditions before the turbine stop valve and those present in the exhaust.

Up to a certain point, which is rarely reached in the conditions of actual design, the efficiency of each stage increases with the value of the ratio $\frac{U}{C_o}$ (see fig. 10).

When the value of $\frac{U}{C_o}$ is known, which corresponds to the desired turbine efficiency, the value of C_o and hence of h_a the heat drop to be utilised per stage can be calculated, since U will be fixed by mechanical considerations.

Then for an equal distribution of energy over the pressure stages,

if H_a = the total heat drop available for the whole turbine,

$$\text{The required number of stages } n = e \cdot \frac{H_a}{h_a}$$

where e is the '*reheat factor*' which depends on the amount of friction and eddy losses, the number of stages, the stage efficiency, etc., and varies between 1·03 and 1·07.

It is usually the practice to give the first stage a larger heat drop than the remainder, thus reducing the total number of wheels required, and having the considerable advantage of reducing the maximum temperature and pressure to which the turbine casing is subjected.

Having apportioned the heat drops, the steam conditions at each stage can be found—most conveniently by the Mollier diagram—and each stage can be designed as indicated under case (*a*).

The Rateau and Zoelly turbines are of this type.

In practice U varies from 350–600 ft. and the steam velocities from 800–1,600 ft. per second.

(e) Multi-Pressure Stage Turbines with Multiple Velocity Stages.

In precisely the same way as has been explained under (*d*), two- or three-row velocity wheels may be employed in the individual pressure stages.

The heat drop which can be advantageously used per pressure stage is then greater than in the case where a single row of blades is used, and thus the required number of pressure stages is reduced, but the efficiency is somewhat decreased, as may be deduced from fig. 10.

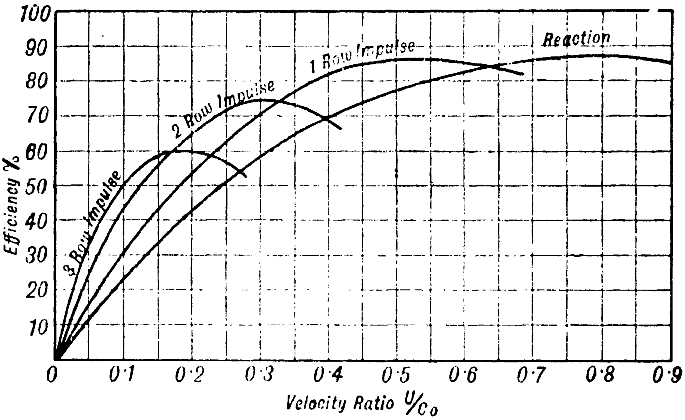


FIG. 10.

The general design of each stage is similar to that of case (*b*).

The Curtis turbine is of this type.

(f) Combination Impulse Turbines.

It has become a common practice to construct pressure-compounded impulse turbines of type (*d*) with a Curtis wheel having two rows of blades in the first stage. By this means a large heat drop can be employed in the first stage with reasonable efficiency and with the consequent advantages already mentioned.

The Curtis-Rateau and Curtis-Zoelly turbines are of this type.

CARRY-OVER ENERGY.

In pressure-compounded impulse turbines, the kinetic energy of the steam leaving the last blade of any pressure stage—except the last—is not entirely lost; part of it at least being recovered in the succeeding stage.

The total energy available in the succeeding stage is then greater than the adiabatic heat drop by an amount equal to $\frac{a\,c^2}{2g}$, where c is the absolute velocity of exit from the velocity blades and a is a constant.

For full admission, Prof. Rateau has suggested the value $a = \frac{2}{3}$, while for partial admission the value $a = \frac{1}{2}$ may be taken.

THE VALUE OF ψ, THE VELOCITY COEFFICIENT FOR BLADES.

The value of ψ has been the subject of researches by Prof. Rateau* and Dr. N. Briling.†

Prof. Rateau found that ψ was a characteristic of each type of blade depending on their size and spacing and slightly increasing with velocity.

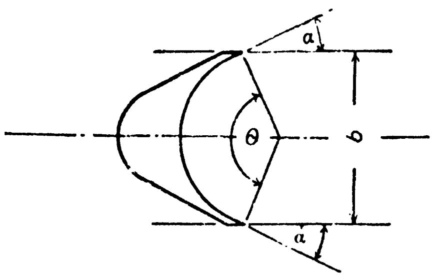
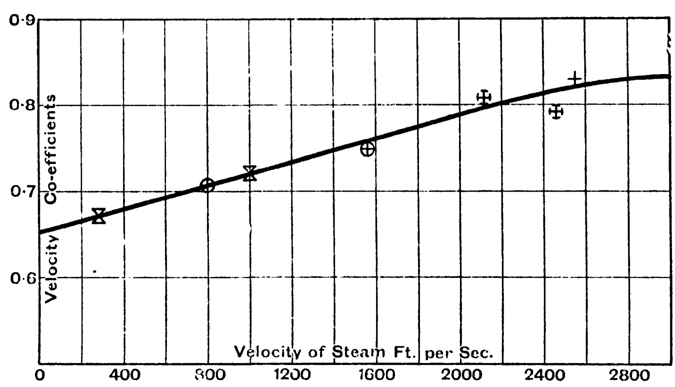


FIG. 11.

VELOCITY COEFFICIENT ψ.



⊕ *Rateau. (Engineering 10/12/09) Mean Values.*

⊕ *{Values deduced from Rateau's experiments on Bucket Reaction; Congrès International de Mécanique Appliquée 1900.*

X *Briling's Experiments with 20 m/m Buckets (Engineering 29/4/10).*

+ *{Stodola "Steam Turbines" deduced from Delaporte's Test of a de Laval Steam Turbine.*

FIG. 12.

Dr. Briling, whose experiments covered a wider field, found that ψ increased with velocity, decreased with the width of the bucket, and appeared to be principally dependent upon the angle through which the steam was diverted in the blade.

* *Engineering*, December 10, 1909. † *Z. V. d. I.*, January 1906.

Briling deduced the following formula from his experiments:

$$\psi = \psi_o - 0\cdot000432\,\theta^{\frac{1}{2}}$$

where θ = total angle in degrees through which steam is turned in the blade (see fig. 11).

ψ_o = 0·90–0·97 for blades with rounded inlet edges.
= 0·95–0·99 for blades with sharp edges.

Mr. H. M. Martin* has given a useful diagram combining Rateau and Briling's results with others deduced from various tests, which is reproduced in fig. 10; from this the following expression can be deduced:

$$\psi = 0\cdot65 + \frac{w}{15{,}000}$$

w being the velocity of the steam relatively to the blade in ft. per sec.

Pitch of Impulse Blades.

The best value of the pitch p has been determined experimentally by Dr. Briling † as being connected with the blade angle $\alpha°$—which he takes as being the same at the inlet and outlet—and the blade width by the following relation:

$$p = \frac{r}{2 \sin \alpha°}$$

$$r = \frac{b}{2 \cos \alpha°}$$

Briling is of the opinion that 30° is the most suitable angle for impulse blading.

In practice, the blade angle varies from 20° to 50°, the largest angles being used at the exhaust end of a turbine when it becomes difficult to provide sufficient area. The pitch varies from 0·5 to 0·75 of the blade width.

REACTION TURBINES.

In a reaction turbine the available heat energy of the steam is converted into kinetic energy both in the stationary and the moving blades.

In the most usual case—*considering a stage as consisting of a fixed blade and the following moving blade*—one-half of the available heat drop per stage is utilised in the fixed blade and the remainder in the moving blade. The turbine is then said to have a half degree of reaction.

Thus, both the fixed and moving blades perform the function of a nozzle in precisely the same way, and are identical in form and thermodynamic purpose.

Since there is a heat drop in the moving blades, there must also be a pressure drop across them. This is a characteristic of the reaction turbine as compared with the impulse turbine.

A second characteristic lies in the fact that the arc of admission to the first stage of a reaction turbine must extend around the complete circumference. For this reason nozzle cutout governing is not possible, and overloads can only be dealt with by admitting steam at some intermediate point in the expansion.

The steam enters the moving blades with a relative velocity approximately in an axial direction, and leaves them at a higher velocity in a direction inclined to the plane of rotation, the blades being impelled forward by the reaction of the leaving steam.

Reaction Blading with Half Degree of Reaction.

In this case since the outlet angles of the fixed and moving blades are the same, and the heat drop utilised in both is the same, the velocity triangles at inlet to and exit from the moving blade are identical (see fig. 13).

* *Steam Turbines—their Theory and Construction*, H. M. Martin.
† *Z. V. d. I.*, January 1906.

Hence

$$c_1 = w_1^1\ ;\ c_1^1 = w_1.$$

If h = heat drop in any stage (one fixed and one moving blade) and with the remaining symbols as were used when considering impulse turbines—

$$c_1 = \frac{1}{2\,U\cos\alpha}\,(25{,}050\,\zeta\,h + U^2);$$

where $(1 - \zeta)$ is the energy loss in the conversion of heat to velocity.

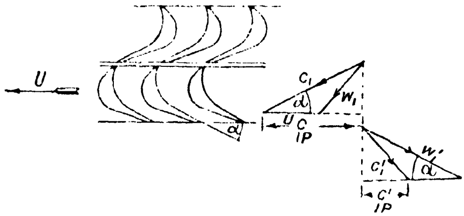


FIG. 13.

Capt. Riall Sankey gave 0·8 as the value of ζ, whereas the statement of Sir Chas. Parsons that the blading efficiency in the low-pressure end of the turbines of the 'Mauretania' reached the value of 85 per cent., shows that ζ may be as high as 0·86 under those conditions, while the efficiencies reported to have been obtained with the Ljungström turbine of quite small capacities could only be reached if ζ was as much as 0·35.

U and α being known, the velocity triangles can be drawn.

The work done per stage

$$= W\,\frac{(c_1^2 - c'^2_1)}{2\,g},\ \text{or,} = W\,\frac{(2k\cos\alpha - 1)}{g}\,v^2,\ \text{where } k = \frac{c_1}{U}.$$

The blading efficiency

$$\eta_B = \frac{c_1^2 - (c'^2_1)}{2g\,J\,.\,h},\ \text{or,} = U^2\,\frac{(2k\cos\alpha - 1)}{J\,.\,g\ h}.$$

The required blade length can be obtained from the relation

$$\pi\,.\,D_m\,.\,l\,.\,q = \frac{W\,.\,v_2}{W_1^1},$$

where D_m = mean diameter of annulus in feet; l = radial length of annulus in feet = (blade height) + (tip clearance); v_2 = the specific volume after the moving blade in cubic feet per lb.; q = annulus area factor.

At any stage the 'annulus area factor' is the ratio of the available area of the channels at exit from the blades in a direction at right angles to α, to the area of the annulus between the drum and the casing at that stage.

The properties of Parsons blading, with the exception of q and α, were published by M. Lelong,* and will be found in Table IV.

As in the case of impulse blading, the blading efficiency varies with $\frac{U}{c_1}$ and reaches a maximum when $\frac{U}{c_1} = 0{\cdot}945$ if $\alpha = 19°$, the normal angle for all but the last stages of reaction turbine. Under this condition U becomes equal to $c_1 \cos\alpha$. Such a high value is not practicable in ordinary working conditions, nor is it necessary for the realisation of high efficiency, because the curve connecting $\frac{U}{c_1}$ and η_B is very flat in the region of its maximum.

Fig. 14 (p. 128), which gives this relation for $\zeta = 0{\cdot}8$, is due to Stodola.†

The efficiency for changes of ζ within the limits actually to be met with can be obtained by increasing the efficiency as obtained from fig. 9 by the same percentage as ζ exceeds 0·8.

In electrical work $\frac{U}{c_1}$ varies from 0·5 to 0·8.

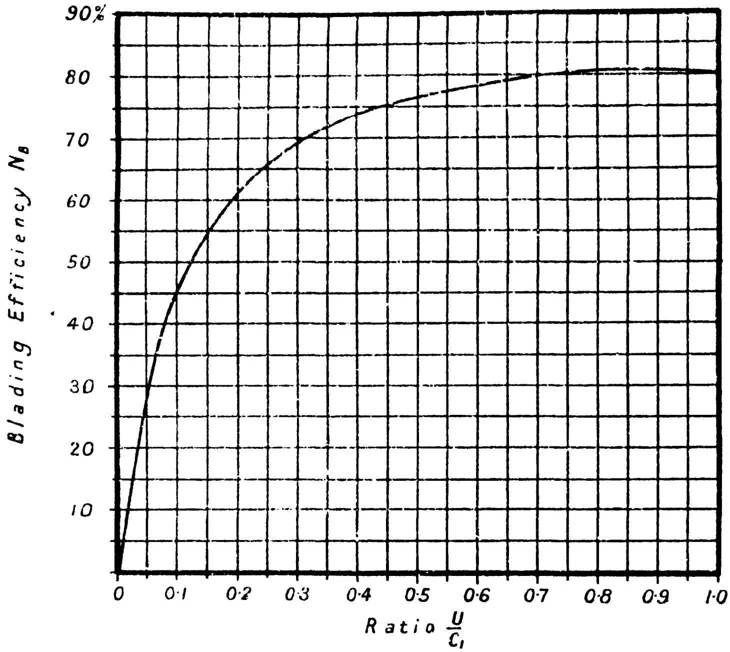
The peripheral velocity of forged steel drums may be as high as 400 ft. per second, and even higher speeds are possible with solid drums, or drums made from special steels of high ultimate strength.

* *Turbines à Vapeur Marines*, R. Lelong; and *Engineering*, March 1, 1912.
† *Die Dampfturbinen*, 4th Ed., A. Stodola.

TABLE V.—DATA RELATING TO BLADES AND CAULKING PIECES FOR PARSONS TURBINES.

Blades.						Caulking Pieces.									
						Casing.					Drum.				
Section Number.	Class of Blade.	Exit Angle of Blade α. In deg.	Annulus Area Factor q.	Weight per foot run.	Usual Limits of Length.	Section Number.	Number Required per in. run.	Weight per foot run.	Grooves. Width.	Grooves. Depth.	Section Number.	Number Required per in. run.	Weight per foot run.	Grooves. Width.	Grooves. Depth.
									In.	In.				In.	In.
120B	Normal	20	0·3	0·041	Up to ¾ ins.	120C	5·8	0·106	0·243	¼	120C	5·8	0·106	0·243	¼
130B	Normal	20	0·3	0·089	¾ in. to 4 ,,	130C	3·7	0·264	0·364	¼	130S	5·2	0·159	$\frac{23}{64}$	¼
130B	Semi-Wing	35	0·53	0·089	¾ ,, ,, 4 ,,	131C	3·3	0·315	0·415	¼	131S	3·7	0·300	$\frac{13}{32}$	¼
132B	Wing	45	0·68	0·077	¾ ,, ,, 4 ,,	132C	3·2	0·400	0·406	¼	132S	3·6	0·320	0·400	¼
240B	Normal	20	0·3	0·180	4 ,, ,, 8 ,,	240C	2·75	0·450	0·510	⅜	240S	3·4	0·400	0·510	⅜
240B	Semi-Wing	35	0·53	0·180	4 ,, ,, 8 ,,	241C	2·54	0·643	0·540	⅜	241S	3·2	0·450	0·540	⅜
250B	Normal	20	0·3	0·246	8 ,, ,, 12 ,,	250C	2·2	0·783	0·610	⅜	250S	2·7	0·600	0·610	½
250B	Semi-Wing	35	0·53	0·246	8 ,, ,, 12 ,,	251C	2·1	0·917	0·661	⅜	251S	2·6	0·670	0·661	½
252B	Wing	45	0·68	0·237	8 ,, ,, 12 ,,	252C	1·96	1·080	0·716	⅜	252S	2·44	0·79	0·716	½

For Double Wing Blades $\alpha = 30°$; $q = 0·85$ in.



EFFICIENCY OF REACTION BLADING FOR $\zeta = 0·8$.

FIG. 14.

The velocity of the steam c varies from 300 ft. per second in the initial to 1,000 ft. per second in the final stages.

In marine work $\frac{U}{c_1}$ varies from 0·3 to 0·45, and for the high-pressure stages the peripheral velocity varies from 80 to 120 ft. per second.

REACTION TURBINE DESIGN.

The blades of Parsons turbines are not usually increased in height from row to row, but are arranged in groups having the same height and the same mean diameter. The angles of the blades of any one group are usually the same, but in some cases they are gradually increased so as to obtain the correct progression of areas for constant steam velocity relatively to the blade.

The blades of such turbines for land purposes are generally placed on three-drum diameters of increasing size, referred to as the H.P., I.P. and L.P. expansions.

The relative distribution of heat drop between the different expansions and their relative geometric properties are given in the following table :—

Expansion.	H.P.	I.P.	L.P.	H.P.	I.P.	L.P.
Adiabatic heat drop used . .	1	1	2	1	1	1
Number of groups of blades .	1	1	2	1	1	1
Mean diameter of annulus . .	1	$\sqrt{2}$	2	1	$\sqrt{2}$	2
No. of rows of blades per group .	1	½	¼	1	½	¼
No. of rows of blades per expansion	1	½	½	1	½	¼

The first of these arrangements is that most commonly adopted.

In electrical work there are from two to three groups of blades in the H.P. expansion.

In any given turbine and for groups of blades with the blade angle the same, the ratio $\frac{D^2 l}{v}$ should be constant, where D = mean diameter of the blades, l = their length, and v = the mean specific volume of the steam in the group.

The last three rings on the L.P. expansion were formerly made the same height, but with increasing discharge angle, the latter corresponding respectively to 'normal,' 'semi-wing' and 'wing.' By this means the final ring (*i.e.* wing blades) has double the discharge area of the ring of 'normal' blades of the same height, but without any increase in the centrifugal stresses. The practicable height ratio of $\frac{\text{blade height}}{\text{rotor diameter}}$ was $\frac{1}{4}$ with blades cut from rolled parallel strip. In modern practice a still larger height ratio ($\frac{1}{3}$) is obtained, with a view to obtaining still larger discharge area in the final ring, by the use of twisted blades which are also, if necessary (on account of centrifugal stresses), tapered from root to tip. In these blades the discharge opening ratio $\frac{e}{p}$ is kept constant from root to tip and equal to about 0·71 (*i.e.* wing), but the discharge angle is progressively varied by twisting. The guide ring is made complementary to the final rotating ring, being twisted, but not tapered.

In order to reduce the unsupported length between bearings, and to enable the tip clearance to be kept small, it is becoming the practice in turbines of 5,000 kw. and over to use two cylinders. The first cylinder contains the H.P. expansion only and can be made of cast steel, very fine clearances being possible by eliminating the danger due to the growth of cast iron.

c_1, for any particular stage, can be calculated from the formula already given when the heat drop h is known.

The blading efficiency throughout the turbine can be found from fig. 14 when the ratio $\frac{U}{c_1}$ is known.

For groups of blades having the same angles and height:

if

n = number of blades in the group considered ; R = ratio of expansion over the group;
H_a = adiabatic heat drop for the group,

then the heat drop used in the first stage

$$h_1 = \frac{(1+e)H}{\left[0{\cdot}07 + \left(\frac{R^{\frac{2}{N}} - 1}{R^{N} - 1}\right)\right]}$$

The heat drops for the other stages of the groups form with h_1 a geometric progression having $R^{\frac{2}{N}}$ as a common ratio.

In most cases it will be sufficient to calculate the first and last stages of any one group.

Disc and Drum Type.

Just as in the development of the pure impulse turbine combinations of the simple types have been evolved, so the disc and drum type has been introduced into the sphere of the reaction turbine.

In this case the high pressure and less efficient part of the reaction turbine is replaced by a Curtis wheel having as a rule two rows of blades. This design possesses the advantages already mentioned in connection with combination impulse turbines, and in addition it results in a considerably shorter and stiffer rotor than in the pure Parsons machine.

TURBINE LOSSES.

Diaphragm Gland Loss.

The diaphragms of a pressure-compounded impulse turbine must obviously be provided with a clearance between themselves and the shaft, thus providing a leakage area through which steam can pass.

The diaphragm gland is usually made either as a labyrinth packing or as a closely fitting white metal bush.

Although the clearances may originally be very small they are greatly increased by whipping of the shaft, or even by the static deflection.

The percentage leakage can be estimated as being equal to the ratio of the clearance area to the nozzle area in any diaphragm.

In order to reduce this loss, some designers employ carbon packings held together by garter springs in the diaphragms.

The diaphragm clearance is usually from $\frac{10}{1000}$ ins. to $\frac{30}{1000}$ ins. radially.

PACKINGS.

Labyrinth Packings.

The glands at the cylinder ends are usually of the labyrinth packing type.

In impulse turbines the labyrinth is invariably of the radial clearance type, while in Parsons turbines the packing at the thrust block end is of the axial clearance type, the other being of the radial clearance type.

Mr. H. M. Martin* has deduced the following formulæ:

$$\text{Weight of leakage steam} = 68 \,.\, A \,.\sqrt{\frac{p_o\left(1 - \frac{1}{r^2}\right)}{v_o\,(N + \log_e r)}}\ \text{lbs. per sec.,}$$

where

p_o = pressure before the gland in lbs. per sq. in.; v_o = specific volume of steam in cu. ft. per lb.; r = ratio of expansion through gland; N = number of labyrinths; A = sectional area of leakage path in sq. ft.

If the velocity with which the steam flows out of the last labyrinth is greater than sound velocity, this formula should be amended, but the correction is small, and as it is in the nature of a reduction in the weight of the leakage steam, it can safely be neglected where N is greater than 8, which is the usual case in practice.

N varies between wide limits and may be from 8 in the case of low-pressure glands of impulse turbines to 30 for the high-pressure glands of Parsons turbines.

The clearance usually allowed varies from $\frac{10}{1000}$ in. to $\frac{20}{1000}$ in., but this may increase in actual working in the same manner as the diaphragm clearances.

In order to reduce the loss of turbine efficiency due to gland loss, which might otherwise be considerable, the high-pressure packing is usually connected at one or two points in its length to some intermediate stage in the turbine.

The leakage over each of these sections can be calculated by the formulæ already given, r now being the ratio of expansion over any section, and p_o, v_o the conditions before that section.

Except in those cases where a water gland is used, the gland at the exhaust end is sealed by steam at a higher pressure than atmospheric, taken from a stage in the turbine, and in order to ensure that air is not leaking into the exhaust, the pressure is adjusted until a small quantity of steam is seen escaping into the engine-room.

The cylinder glands of the A.E.G., Zoelly, and Curtis turbines are of the carbon type, for which a considerable reduction in leakage is claimed.

The advantages of these are questionable since they cannot be renewed or adjusted without opening the turbine, and in some designs it is even necessary to strip the rotor.

The carbon rings which are in segments are kept together and caused to bear on the shaft by garter springs.

Dummy Loss.

In the majority of single-flow Parsons turbines the axial thrust on the blades and drum is balanced by dummy pistons. A labyrinth packing is formed on the periphery of the dummy pistons to reduce the leakage of steam which would otherwise take place across the dummy piston to the exhaust to which they are connected on one side.

The loss of steam through the dummy clearances can be calculated in precisely the same way as the loss through a labyrinth packing.

The double-flow type of turbine obviating the necessity of balance pistons eliminates this loss.

* *Steam Turbines—their Theory and Construction*, H. M. Martin.

Disc Friction.

A considerable amount of work may be absorbed in rotating the discs of impulse turbines in the dense steam present in the first stages of a pressure-compounded (disc) turbine.

Professor Stodola * has given the following formula for the disc friction loss:

$$\text{B.H.P.} = \frac{6.1}{10^{9}} D^{2} U^{3} \gamma,$$

where,

D = mean diameter of blades in feet; U = mean peripheral velocity feet per sec.; γ = density of steam in lbs. per cu. ft.

The experiments upon which this formula is based were performed either in air, superheated steam, or in steam whose exact condition as regards wetness was unknown.

Professor Lewicki has demonstrated that the combined work of disc friction and windage falls off with superheat, some of his experiments showing a reduction of 10 per cent. in the work done for an increase of superheat of 100° C. As the total loss due to disc friction is usually of the order of a few per cent. of the output of the turbine, the reduction with superheat is of little importance.

On the other hand, it seems probable that the disc friction would increase rapidly with the wetness of the steam, but there are at present no data for estimating this increase. For these reasons, the disc friction loss for any given turbine cannot be calculated with certainty.

Windage.

In addition to the disc friction, loss results from the fanning action of the blades as they rotate.

The laws governing this action cannot be regarded as established, the range of experiments has been limited, and results by the same experimenters are discordant. Professor Stodola* gives the following formula:

$$\text{B.H.P.} = \frac{4.6}{10^{7}} . D . l^{1.5} . U^{3} . \gamma,$$

where,

l = blade length in inches; D = mean diameter of blades in ft.; U = mean peripheral velocity in ft. per sec.; γ = density of steam in lbs. per cubic ft.

Mr. W Kerr carried out some careful tests on this subject, and by comparing his own results with those of Professor Stodola and Dr. Lasche, suggests that for wheels carrying two rows of blades the work calculated by the above formula must be multiplied by 1·23, for three rows by 1·8, and for four rows by 2·9.†

When mixed pressure turbines are working with low-pressure steam the high-pressure end is rotating idly in an atmosphere of steam, and with disc turbines the work done in disc friction and windage is not negligible. With reaction turbines a windage loss alone is present, and this is mainly due to the fanning action of reaction blading. Sir C. A. Parsons has stated that in the case of a reverse element of a marine turbine rotating in a vacuum the loss is of the order of 1 per cent., so that the loss due to the rotation of the high-pressure element of a mixed-pressure Parsons turbine rotating in steam twelve times as dense as a 28-in. vacuum may by considerable.

Water Glands.

An interesting feature in the development of turbine design consists in the introduction of the water gland.

In this type, an air-seal is produced by a paddle-wheel acting as a centrifugal pump. The pressure difference over the gland is balanced by the centrifugal head produced in the water as it is whirled around covering areas of different internal radii on the two faces of the paddle.

The ribs on the sides of the paddle should be ¼ in. deep.

If P_1 (fig. 15) = the pressure on the high-pressure side in lbs. per square ft.; P_2 = the pressure on the low-pressure side in lbs. per square ft.; R_1 = inner radius in ft. of the whirling water on the H.P. face; R_2 = inner radius in ft. of the whirling water on the L.P. face; ρ = density of water = 62·3; ω = angular velocity of paddle in radians per second.

Then

$$P_2 - P_1 = \frac{\omega^2 \rho}{240\,A}(R_1^2 - R_2^2) \text{ and } P_2 - P_1 = \frac{\omega^2 \rho}{240\,A}(R_2^2 - R_1^2)$$

The value of A varies from 0·9 to 1·10″.

Although the water gland when first introduced gave rise to troubles, it has proved successful when used in conjunction with a labyrinth packing, for the purpose of preventing air leaking into the turbine casing.

Since a sufficient centrifugal head to form a seal is not developed until the turbine is running at some speed near the normal, it is necessary to steam-seal the glands as the turbine is started up.

The gland must be supplied with a constant stream of water to make up the loss which takes

* *Die Dampfturbinen*, 4th Ed., Professor A. Stodola.

† *Engineering*, August 22, 1913.

place by evaporation on the vacuum side of the seal. The water is admitted at the periphery of the gland, and should be supplied from a head of about 15 ft. This will, of course, be equal to (P_2-P_1).

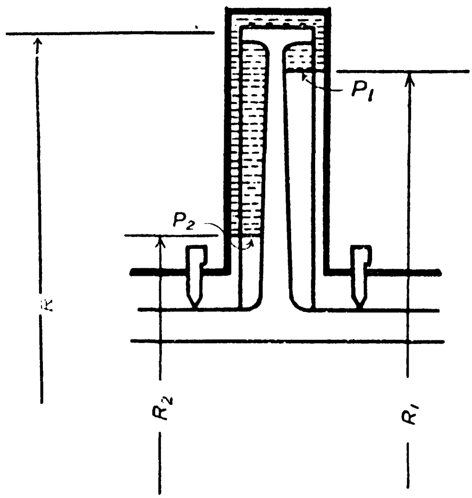


FIG. 15.

A water gland cannot be used to seal against steam at a pressure above atmospheric, because the water boils and the seal is destroyed.

Tip Clearance Loss.

As has already been pointed out, there is always a difference of pressure over both the fixed and rotating blades of a reaction turbine. In consequence of this there is always a leakage of steam

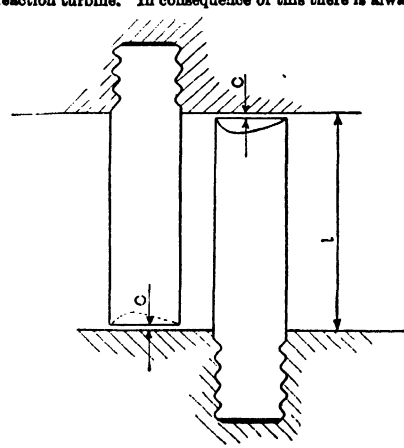


FIG. 16.

at the tips of the blades through the working clearances which have to be provided. In parallel flow reaction turbines there are two principal ways of providing these clearances:

(1) By means of radial clearance reaction blading (see fig. 16).

(2) By means of axial clearance or 'end tightened' reaction blading (see fig. 17).

The clearance C usually allowed in design varies approximately between the following limits for radial blading, all dimensions being in inches:

$$c = 0{\cdot}0008\,d \text{ to } c = 0{\cdot}0008\,d + 0{\cdot}004 \quad + 0{\cdot}015,$$

where,

d = mean diameter of blades.

l = blade height,

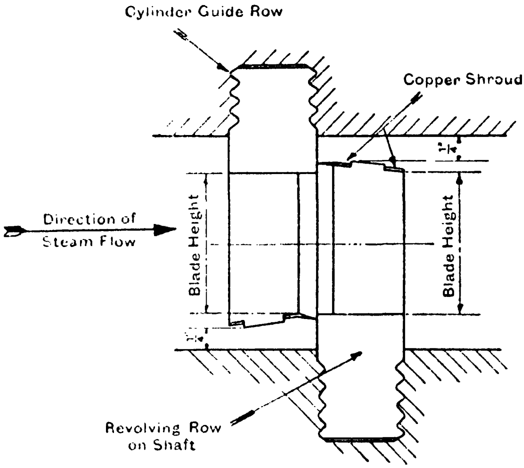


FIG. 17.—High-Pressure Turbine 'End Tightened' Reaction Blading.

If q is the annulus area factor—approximately one-third for normal reaction blading—the loss due to tip leakage can be expressed by $\frac{\lambda c}{q l}$; the value of λ deduced from the work of a number of experimenters varies from 1 to 2. In the case of end tightened reaction turbine blading illustrated in fig. 17 the fine clearances are arranged in an axial instead of a radial direction. This type of blading is usually confined to the high-pressure end of the turbine, and an adjustable thrust block is provided to enable this clearance to be adjusted.

Bearings.

l = the length of a bearing in inches; d = diameter of journal in inches; p = pressure on bearing per square inch projected area; N = speed of turbine in R.P.M.; μ = coefficient of friction; v = peripheral speed of journal in feet per second.

Then the work done in overcoming the bearing friction is

$$\text{Friction work} = \frac{\mu \pi l d^2 p N}{720} \text{ ft.-lbs. per sec.}$$

The heat equivalent of this work must be carried away from each bearing if they are to remain at a constant temperature.

According to Lasche's experiments, if $t°$ F. is the final steady temperature of the bearing, the coefficient of friction is given by the following relation:

$$\mu \, . \, p \, . \, (t° - 32°) = 51{\cdot}2.$$

For turbine bearings $t°$ should be from 120° F. to 150° F.

Assuming a temperature of 140° F., which is about an average value, these expressions can be reduced to:

$$\text{Work done in friction} = 3\cdot75\left(\frac{l}{10}\right)\left(\frac{d^2}{100}\right)\left(\frac{N}{1{,}000}\right)\text{ b.h.p.}$$

With average bearings bedded in the normal way, it is found that the work done exceeds the amount just given, being usually about twice as great.

One of the striking features of the recent developments in steam turbines has been the successive increases in the specific loadings and rubbing speeds on bearings.

This rating up of bearings has been accompanied by uniform success in all cases where the natural requirements of good lubrication have been safeguarded.

The oil supplied to the bearing has two functions to perform: it must sustain the oil film, and it must flush the bearing copiously to carry away the heat generated in shearing the oil film.

The quantity of oil required by the latter consideration is by far the greater of the two; the necessary amount of oil can readily be estimated from the relation that for ordinary turbine oils—

1·0 b.h.p. in oil friction raises 1·0 gallon of oil per minute by 10° F.

In the average case the temperature of the oil passing through the bearings is such that its temperature rise in passing through the bearing is from 20° to 40° F.; therefore, for 40° F. rise in temperature the oil quantity which should be supplied to a bearing is given by oil which should be supplied $= 2 \times \dfrac{3\cdot75ld^2N}{40 \times 10^6}$ gals. per min., or, approximately, $= \dfrac{2ld^2N}{10^6}$ gals. per min.

This represents the minimum quantity required, and may be increased up to double this amount with advantage.

The lengths of turbine bearings vary from $l = 2\cdot5d$ to $l = d$, the former value being usual for lightly loaded slow-speed bearings of older design, and the latter being approached with modern high-speed high-rated bearings.

At the present state of the art it is impossible to state the upper limits for permissible loading and rubbing velocity. Bearings are now quite common with loadings up to 110 lbs. per sq. in. and with rubbing velocities up to 130 ft. per sec.

According to modern ideas and practice, the old rules of the form pv = constant are quite unsound—in fact, when p is increased, v can and should be increased.

After leaving the bearings the oil is passed through an oil cooler, where it is cooled to some temperature usually below 110° F., although, of course, the actual temperature depends upon that of the cooling water available.

Turbine bearings give remarkably little trouble, provided the oil is kept clean and there is no tendency to emulsify.

In very small turbines the bearings are sometimes ring-lubricated, and in such cases the peripheral speed should not exceed 40 ft. per sec.; at higher speeds the oil tends to be thrown to the outside of the ring by centrifugal force.

Even in very small turbines there is now a tendency to fit a forced lubrication system provided with an oil cooler.

Recent Developments in Steam Conditions and Modification in Operating Cycle.

The component elements in a modern steam turbine and boiler plant are indicated in fig. 18.*

The various parts shown of steam turbine, regenerative feed heaters A, B, C, and D, economisers, boiler, superheater, resuperheater and air preheater, may be used all together, or one or other of the various elements may be omitted. For instance, the resuperheater appears desirable if steam pressures above 500 lbs./sq. in. are used, while air preheating will usually be employed if the feed

* The figures and data used in this section are taken from 'Economic Value of Increased Steam Pressures' of 1927, or from 'Tendencies in Steam Turbine Development' of 1929, both by H. L. Guy, before the Institution of Mechanical Engineers.

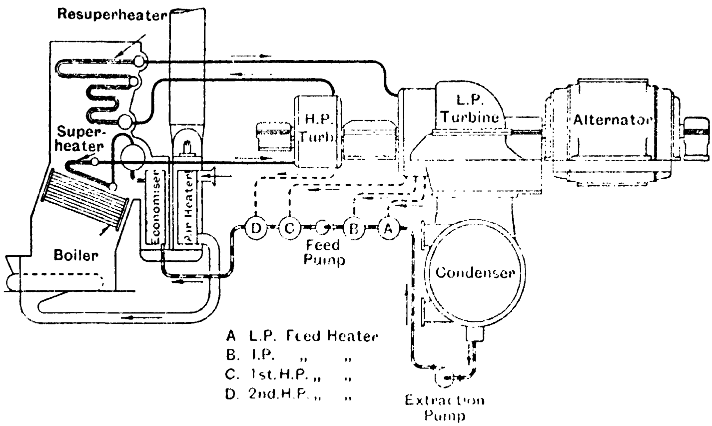

Resuperheater
Super-heater
Boiler
Economiser
Air Heater
H.P. Turb.
L.P. Turbine
Alternator
Feed Pump
Condenser
Extraction Pump
A L.P. Feed Heater
B. I.P. " "
C. 1st.H.P. " "
D. 2nd.H.P. " "

FIG. 18.

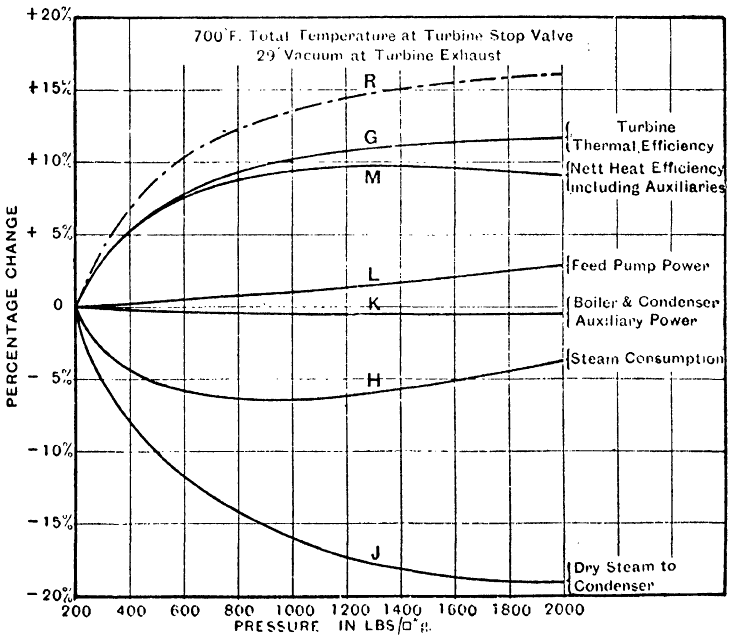

700°F. Total Temperature at Turbine Stop Valve
29" Vacuum at Turbine Exhaust
R
G
M
L
K
H
J
Turbine Thermal Efficiency
Nett Heat Efficiency including Auxiliaries
Feed Pump Power
Boiler & Condenser Auxiliary Power
Steam Consumption
Dry Steam to Condenser
+20%
+15%
+10%
+5%
0
-5%
-10%
-15%
-20%
PERCENTAGE CHANGE
200
400
600
800
1000
1200
1400
1600
1800
2000
PRESSURE IN LBS/□"g.

FIG. 19.

water is heated above 230° F. by the turbine feed heaters. In order to consider the influence of these elements on plant efficiency, the influence of steam pressure, steam temperature, regenerative feed heating and resuperheating will be considered separately.

Increased Steam Pressure.

The influence of steam pressure on heat consumption of the steam turbine itself is shown in fig. 19 by curve G; while curve M shows the corresponding gain when the effect of the turbine and boiler auxiliaries is taken into account. The meaning of the remaining curves in the figure will be evident.

Curve R shows the way in which the gain flowing from the use of increased steam pressure is increased by the use of four stages of regenerative feed heating, and therefore represents th relative gain which can be realised in a fairly elaborate modern installation of reasonable capac The last reservation arises from the fact that the gain which can be obtained by using incre pressure depends very much on the capacity and speed of the installation. The figures giv apply to an installation of about 20,000 kw. capacity if the speed is 3,000 r.p.m., or 40,000 kw capacity if the speed is 1,500 r.p.m. For much larger installations the returns will be ve slightly increased. For sensibly smaller capacities the gains will be materially reduced. Fr curve R the following table has been deduced, showing the pressure steps and pressures wh correspond to definite gains in coal consumption of 3 per cent. each time:

Table VI.—Increase in Steam Pressure for 3 per Cent. Steps in Thermal Efficiency.

Basis.—700° F. total temperature, 29-in. vacuum, four-stage feed heating. Including boiler and condenser auxiliaries. Capacity not less than 40,000 kw. M.C.R. at 1,500 r.p.m.; 20,000 kw. M.C.R. at 3,000 r.p.m.; 10,000 kw. M.C.R. at 6,000 r.p.m.

Increase in thermal efficiency	3%	6%	9%	12%	15%
Initial Pressure.	Final Pressure.				
From 200 lbs. per sq. in. g. to	265	365	505	764	1,360
„ 250 „ „ „ „	342	472	703	1,200	—
„ 350 „ „ „ „	495	755	1,390	—	—
„ 500 „ „ „ „	785	1,530	—	—	—
„ 750 „ „ „ „	1,470	—	—	—	—
„ 1,400 „ „ „ „	—	—	—	—	—
Reduction in coal consumption	2·92%	5·66%	8·26%	10·71%	13·04%

For 900° *F.*—Gain with pressure as above between 200 lbs. per sq. in. and 750 lbs. per sq. in. Gain with pressure increased by $\frac{1}{6}$ per cent. per 100 lbs. per sq. in. between 750 lbs. and 1,400 lbs. per sq. in.

For 28-*in. Vacuum.*—Gain with pressure increased by 0·3 per cent. per 100 lbs. per sq. in. between 200 lbs. and 750 lbs. per sq. in. Gain with pressure increased by 0·2 per cent. per 100 lbs. per sq. in. between 750 lbs. and 1,400 lbs. per sq. in.

The absolute values of the most important quantities entering into the calculation of the curves in fig. 19 are given in Table VIII. The figures given in lines 21 and 22 under 'Plant Efficiency' are based on a generator efficiency of 96 per cent. and a boiler efficiency of 84 per cent. It should be stated that all the preceding figures are based on an initial temperature of 700° F. and on a vacuum of 29". The relative position is not sensibly affected by increased steam temperature, but the net gains increase as the vacuum is decreased. For instance, the gain with 28" vacuum at 2,000 lbs./sq. in. is some 2 per cent. greater than that shown for 29" vacuum.

Increased Steam Temperature.

The most recent tendency in turbine development has been in the direction of increasing the initial temperature at which steam is supplied to the turbine. The increases in thermal efficiency resulting from increase in initial temperature are given in Table VII.

TABLE VII.—INCREASE IN TOTAL TEMPERATURE FOR 3 PER CENT. STEPS IN THERMAL EFFICIENCY.

Basis.—700° F. total temperature, 29-in. vacuum, four-stage feed heating. Including boiler and condenser auxiliaries. Capacity not less than 4,000 kw. M.C.R. at 1,500 r.p.m.; 2,000 kw. M.C.R. at 3,000 r.p.m.; 1,000 kw. M.C.R. at 6,000 r.p.m.

Increase in thermal efficiency . . .	3%	6%	9%	12%
Initial Pressure.	Increase in Temperature, ° F.			
200 lbs. per sq. in. g.	77	151	227	302
250 " " "	77	155	232	309
350 " " "	79	158	237	—
500 " " "	79	159	—	—
750 " " "	77	—	—	—
1,400 " " "	65	—	—	—
Reduction in coal consumption . .	2·92%	5·66%	8·26%	10·71%

Comparison of Tables VI and VII reveals the following interesting characteristics of pressure and temperature increases.

If the total temperature is kept constant, the changes in pressure necessary to secure a given gain in efficiency increase very rapidly, both with the initial pressure and with the magnitude of the gain in efficiency considered. On the other hand, the increase in temperature necessary to secure a given gain in efficiency is sensibly constant at all initial steam pressures whether the step in efficiency be 3 per cent. or 12 per cent.

In contrast with increases in initial pressure, the thermal advantage of increased temperature does not sensibly depend upon the size of the prime mover. A further distinction arises from the fact that a much smaller proportion of the complete boiler and turbine plant is affected by raising the temperature, than by raising the pressure.

Note.—Recently acquired knowledge on the properties of materials at elevated temperature has resulted in recognition of the fact that other things being equal steam pressures must be decreased as the steam temperature is increased. It is probable that the most desirable condition in the immediate future will be obtained by the adoption of a moderate steam pressure coupled with a higher steam temperature than has been normally used in the past.

Regenerative feed heating.

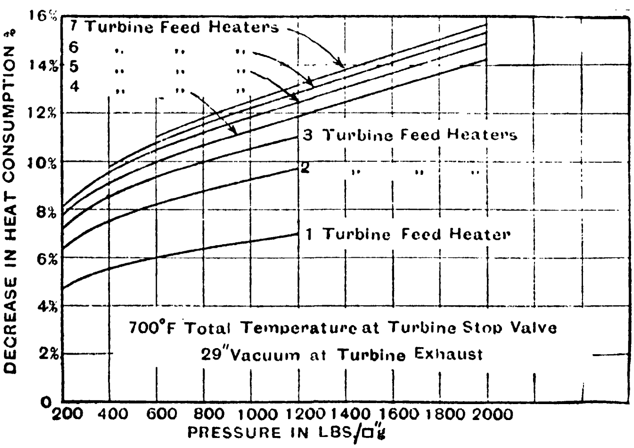


FIG. 20.

TABLE VIII — VARIATION IN EFFICIENCY WITH STEAM PRESSURE.
(700° F., 29-in. Vacuum.)

1	Initial pressure (Lbs./sq. in. g.)	200·0	285·3	385·3	485·3	585·3	785·3	985·3	1185·3	1385·3	1585·3	1785·3	1985·3
2	Initial pressure (Lbs./sq. in. Abs.)	214·7	300·0	400·0	500·0	600·0	800·0	1000·0	1200·0	1400·0	1600·0	1800·0	2000·0
3	Superheat (°F.)	321·1	282·2	254·5	231·4	211·7	179·1	152·7	130·3	110·7	93·3	77·6	63·2
4	Total heat from 78·9° F. (B.Th.U/Lb.)	1330·7	1326·2	1321·1	1315·9	1310·7	1300·4	1290·0	1279·2	1269·4	1259·0	1248·7	1238·4
5	Available heat to 29-in. vac. (B.Th.U/Lb.)	453·2	470·2	484·1	494·1	501·6	512·3	519·1	523·6	526·3	527·7	528·4	528·2
6	Theoretical thermal effy. (%)	34·06	35·45	36·64	37·55	38·27	39·39	40·24	40·91	41·46	41·91	42·32	42·65
7	Reheat factor	1·0601	1·0597	1·0594	1·0592	1·0594	1·0604	1·0620	1·0624	1·0666	1·0686	1·0701	1·0711
8	Final dryness	0·9050	0·8875	0·8720	0·8595	0·8490	0·8300	0·8130	0·7980	0·7855	0·7755	0·7675	0·761
9	Turbine losses factor	1·0000	0·9952	0·9910	0·9878	0·9852	0·9810	0·9778	0·9752	0·9729	0·9711	0·9694	0·9679
10	Comparative turbine effy. (%)	80·00	79·04	78·10	77·33	76·69	75·53	74·60	73·79	73·07	72·44	71·86	71·39
11	Comparative thermal effy. of turbine (%)	27·25	28·02	28·62	29·04	29·34	29·75	30·02	30·19	30·30	30·36	30·41	30·45
12	Power to feed pump (%)	0·35	0·45	0·58	0·71	0·85	1·14	1·42	1·74	2·07	2·41	2·77	3·15
13	Power to boiler auxiliaries (%)	1·50	1·46	1·43	1·41	1·39	1·37	1·36	1·35	1·35	1·35	1·34	1·34
14	Power to condenser auxiliaries (%)	2·00	1·91	1·85	1·80	1·77	1·72	1·68	1·65	1·64	1·63	1·62	1·62
15	Total power to auxiliaries (%)	3·85	3·82	3·86	3·92	4·01	4·23	4·46	4·74	5·06	5·39	5·73	6·11
16	Thermal effy. of turbine corr. for aux. (%)	26·20	26·94	27·51	27·89	28·17	28·50	28·68	28·75	28·76	28·72	28·67	28·58
17	Decrease in heat cons. with 4 F.H. (%)	7·72	8·42	8·99	9·36	9·86	10·60	11·21	11·83	12·43	13·02	13·60	14·20
18	Thermal effy. of turbine with 4 F.H. (%)	29·53	30·59	31·56	32·04	32·55	33·28	33·81	34·23	34·60	34·90	35·19	35·49
19	Total power to aux. with 4 F.H. (%)	3·42	3·40	3·42	3·45	3·51	3·79	4·11	4·52	4·98	5·42	5·96	6·44
20	Thermal effy. of turbine corr. for aux. with 4 F.H. (%)	28·53	29·55	30·38	30·94	31·41	32·02	32·41	32·69	32·88	33·01	33·09	33·20
21	Plant effy. without F.H. (%)	21·12	21·72	22·18	22·49	22·72	22·98	23·14	23·19	23·20	23·17	23·12	23·06
22	Plant effy. with 4 F.H. (%)	23·01	23·83	24·50	24·95	25·33	25·90	26·13	26·36	26·51	26·62	26·68	26.77

In recent years the practice of heating the feed water progressively by feed heaters supplied with steam tapped from the main turbine at various points in the expansion has developed until it has become standard. From one to four feed heaters are used, depending on the size and elaboration of the plant. The gains in coal consumption which can be expected at various steam pressures with from one to seven feed heaters are shown on fig. 20 (p. 137) for installations including an economiser and air preheater. If no air preheating is done, these gains are reduced to about two-thirds of those shown in this figure.

Resuperheating.

If steam is taken out of the turbine after partial expansion and passed through a superheater, the thermal efficiency is improved and consequently the coal consumption reduced. This gain in efficiency varies with the points in expansion at which resuperheating takes place. The upper

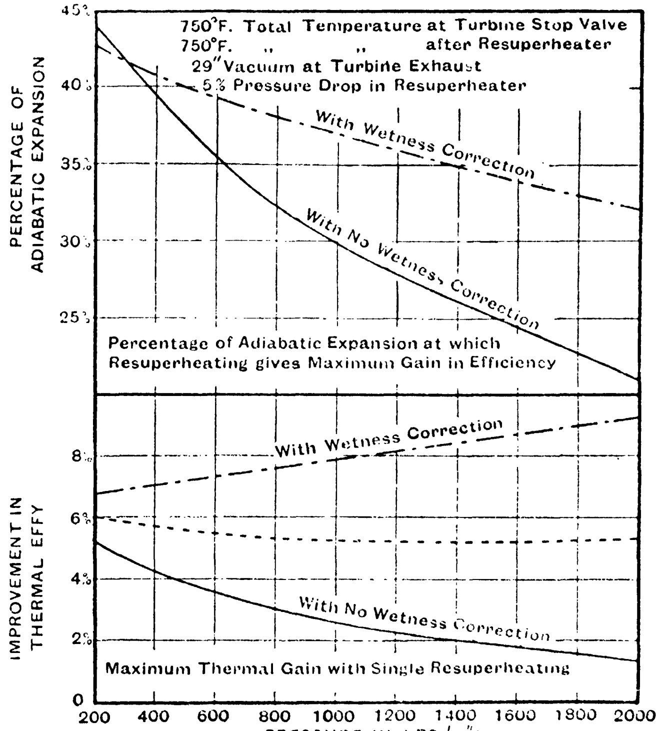


FIG. 21.

part of fig. 21 shows the percentages given of the total expansion which should precede resuperheating. The two curves shown should be treated as giving two limits. Again, in the lower portion, the two curves show the two limits for the reduction in coal consumption. The dotted curve shows the mean value and represents about the percentage which has so far been obtained by resuperheating.

TYPES OF TURBINES.

Turbines may be classified according to the steam conditions under which they operate. It should be noted that the energy available in adiabatic expansion of steam from 180 lbs. per sq. in. g. and 100° F. superheat to 28 ins. vacuum, is about twice that available if the expansion is carried down to atmospheric pressure only. Hence the addition of condensing plant approximately doubles the amount of work which can be done per lb. of coal, without necessitating any increase in boiler capacity.

HIGH-PRESSURE TURBINES.

High-pressure turbines take their steam directly from the boilers, and expansion takes place in the turbine down to the pressure in the condenser. This may be regarded as the fundamental class of turbine. The steam conditions are usually 350–150 lbs. per sq. in. g. pressure, 0°–300° F. superheat, and 27½ ins. to 29 ins. vacuum (30 ins. Barm.). Of course, the range of possible conditions is greater than these: for instance, in collieries and steelworks, steam pressures of 100–50 lbs. per sq. in. are to be met with, while steam temperatures of 800° F. will probably be adopted in the near future.

LOW-PRESSURE TURBINES.

Low-pressure turbines take their steam at about atmospheric pressure, the source almost invariably being the exhaust from non-condensing reciprocating engines. The initial steam pressure is usually from 20–15 lbs. per sq. in. abs., and should in all cases be above atmospheric pressure in order to eliminate the risk of air leakage into the steam pipe line. Low-pressure turbines require no governor gear if connected either mechanically or electrically to the recipro-

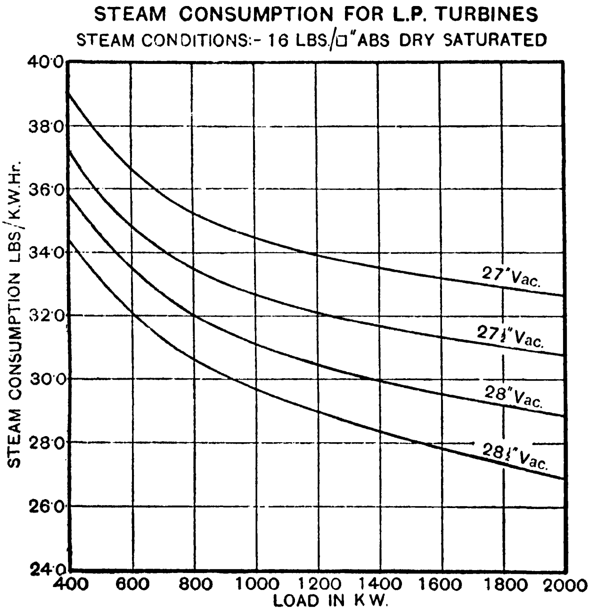


FIG. 22.

cating engine from which they obtain their steam. In the event of the engine being shut down the turbine is supplied with steam from the boilers through a reducing valve.

The efficiency with which low-pressure steam can be used naturally depends not only on the size and speed of the plant, but also on its capacity and costliness. As an average the steam consumption shown in fig. 22 can be used for estimating purposes when dry steam is supplied to a low-pressure turbine at a pressure of 16 lbs./sq. in. absolute.

Back-Pressure Turbines.

Back-pressure turbines take their steam directly from the boilers and exhaust at or above atmospheric pressure. Such turbines may be used in cases where the exhaust steam is required for heating purposes, and the back pressure usually ranges from 1 to 20 lbs. per sq. in. g.

In certain cases the back pressure may be as high as 40 lbs. per sq. in. g.

Back-pressure turbines are used to a considerable extent to drive the auxiliaries, their exhaust steam being utilised for feed heating; by this means the losses in the turbine are recovered in the feed heater, and an overall efficiency for back-pressure turbine and feed heater of over 90 per cent. is obtained.

Mixed-Pressure Turbines.

When L.P. steam is utilised in an L.P. turbine it is obviously desirable that the turbine set may be operated at will independently of the reciprocating engines from which it obtains its steam, so that in the event of the engine being shut down the turbine may operate with reasonable economy. This becomes possible by the addition of what is really a B.P. exhausting into the L.P. turbine, and having a common casing and spindle. This combination forms a mixed-pressure turbine, which can operate not only on either H.P. or L.P. steam, but if sufficient L.P. steam is not available to carry the load, the deficit is automatically supplied by admitting H.P. steam to the H.P. end of the turbine.

The governor gear is automatic and adjusts itself so that all the L.P. steam available is used in preference to H.P. steam.

In mixed-pressure turbines the governor gear is of special importance, as for successful working in parallel with other alternators the changes from L.P. steam to H.P. steam must be effected without any appreciable change in speed or load of turbo-alternator.

Reducing Turbines.

With a B.P. turbine the amount of steam available for heating purposes, and the work done by the turbine, are directly connected. In most installations it is desirable to be able to vary these quantities independently of each other, and to meet this demand 'Reducing' turbines were introduced.

Power Obtained from Heating Steam.

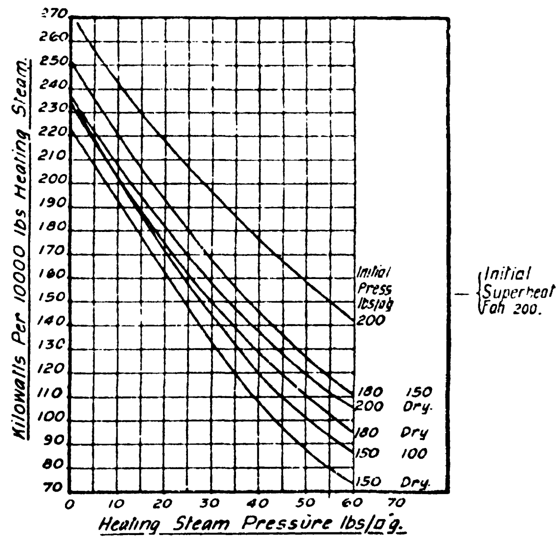


Fig. 23.

An R.P. turbine consists of a B.P. turbine to which an L.P. turbine is added. The B.P. turbine is designed for the maximum amount of steam required by the heaters it supplies.

As the demand for heating steam decreases, an automatic governor gear by-passes the supply to the L.P. end of the turbine, and at the same time the supply of steam to the H.P. end is decreased.

In this way the load the generator can carry is independent of the amount of heating steam required.

The governor gear should be operated in such a way that the pressure of the heating steam remains constant.

An increasing demand for this type of machine has arisen in recent years as a result of the extremely economical results which can be obtained in those manufacturing works requiring both electrical power and heating steam.

In order to obtain the best result the pressure at which steam is taken from the turbine for heating purposes should be as low as possible in view of the requirements of the heating process. The effect of back pressure on the consumption of heating steam can be seen from the following table, while the variation in the power which can be obtained from back pressure on reducing turbines is shown in fig. 23 for various steam conditions.

In recent years marked improvements have been made in the efficiency which can be obtained in H.P. ends of steam turbines. Whereas formerly the first step in expansion was accomplished with an efficiency of from 50 to 60 per cent., efficiencies of from 70 to 80 per cent. can now be realised in turbines of reasonable size and of other than the cheapest construction. It follows that in back pressure or reducing turbines of some considerable size, elaboration and costliness, the kw. obtainable per 10,000 lbs. of steam per hour can, in such cases, exceed those given in fig. 23 by some 25 per cent.

TABLE IX.

PERCENTAGE CHANGE IN STEAM CONSUMPTION OF WORK CAPACITY OF HEATING STEAM FOR A CHANGE IN HEATING STEAM PRESSURE OF 5 LBS. SQ. IN. G.

Initial Pressure. lbs. sq. in.	Initial Superheat. Degrees F.				
	Dry.	50	100	150	200
200	6·85	6·5	6·15	5·8	5·45
190	7·3	6·92	6·52	6·15	5·77
180	7·75	7·35	6·9	6·5	6·1
170	8·22	7·75	7·25	6·82	6·37
160	8·7	8·15	7·6	7·15	6·65
150	9·25	8·65	8 1	7·5	6·95

TURBINE TEST CORRECTIONS FOR STEAM CONDITIONS.

While it is rarely possible to make tests upon a turbine under the exact steam conditions for which it has been designed, the performance which may be expected under the designed conditions can be estimated from the steam consumption actually obtained during the test.

The usual practice consists in correcting the test consumption for the difference between the steam superheat, pressure, and vacuum during the test, and that which is specified for the turbine

These 'corrections' are arranged as coefficients by which the observed consumptions must be multiplied, and are determined for any steam conditions from consideration of the changes in available heat-drop and turbine efficiency which take place with any change in that condition.

A very complete list of these corrections has been published by Mr. K. Baumann,* and the values he gives are incorporated in this section.

Superheat Correction.

Where the superheat is different from that specified, the following corrections should be made, both for high-pressure or low-pressure turbines.

Between :—

0–100° F. superheat 1 per cent. improvement of steam consumption for every 10° F. superheat.

100–200° F. superheat 1 per cent. improvement of steam consumption for every 12° F. superheat.

200–300° F. superheat 1 per cent. improvement of steam consumption for every 14° F. superheat.

Correction for Initial Wetness of Steam.

For each 1 per cent. wetness the steam consumption, if measured as condensed water, will be 2 per cent. too high.

* *Recent Developments in Steam Turbine Practice*, Inst.E.E., April 1912

Pressure Correction.

The correction which is to be made for the pressure depends upon whether the change in pressure is before the stop valve or before the nozzles.

Thus, if in the case of a high-pressure turbine working on a constant generator load the pressure before the stop valve is raised, say 10 lbs. per sq. in., the governor valve closes slightly, but the pressure before the nozzle will be very little altered, the only result being that the steam is throttled to a greater extent than formerly.

The effect of such an increase of pressure will be that the steam consumption will be improved by the small amount corresponding to the superheat due to the additional throttling.

In such cases the steam consumption will be improved by 0·5 per cent. for every 10 per cent. increase of pressure.

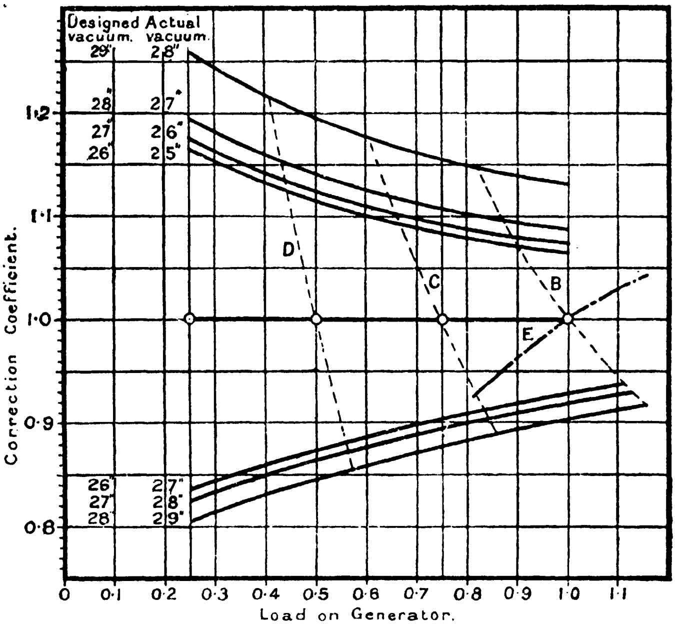


FIG. 24.—Vacuum Correction for Low-pressure Turbines.

When, on the other hand, the pressure before the nozzles is increased, the correction is of quite different magnitude and is:

For high-pressure turbines, an improvement of 1½ per cent. in steam consumption for 10 per cent. increase in pressure.

For low-pressure turbines, an improvement of 4 per cent. in steam consumption for 10 per cent. increase in pressure.

Vacuum Correction.

In any test, the correction for vacuum is usually of greater magnitude than that for pressure or superheat. Although every care has been exercised in estimating these corrections, it is obviously desirable that the total correction should be as small as possible, and if only for this reason every effort should be made to obtain during a test a vacuum as little different as practicable from that for which the turbine was designed.

Table X. gives the correction for vacuum which should be employed for various designed conditions with high-pressure turbines or mixed-pressure turbines, working with high-pressure steam.

TABLE X.—VACUUM CORRECTION FOR HIGH-PRESSURE AND MIXED-PRESSURE TURBINES.

Partial loads are expressed as fractions of the full load for which turbine is designed.

The correction is to be made when the turbine is run on a vacuum other than that for which it was designed.

Designed Steam Conditions. Pressure, Lbs. per sq. in. g.	Supt. °F.	Vacuum ins.	Theo. Corr. for 1 in. Vac. Variation.	Actual Correction for 1 in. Vac. Variation. H.P. 5/4 Load M.P. 4/4	H.P. 3/4 M.P. 5/8	H.P. 1/8 M.P. 1/4	H.P. 1/4 M.P. 1/8	Range of Vac. Variation. Ins.
180	150	26	4·4	3·0	3·8	4·9	6·5	24—26—27
		27	5·5	3·7	4·6	5·6	7·6	25—27—28
		28	7·0	5·1	6·2	7·5	9·7	26—28—28½
		29	10·3	7·5	8·5	10·2	12·8	28—29
140	150	26	4·7	3·2	4·2	5·0	6·9	24—26—27
		27	5·8	3·9	4·8	5·9	8·0	25—27—28
		28	7·3	5·3	6·3	7·8	10·2	26—28—28½
		29	11·0	7·8	8·8	10·7	13·5	28—29
100	150	26	5·0	3·5	4·5	5·5	7·5	24—26—27
		27	6·2	4·2	5·3	6·4	8·7	25—27—28
		28	7·9	5·7	6·6	8·4	10·9	26—28—28½
		29	11·9	8·5	9·5	11·5	14·5	28—29
75	150	26	5·4	3·8	4·8	6·0	8·0	24—26—27
		27	6·6	4·6	5·6	6·9	9·4	25—27—28
		28	8·5	6·1	7·1	9·0	11·6	26—28—28½
		29	12·7	9·1	10·3	12·3	15·5	28—29
50	150	26	5·9	4·3	5·2	6·7	8·7	24—26—27
		27	7·2	5·0	6·0	7·7	10·3	25—27—28
		28	9·2	6·6	7·7	9·9	12·6	26—28—29
		29	13·5	9·9	11·3	13·3	16·8	28—29

The Theoretical Correction is given for 150° F. Superheat.

The actual correction given for various loads can be used for any superheat between saturation and 300° F.

It should be noted that at any load and for the same designed steam conditions and actual vacuum, the correction in the case of a mixed-pressure turbine (M.P.) is greater than for a high-pressure turbine (H.P.).

Thus, for example, the correction for a M.P. turbine at full load and working with high-pressure steam is the same as for a H.P. turbine on half load. The assumption underlying this is that the M.P. turbine is designed to give the same output on H.P. as on L.P. steam, and that, therefore, the conditions in the nozzles and blading of the L.P. end of the M.P. turbine are the same as that of a H.P. turbine designed for twice that output. Changes in vacuum only affect the conditions in the last few stages of a turbine, and these in a M.P. turbine on H.P. steam are operating under the same conditions as those of a H.P. turbine at half load.

Vacuum corrections for low-pressure turbines or for mixed-pressure turbines working with low pressure steam are still larger than those already considered.

In fig. 24 a coefficient will be found for various loads on the generator which is the ratio :—

(Steam consumptions for 1 in. above or below the designed vacuum.)

(Steam consumption for the designed vacuum.)

While the corrections as given above can be applied to the generality of cases, special conditions may arise where corrections must be separately considered.

Thus, it is possible to obtain little or no benefit by increasing the vacuum with a turbine designed for a certain load at a certain vacuum and modified to operate on a considerably higher load and a higher vacuum; in such a case the velocity in the exhaust would be exceedingly high, and may even approximate to sound velocity.

In back pressure turbines, or the high pressure end of reducing turbines, all corrections will be considerably larger than those given above, and these require separate consideration in each case.

It cannot be too strongly emphasised that every endeavour should be made to arrange the test conditions in such a way that all corrections are as small as possible.

Centrifugal Stresses in Drums and Discs.

Stresses in thin Ring or Drum.

Mean tangential stress $= 2\,(D)^2\left(\frac{N}{1000}\right)^2$ lbs./sq. ins.

Solid Parallel-sided Disc.

If U_2 = peripheral velocity at the outside diam. in ft./sec.,

then radial stress at any radius x is $\sigma_r = \frac{4{\cdot}36U_2^2}{100}\left[1 - \frac{x^2}{x_2^2}\right]$ lbs./sq. in.

and the tangential stress $\sigma_t = \frac{U_2^2}{100}\left[4{\cdot}36 - 2{\cdot}51\frac{x^2}{x_2^2}\right]$ lbs./sq. ins.

The maximum stress at the centre is given by

$$\sigma_{r\,max} = \sigma_{t\,max} = \frac{{\cdot}36\,U_2^2}{100}\ \text{lbs./sq. ins.}$$

If the diameter of the disc = D inches, this can be written

$$\sigma_{max} = 0{\cdot}835\,(D)^2\left(\frac{N}{1000}\right)^2\ \text{lbs./sq. ins.}$$

Parallel-sided Disc with Hole in Centre.

At any radius x

Radial stress $\sigma_r = \frac{4{\cdot}36}{100}U_2^2\left[1 + \frac{(x_1^2 - x^2)}{x_2^2} - \frac{x_1^2}{x^2}\right]$ lbs./sq. ins.

Tangential stress $\sigma_t = \frac{U_2^2}{100}\left[4{\cdot}36\left(1 + \frac{x_1^2}{x_2^2} + \frac{x_1^2}{x^2}\right) - 2{\cdot}51\frac{x^2}{x_2^2}\right]$ lbs./sq. ins.

Dimensional symbols as shown in fig. 25.

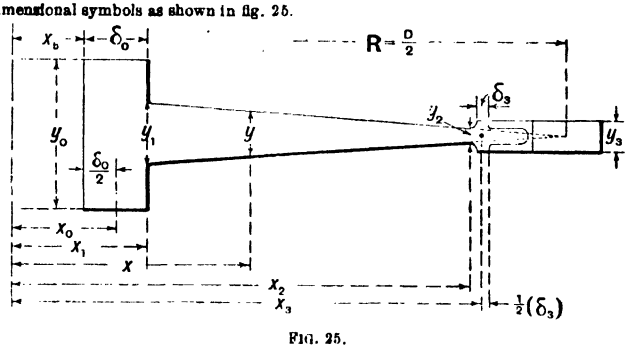


FIG. 25.

Disc with Hyperbolic Profile.

If the profile can be expressed as a hyperbolic function of the form $y = \frac{(\text{constant})}{x^a}$, the following analysis can be used ·

Let

ω = angular velocity in radians/sec. $= \frac{N \times 2\pi}{60}$;

E = Young's modulus of elasticity = 30×10^6 lbs./sq. in. for steel;

ν = Poisson's ratio = 0·3 for steel;

σ_3 = load on outer edge of disc due to blade loading in lbs./sq. in.;

σ_0 = stress at bore due to pinch fit;

μ = mass per unit volume;

then
$$a = \frac{\log\left(\frac{y_2}{y_1}\right)}{\log\left(\frac{x_1}{x_2}\right)}.$$

$$\psi_1 = \frac{a}{2} + \sqrt{\frac{a^2}{4} + a\nu + 1}, \quad \psi_2 = \frac{a}{2} - \sqrt{\frac{a^2}{4} + a\nu + 1}.$$

and
$$a = -\frac{(1-\nu^2)\mu\omega^2}{E[8-(3+\nu)a]}$$

then radial stress at any point

$$\sigma_r = \frac{E}{(1-\nu^2)}\left[(3+\nu)\,a\,x^2 + (\psi_1+\nu)b_1 x^{\psi_1-1} + (\psi_2+\nu)b_2 x^{\psi_2-1}\right] \quad \ldots.(1)$$

and tangential stress

$$\sigma_t = \frac{E}{(1-\nu^2)}\left[(1+3\nu)\,a x^2 + (1+\psi_1\nu)b_1 x^{\psi_1-1} + (1+\psi_2\nu)b_2 x^{\psi_2-1}\right] \ldots.(2)$$

In order to calculate constants b_1 and b_2, make use of the boundary conditions :—

Extension at outside, where $x = x_2$

$$\xi_2 = \frac{x_2^2}{E\,\delta_2 y_2}\left(\sigma_2 y_2 + \mu\omega^2\delta_2 y_2 x_2 - \sigma r_2 \frac{x_2 y_2}{x_2}\right) \quad \ldots.(3)$$

Extension at inside, where $x = x_1$

$$\xi_1 = \left(\frac{x_0^2}{E\,\delta_0 y_0}\sigma_0 y_0 + \mu\omega^2\delta_0 y_0 x_0 + \sigma r_1 \frac{x_1 y_1}{x_0}\right) \quad \ldots.(4)$$

But
$$\xi \text{ also} = a x_1^3 + b_1 x_1^{\psi_1} + b_2 x_1^{\psi_2} \ldots\ldots\ldots\ldots\ldots\ldots.(5)$$

and
$$\xi_2 = a x_2^3 + b_1 x_2^{\psi_1} + b_2 x_2^{\psi_2} \ldots\ldots\ldots\ldots\ldots\ldots.(6)$$

By equating (3) and (6), also (4) and (5), the two resulting equations can be solved for b_1 and b_2.

When these values are put in (1) and (2) the radial and tangential stresses at any point in the disc can be obtained.

Disc with Straight-sided Taper Profile.

Treating a disc as having a conical profile, Mr. H. M. Martin * has derived a means of arriving at the stresses, which, together with the amendment to incorporate the effect of the hub, as suggested by Mr. B. Hodkinson,† provides a useful method.

The stresses throughout a conical disc are:

Radial $= \sigma_r = Tp_0 + Ap_1 + Bp_2$ ....(1)

Tangential $= \sigma_t = Tq_0 + Aq_1 + Bq_2$..(2)

where T = stress in a thin ring

$= 2D^2\left(\frac{N}{1{,}000}\right)^2$ lbs./sq. in. for steel.

D = diameter of completed cone.

$p_0, p_1, p_2, q_0, q_1, q_2$, are given in the curves, fig. 26, co-ordinated with the ratio $\frac{x}{R}$.

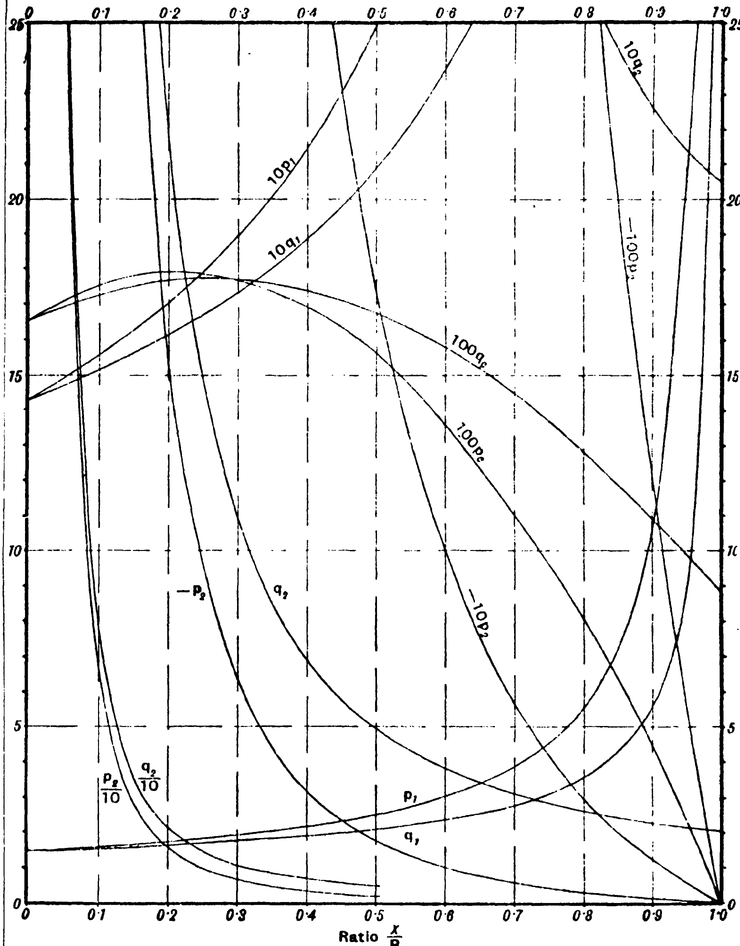
The constants A and B have always to be found by writing 'boundary conditions.' Thus at the inner limit of the conical portion where it joins the hub :

$$\sigma t_1 = \left\{\sigma_4 \frac{2x^2}{x_1^2 - x^2} + 2\left(\frac{N}{500}\right)^2(0{\cdot}175x_1^2 + 0{\cdot}825x_b^2)\right\}$$
$$+ \sigma_{r_1}\left\{\frac{y_1}{y_0}\left[\frac{x_1^2+x^2}{x_1^2-x^2} - 0{\cdot}3\right] + 0{\cdot}3\right\} \ldots\ldots\ldots\ldots(3)$$

where σ_4 is the 'pinch' pressure inside the bore in lbs./sq. in. At the rim

$$\sigma_{t_2} = \left\{\frac{\text{Total centrifugal force of blades}}{2\pi\delta_2 y_2} + 2x_2^2\left(\frac{N}{500}\right)^2\right\} - \sigma_{r_2}\left\{\frac{x_2 y_2}{\delta_2 y_2} - 0{\cdot}3\right\} \ldots(4)$$

* See *Engineering*, 5th & 26th January 1923.
† See *Engineering*, 31st August 1923 and 7th September 1923.



0
0·1
0·2
0·3
0·4
0·5
0·6
0·7
0·8
0·9
1·0
25
20
15
10
5
0
$10p_1$
$10q_1$
$10q_2$
$-100p_2$
$100q_c$
$100p_c$
$-10p_2$
$-p_2$
q_2
$\frac{p_2}{10}$
$\frac{q_2}{10}$
p_1
q_1
Ratio $\frac{x}{R}$

FIG. 26.

Putting in equations (1) and (2) the values of p_c, p_1, p_2, q_c, q_1, q_2, at the inner radius of the cone, and using the resulting expressions as σr_1 and σt_1 in equation (3), an equation involving only A and B is obtained. Writing similarly for the outer radius of the cone a second equation involving A and B, these constants can be calculated, whence the stresses anywhere in the disc are known.

The stresses throughout the hub then are:

$$\sigma r = \sigma_{r_1} \frac{y_1 (x^2 - x_b^2)}{y_0\, x_1^2 - x_b^2} \frac{x_1^2}{x^2} - \sigma_4 \frac{x_b^2}{x^2} \frac{(x_1^2 - x^2)}{(x_1^2 - x_b^2)} + 0\cdot825 \left(\frac{N}{500}\right)^2 \left(x_1^2 + x_b^2 - x^2 - \frac{x_b^2 x_1^2}{x^2}\right).$$

$$\sigma t = \sigma_{r_1} \frac{y_1 (x^2 + x_b^2)}{y_0 (x_1^2 - x_b^2)} \frac{x_1^2}{x^2} + \sigma_4 \frac{x^2}{x^2} \frac{(x_1^2 + x^2)}{(x_1^2 - x_b^2)} + 0\cdot825 \left(\frac{N}{500}\right)^2 \left(x_1^2 + x_b^2 - 0\cdot575\, x^2 + \frac{x_b^2 x_1^2}{x^2}\right).$$

Pinch fit for Discs and Drums.

In order to ensure that the wheels shall be tight on the shaft at normal speed, they must be pressed on the shafts with an interference fit at least equal to the elastic expansion of the bore due to centrifugal stresses.

This condition can be fulfilled without the maximum stresses in the wheel being greater when the shaft is at rest than the centrifugal stress at normal speed, because in wheels of normal proportions the wheel bore expands during pressing on by only about two-thirds of the pinch allowance, the shaft diameter compressing by the remaining one-third.

CRITICAL SPEEDS.

TURBINE SHAFTS.

At the critical speed of a loaded shaft, a small disturbance sets up vibrations which may reach a dangerous magnitude. For a uniform shaft, when the obliquity of the loading masses may be neglected, the critical speed coincides with the natural frequency of transverse vibration.

Although turbines have been run for considerable periods at their critical speeds without failure, this is a very dangerous proceeding and should be carefully avoided. It follows, therefore, that the critical speed should always be estimated, and should be at least 30 per cent. removed from the running speed.

This particularly applies to turbines of the disc type, and the calculation can be made with some degree of certainty. In drum turbines, the calculation is much more difficult, but in such cases as usually arise the critical speed is much above the running speed.

Table XI. (p. 150) gives the formula for calculating the critical speeds of shafts of uniform diameter in certain standard cases.

Dunkerley has evolved theoretically, and demonstrated experimentally, a law from which can be determined the critical speed of a loaded system which corresponds to any combination of the standard cases.

By means of Table XI. in conjunction with Dunkerley's formula, the first critical speeds of sensibly uniform shafts can be approximately calculated for any combination of loading, and any method of fixing the shaft at the bearings.

Thus if a rotor consists of a uniform shaft loaded with weights $W_2 W_3$ etc. as external loads

n_1 = the critical speed of the shaft without the external loads;

n_2, n_3, etc. = the critical speed of a massless shaft loaded with weights W_2, W_3 etc.;

then the critical speed N of the system is given by

$$\frac{1}{N^2} = \frac{1}{n_1^2} + \frac{1}{n_2^2} + \frac{1}{n_3^2} + \ldots$$

In those cases of rotors having shafts stepped in diameter, as is usually the case in practice, the method already given is only a rough approximation, owing to the difficulty of judging the diameter of the equivalent uniform shaft.

Such cases may be dealt with by obtaining the maximum static deflection y of the shaft under various loads, by the graphical methods due to Mohr.

Then if

N = critical speed in R.P.M.;

$N^2 y = C$, where C is constant.

For impulse turbine shafts supported in bearings which do not fix the direction of the shaft Mr. K. Baumann * has found that the value of C varies from 37,700 to 38,000 if y is in inches.

WHEEL VIBRATION.

Under the influence of such periodic disturbances as partial admission, or even irregularities in steam admission, turbine wheels may be set into synchronous vibration, about symmetrically placed nodal diameters or nodal circles, concentric with the wheel itself. Such vibration may be avoided by correct design.

* *Recent Developments in Steam Turbine Practice*, Inst.E.E., April 1912.

As whirling of shafts may be avoided by the use of rigid or of flexible shafts, so too wheel vibration may be avoided by the use of rigid wheels, every one of whose critical speeds lies well above each of the impulses in the turbine which may stimulate that vibration, and by flexible wheels, where the curve of possible impulses passes between two critical speeds of the wheels.

Of these two methods, that of the rigid wheel is the simpler as this method permits of any desired margin of safety, and, as the effect of rotation is always to increase the frequency for any mode, with the rigid-wheel information as to the magnitude of this increase need only be approximate, and it is only necessary to carry out tests on stationary wheels.

With flexible wheels the margin of safety is necessarily limited to half the range between the critical speeds lying above and below the impulse line. As this range is small it is necessary to know with accuracy the effect of rotation, and also of such factors as temperature.

The determination of the natural frequencies for the various modes of vibration is most readily accomplished by testing, the analytical method developed by Stodola involving very lengthy calculations on account of the number of variables.

For any mode of vibration the natural frequency of a rotating wheel depends on the rigidity of the wheel itself (as in the case of a stationary wheel) and the stiffening effect of centrifugal forces due to rotation. In the case of the usual turbine wheels, the former is by far the more important factor. The expression for a rotating wheel for any particular mode may be written :

$$f_r^2 = f_t^2 + Bn^2;$$

where f_r is the natural frequency of the rotating wheel,
f_t is the natural frequency of same wheel not rotating,
n is the speed of rotation.

For the coefficient 'B' Campbell suggests values from 2 to 3 as a result of extensive experimental work.*

As the periodic disturbing impulses likely to cause wheel vibration are mostly stationary in space it is not surprising that most cases of wheel vibration have been the result of wave forms travelling backward in the rotating wheel and stationary in space. For such vibrations it may readily be shown that the frequency of the periodic disturbances for 1, 2, 3, etc., nodal diameters is once, twice, three times, etc., the speed of rotation of the machine. Thus, for example, a speed of 3,000 r.p.m. would be unsuitable for a wheel whose natural frequency for four-diameter vibration was 200 per sec.

Blade Vibration.

Vibrations of moving blades are usually classified as tangential when the vibration occurs approximately in the plane of the wheel, or axial when the vibration is normal to the plane of the wheel.

Tangential vibration at the fundamental frequency has been mainly responsible for blade vibration troubles in steam turbines, the blades vibrating at a frequency which is a small, simple multiple of the speed of rotation of the machine. As a first approximation, the natural frequency of vibration may be calculated from the usual formula for a cantilever (Table XI.), but the vibration will be modified by a number of causes, such as the degree of rigidity of the attachment to the wheel or drum, the accuracy of manufacture of the blades themselves, and the stiffening effect of centrifugal forces due to rotation. Where blades are shrouded or laced in packets the blades in each packet will normally vibrate in phase, and the consequent bending of shrouding or lacing wire has the effect of increasing the frequency of vibration. The influence on blade frequency of the number of blades tied together has been investigated by Heckmann,† who states that the frequency is progressively increased up to four blades, but that beyond this point, up to twenty blades, the further increase is negligible.

Tangential vibrations at frequencies of the first or second harmonics have been known to cause trouble in some few cases, and here the rapidity of the stress reversals consequent on the high frequencies involved, usually causes failure by fatigue after very short periods of running. In such vibrations the stiffening effect of rotation is usually small, and in those cases where the disturbing impulses necessitate the vibration of blades out of phase with others in the same packet, vibration may be largely prevented by the introduction of a lacing wire at the point of maximum amplitude.

Axial blade vibration of the blades may be regarded as the limiting case of the vibration of a bladed wheel, such as would occur, for example, with blades attached to a wheel of infinite rigidity. In the normal case it is probable that the wheel itself takes some part in the vibration, and difficulties must be avoided by methods similar to those employed in dealing with wheel vibration.

In blade vibration, as in wheel vibration, the number of factors involved makes it difficult to determine the natural frequency with any accuracy by calculation alone : reliable figures can only be obtained by tests on the blades themselves.

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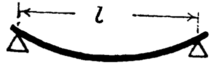
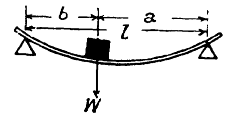
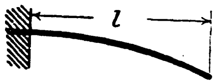
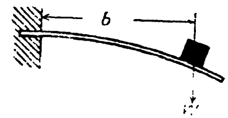
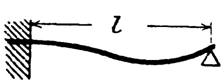
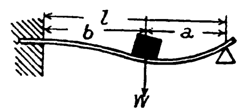
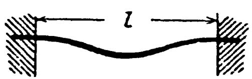
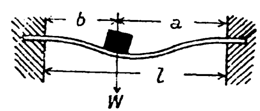
* Amer. Soc. Mech. Eng., May 1924.

† *Tangential Vibration of Steam Turbine Buckets*, Campbell and Heckman. A.S.M.E., Dec. 1925.

TABLE XI.

Uniform Shaft with Distributed Load.		Concentrated Load Carried by a Weightless Shaft.	
Method of Support.	Critical Speed in R.P.M.	Method of Support.	Critical Speed in R.P.M.
	$N = \frac{1,853}{l^2}\sqrt{\frac{E\,.\,I}{w}}$		$N = 325\sqrt{\frac{E\,.\,I\,.\,l}{W\,.\,a^2\,.\,b^2}}$
	$N = \frac{662}{l^2}\sqrt{\frac{E\,.\,I}{w}}$		$N = 325\sqrt{\frac{E\,.\,I}{W\,.\,b^3}}$
	$N = \frac{2,960}{l^2}\sqrt{\frac{E\,.\,I}{w}}$		$N = 246\sqrt{\frac{E\,.\,I\,.\,l^3}{W\,.\,a^3\,.\,b^3}}$ Approximate; holds if load near centre.
	$N = \frac{4,200}{l^2}\sqrt{\frac{E\,.\,I}{w}}$		$N = 325\sqrt{\frac{E\,.\,I\,.\,l^3}{W\,.\,a^3\,.\,b^3}}$ Approximate; holds if load near centre.

N = Critical Speed in R.P.M.
w = Weight of Shaft or other Distributed Load in lbs. per inch run.
W = Weight of Concentrated Load in lbs.
a, b, or l = Length of Span in ins.

E = Young's Modulus of Elasticity in lbs. per sq. in.
I = Transverse Moment of Inertia in inch units = $\frac{\pi d^4}{64}$ for Round Shafts.
d = Diam. of Shaft in ins.

Note Units employed.

MATERIALS FOR TURBINE CONSTRUCTION.

The use of such high peripheral speeds as are common in turbine practice introduced a new consideration into machine design; because, unlike the conditions existing in reciprocating engine parts, the stresses in the various members are in the main due to centrifugal forces resulting from their own rotation. The stresses in a given turbine will, therefore, be proportional to the square of the numbers of revolutions per minute, and to the density of the material.

With an increase in the possible speed of rotation of a given turbine, the efficiency will in general be improved, since the average ratio $\frac{U}{C_0}$ in the various stages will be increased, or a greater heat drop can be utilised with the original efficiency.

On the other hand, the available heat drop per stage might be increased so that the required number of stages for a given efficiency can be correspondingly reduced, or with the same number of stages the mean diameter of the blades can be reduced inversely as the revolutions per minute, both of which alternatives would result in a cheaper turbine.

Hence, it will be seen that in turbine work it is eminently desirable to use the highest possible peripheral speeds, the limit of which will be determined by the ultimate strength of the material employed.

British Electrical and Allied Manufacturers Association's Specifications for Turbine Forgings, Etc.

The following particulars have been extracted from the Association's specifications for forgings for turbine discs, shafts, and general forgings for use on land turbines.

The following table gives the minimum physical properties acceptable for each of four grades of materials:

Grade	I.			II.			III.			IV.		
*	L.	T.	R.	L.	T.	R.	L.	T.	R.	L.	T.	R.
Minimum ult. tensile tons per sq. in.	30	30	30	35	35	35	40	40	40	48	48	48
Yield point .	15	15	15	17½	17½	17½	20	20	20	30	30	30
Minimum elongation per cent.	30	23	20	25	20	18	22	18	16	17	14	12
Minimum reduction in area, per cent.	45	40	38	40	35	32	35	32	30	40	35	30
Bending Angle .	180°	160°	150°	180°	150°	135°	180°	150°	120°	180°	130°	100°
Test radius .	⅜″	⅜″	⅜″	½″	½″	½″	¾″	¾″	¾″	¾″	¾″	¾″

* L.—Longitudinal. T.—Tangential. R.—Radial.

Grades Nos. I., II., and III. are for carbon steels, whereas grade IV. applies to an alloy steel. A margin of 3 per cent. below the reductions in area given in the above table is allowed to the steel maker providing this is compensated by a corresponding increase of 3 per cent. on the elongation.

Bend Tests.

A cold bend test is made upon test pieces having a rectangular section of ⅞ in. by ⅜ in. These test pieces shall be machined and the edges rounded to a radius of $\frac{1}{32}$nd. The test pieces shall be bent over the thinner section, and the bending shall be done by pressure or by blows. The test pieces must withstand being bent around the radii and through the angles given in the table without fracture.

Additional Test before Rejection.

If either tensile or bending test fail to fulfil the test requirements, and the inspector agrees that the test piece does not clearly represent the quality of the material, two duplicate specimens may, if the maker wishes, be tested, and if the results from both are satisfactory the quality of the material shall be judged therefrom, and not from the original test which failed. If, however, either of the duplicate tests fail the material represented will not be approved, and the manufacturer shall have the option of retreating and again presenting for test.

Table XII. gives the physical properties of steels used for steam turbine blading, the data for which have been supplied by the courtesy of Messrs. Thomas Firth & Sons, of Sheffield.

The blades of Parsons turbines are commonly of brass, the usual mixture being alloy of 63 per cent. copper to 37 per cent. zinc, having an ultimate strength of 20 tons per sq. in., an elastic limit of 12 tons per sq. in., and an elongation of 40–50 per cent.

The blades of impulse turbines are usually made of 5 per cent. nickel steel, although bronze blades are sometimes called for.

The physical properties of bronze are greatly dependent on the treatment to which it may have been subjected, cold working having the effect of raising the ultimate strength and reducing the ductility.

Such an increase in the ultimate strength is to a large extent artificial, the material returning to its virgin state under the influence of high temperature or vibration.

The importance of this effect will be emphasised by the figures published by M. Breuil * and reproduced in Table XIII., respecting the effect of temperature on some materials used for blading.

TABLE XII.—PHYSICAL PROPERTIES OF STEELS USED IN STEAM TURBINE CONSTRUCTION

Nature of Material.	Yield Point, Tons per sq. in.	Ultimate Strength, Tons per sq. in.	Elongation on 2-in. Gauge Points.	Purpose for which Material is Employed.	Remarks.
			Per Cent.		
5 per cent. nickel steel sheets	25	40	20	Turbine blades and shrouding strips supplied mild in sheets for stamping and bending	—
5 per cent. nickel steel sheets	—	35–42	20–24	Turbine blades and shrouding strips	Higher carbon content than above
5 per cent. nickel steel bars	30–35	40–45	25	To be drawn into blade sections	—

For some years so-called rustless irons and steels have been used for turbine blades. These materials can now be regarded as of proved usefulness for turbine blading, owing not only to their comparatively high resistance to erosion and corrosion, but to their attractive mechanical properties.

Rustless steels are a chromium alloy, usually for turbine blading of composition within the following limits : carbon, 0·10 to 0·40 per cent.; chromium, 12 to 15 per cent. Ultimate strength, 40 to 50 tons/sq. in.; yield point, 30 tons/sq. in; elongation, 25 per cent.; izod test, 25 ft./lbs.

Most non-ferrous alloys suffer under high temperatures in a loss of mechanical strength and other desirable physical properties.

The following table illustrates the effect of temperature on some typical materials :

TABLE XIII.—NON-FERROUS MATERIALS FOR TURBINE BLADING.

Nature of Material.	Elastic Limit, Tons per Sq. In.	Ultimate Strength. Tons per Sq. In.	Elongation.	Temperature.
			Per Cent	
70 per cent. copper, 28 per cent. pure zinc	34·9	36·2	11	Ordinary
	34·3	36·2	8·5	100° C.
	32·7	34·3	11	200°
	22·2	24·1	30	300°
	7·6	12·7	75	400°
85 per cent. copper, 15 per cent. manganese	38·7	41·3	10·2	Ordinary
	36·8	40·0	10·5	100° C.
	37·5	41·0	11·0	200°
	33·0	36·8	21·0	300°
	21·0	24·1	68·0	400°

* *Engineering*, July 5, 1912.

From this it follows that before figures relating to the strengths of alloys can be accepted as working data the effect of temperature and the previous history of the specimens fractured in the test should be known.

Phosphor bronze has been largely employed for impulse turbine blading when conditions were such that abnormal corrosion was expected, and gives satisfactory results with dry steam where the steam velocities are moderate and the stresses induced in the blades relatively small. The following represent the average properties of this material as given by Mr. W. B. Parker.*

Maximum tensile stress . .	Not under 24 tons per sq. in.
Elastic limit	" 12 "
Elongation on 2 ins. . .	13 per cent.
Modulus of elasticity . .	16,000,000 to 17,800,000 lbs. per sq. in. within a stress range of 0 to 10 tons per sq. in.

The average composition is as follows:

Tin	From 3 to 5 per cent.
Copper	From 94·9 to 96·9 per cent.
Phosphorus	Not less than ·1 per cent.

Nickel-copper alloys are now being used for the manufacture of turbine blading, the best known being Monel metal (*Henry Wiggin & Co., Ltd.*). This metal when drawn into the form of turbine blading has the following properties:

	At 60° F.	At 750° F.
Ultimate stress . . .	41·9 tons per sq. in.	36·4 tons per sq. in.
Yield point . . .	37·7 "	31·2 "
Elongation on 2 ins. .	24 per cent.	16·5 per cent.
Reduction of area . .	60 "	60 "

Modulus of elasticity: 22,500,000 lbs. per sq. in. over a stress range of 1 to 15 tons per sq. in.

The composition (tests by the National Physical Laboratory) is as follows:

Nickel	67%
Copper	28%
Iron	1–2%
Manganese	1–2%
Lead, tin, and antimony . . .	Nil.
Also silicon and carbon.	

Properties of Steel at High Temperatures.

The use of higher initial pressures and temperatures in steam turbines (and boilers) has been seriously hindered by lack of precise information on the behaviour of materials, particularly steels, at high temperatures. During recent years a good deal of work has been done on the subject, and though much still remains to be investigated, it is now possible to indicate, in general terms, the effect of high temperatures on the more important physical properties.

Ultimate and Yield Strengths.

The curves in figs. 27 and 28, due to Mr. K. Baumann,† show the ultimate tensile stress and yield point, or the limit of proportionality, for a 0·15 per cent. carbon steel and for an alloy steel, at various temperatures. It will be noted that the limit of proportionality of the carbon steel diminishes very rapidly at temperatures beyond 300° F., whilst the yield stress of the alloy steel falls off very little up to about 700° F., indicating that this steel is suitable for use at high temperatures. Mr. Baumann proposes a factor of safety of at least 2 on the proportional limit or 3 on the yield stress.

* *Steam Turbine Blading*, Institute of Metals, September 1915.

† *Some Considerations Affecting the Future Development of the Steam Cycle*, I.Mech.E., 1930.

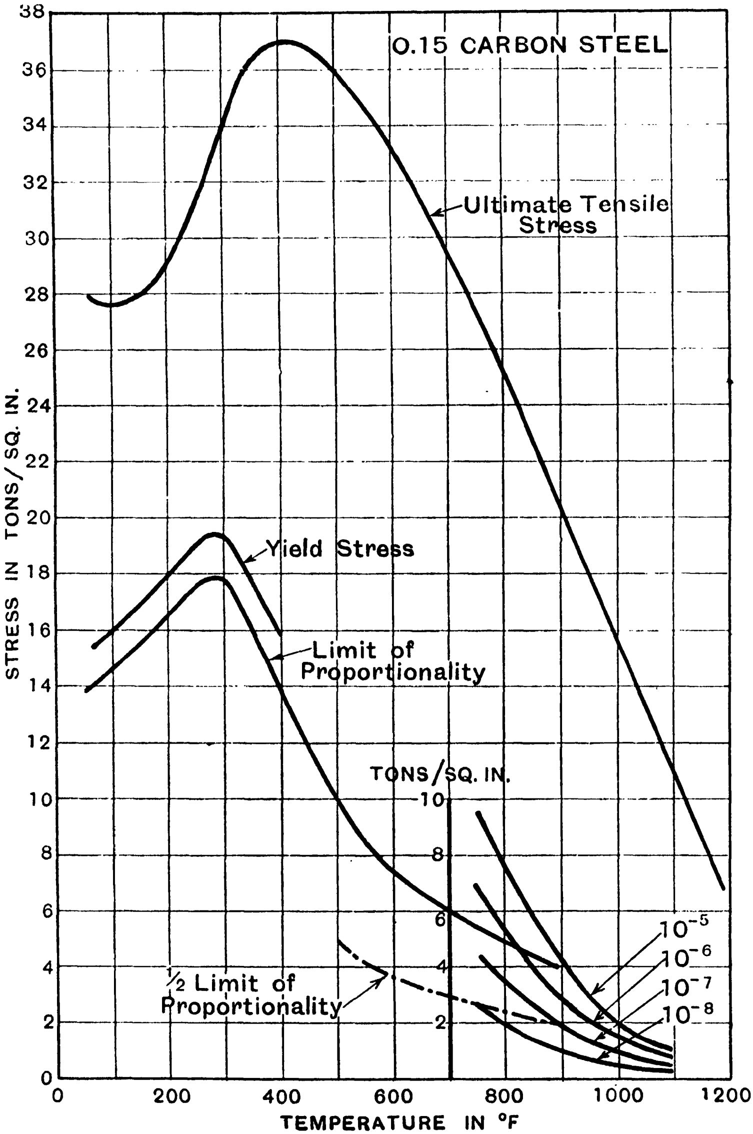

0.15 CARBON STEEL
Ultimate Tensile Stress
Yield Stress
Limit of Proportionality
½ Limit of Proportionality
TONS/SQ. IN.
10
8
6
4
2
10^{-5}
10^{-6}
10^{-7}
10^{-8}
STRESS IN TONS/SQ. IN.
38
36
34
32
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
0
0
200
400
600
800
1000
1200
TEMPERATURE IN °F

FIG. 27.

CREEP.

It has become recognised that materials sufficiently stressed at elevated temperatures extend continuously or 'creep.' The rate of creep depends on the stress and temperature to which the material is subjected and the character of the material itself. In order to ensure safety it is necessary to keep the rates of creep below certain defined amounts. It seems possible that creep occurs with stress at all temperatures, but increases very rapidly and at an increasing rate, with

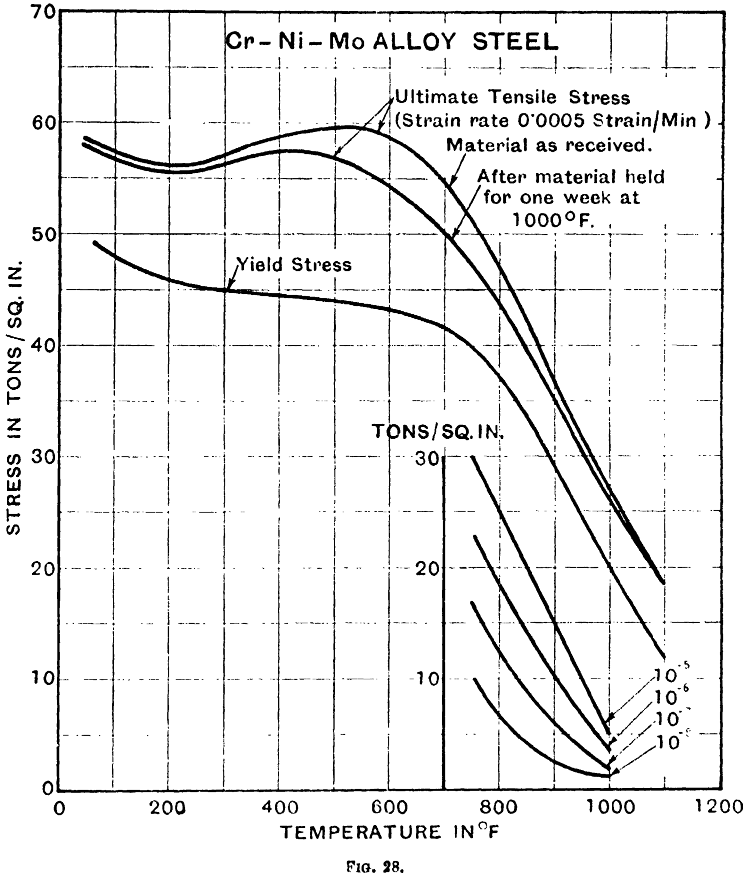


FIG. 28.

temperature and stress. It follows, therefore, that to maintain at higher temperatures the factors of safety on which design experience has been accumulated, it is necessary either to reduce the stresses to which the material is subjected, or to utilise materials known to possess high resistance to creep.

Fig. 29, produced by Mr. R. W. Bailey * from the results of his own tests and those of other investigators, shows the comparative creep resistance of medium carbon steels at temperatures from 750° F. to 1000° F., and indicates the reduction in stress necessary to maintain, for a given steel, the creep rates obtaining at 750° F.

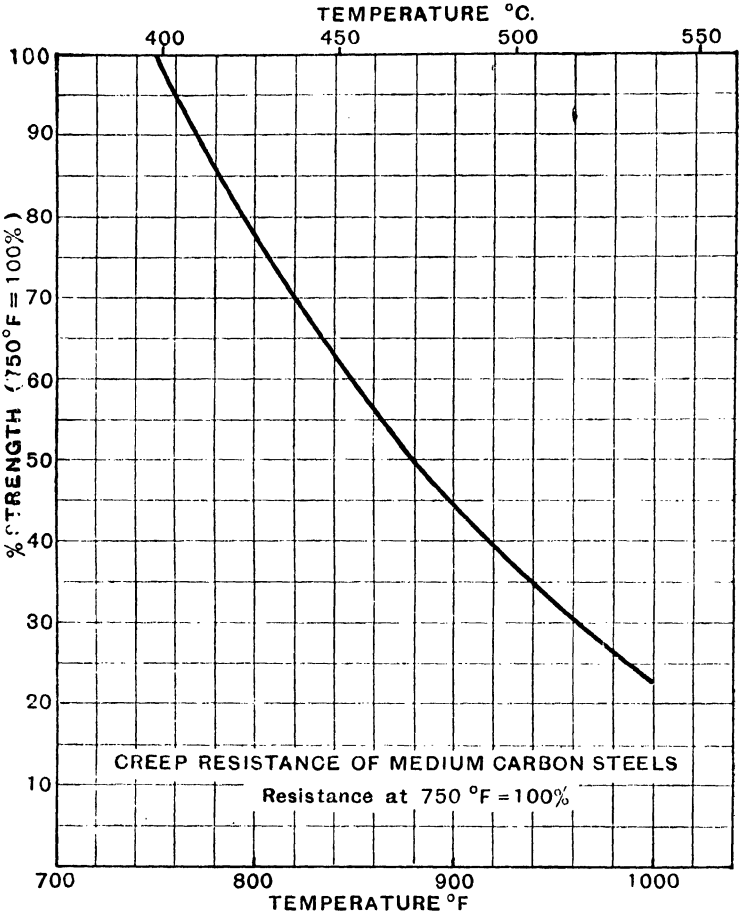


FIG. 29.

In figs. 29 and 30 are shown also the probable creep strengths of the materials cited, at the higher temperatures, for creep rates of 10^{-8} to 10^{-5}. It should be noted, however, that these curves cannot yet be regarded as established but represent an intelligent anticipation of the creep properties of the materials, from which the actual properties will probably not differ greatly.

* 'Creep of Steel under Simple and Compound Stresses.' World Power Conference, Tokyo, 1929.

The following permissible creep rates are suggested by Baumann * for various parts of steam plant.

Turbine discs pressed on shafts	10^{-9}
Bolted flanges, turbine cylinders	10^{-8}
Steam piping, welded joints, boiler tubes	10^{-7}
Superheater tubes	10^{-6}–10^{-5}

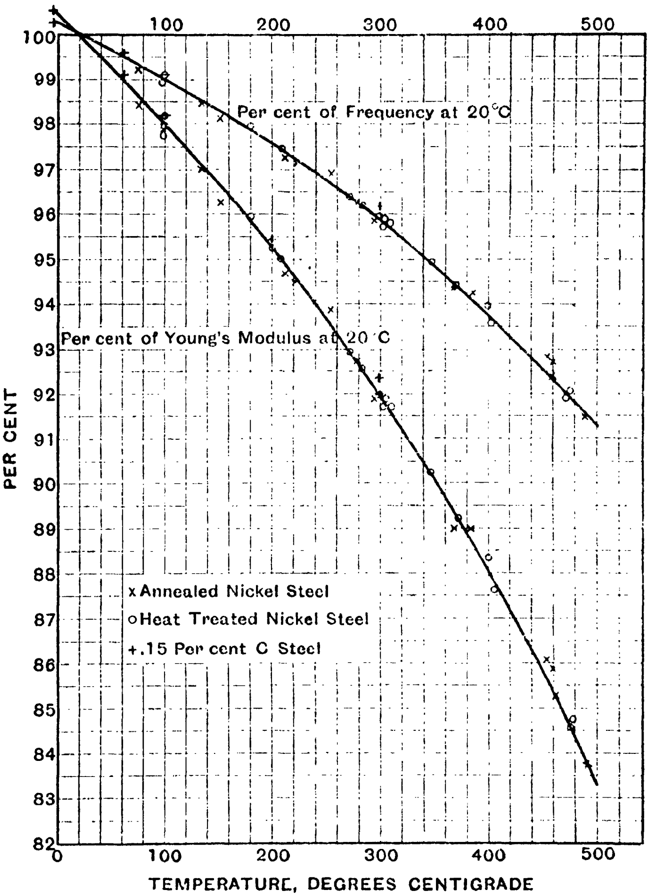


FIG. 30

It will be evident from figs. 27 and 28 that whilst in some cases permissible stresses will be limited by considerations of creep, in other cases the limit will be imposed by the necessary factors of safety on yield stress or proportional limit.

* *Loc. cit.*

Young's Modulus.

The variation with temperature of the Modulus of Elasticity, E, is shown for various steels in fig. 30,* which gives also the corresponding change in natural frequency of vibration. At 900° F. (480° C.), for example, the value of E is only 85 per cent. of its value at 60° F., so that deflection will be 18 per cent. greater and critical speeds 8 per cent. less than at 60° F.

Lubrication System.

In all cases except where ring lubrication is employed for small turbines, an oil pump is provided to maintain a pressure of about 10 lbs. per sq. in. in the lubricating system. It should be emphasised that the actual oil pressure is of secondary importance, the prime object being to flush the bearings with a copious flood of oil, and that for this purpose 2 lbs. per sq. in. may be sufficient in a properly designed system. The oil on leaving the pump passes through a cooler of sufficient capacity to maintain the temperature of the oil leaving the bearings at about 60–68° F. above that of the inlet cooling water, bearing temperatures of 150° F. being quite permissible. The main oil pump is usually driven by gearing from the turbine shaft, and the normal pressure is reached when the turbine motor is running at about one-third the normal speed. An auxiliary pump, hand-driven in the case of small turbines, but usually steam-driven, and controlled by an automatic regulator for larger machines, is usually provided to supply oil under pressure during starting up or shutting down, when the speed of the set is less than one-third the normal. An oil strainer should be placed on the suction side of the pumps to prevent grit being circulated. Two to three gallons of oil should be taken out of the tank daily and carefully filtered, the level of the tank being kept constant by making up with freshly filtered oil or, when necessary, new oil. Once in each twelve months the oil tank and all accessible portions of the lubricating system should be emptied and cleaned; the tank should then be filled sufficiently with paraffin oil to enable the whole system to be flushed by operating the hand pump. The paraffin oil dissolves any gummy matter which may have deposited from the lubricating oil and thoroughly flushes away all dirt to the oil tank; the paraffin can then be removed and new or freshly filtered oil put in. These simple precautions will result in a real economy in oil consumption and a freedom from troubles and maintenance charges which will amply repay the trouble involved.

There is a large number of lubricating oils supplied by makers of standing, all of which will give satisfactory results in operation. For turbines driving geared installations an oil to the following specification has been found to give satisfactory results:

Heavy Turbine Oil.

The specific gravity at 60° F. should be from 0·89 minimum to 0·91 maximum.

The flash point taken by a closed test should not be less than 390° F.

The viscosity measured with a Redwood viscometer should be within the following limits:

At 68° F.	Minimum 650 seconds.	Maximum 800 seconds.
At 140° F.	„ 90 „	„ 100 „

The acidity expressed as SO_3 should not exceed 0·005 per cent.

One of the most important properties lies in the freedom of a suitable oil from a tendency to form an emulsion. Suitable oils should satisfactorily pass the following test:

When steam is passed into 100 c.c. of oil until the total volume of the condensate and oil measures 200 c.c., the whole of the oil shall separate out within ten minutes, leaving the water clear.

The sludge value of the oil carried out, according to Dr. Michie's test should not exceed 1·2 per cent.

The following specification would apply to oils suitable for steam turbines driving generators, etc., direct:

Light Turbine Oil.

The specific gravity at 60° F. shall be 0·87 minimum to 0·89 maximum.

The flash point of the oil measured by a closed test shall not be less than 365° F.

The viscosity measured with a Redwood viscometer shall conform to the following:

At 69° F.	Minimum 300 seconds.	Maximum 400 seconds.
At 140° F.	„ 60 „	„ 75 „

The acidity expressed as SO_3 shall not exceed 0·005 per cent.

When steam is passed into 100 c.c. of oil until the total volume of condensate and oil measures 200 c.c., the whole of the oil shall separate out within five minutes, leaving the water clear.

The sludge value, measured by Dr. Michie's test, shall not exceed 1·2 per cent.

* Kimball & Lovell. American Physical Society, *Physical Review*, vol. 26, No. 1, July 1925.

PROCEDURE IN STARTING UP LARGE TURBINES.

While the general guidance given in the following remarks is, in the main, intended to apply to relatively large units (say turbines of 10,000 kw. capacity and larger), the procedure is an excellent one to follow on smaller sets.

The time intervals indicated can be reduced as the size of the set is reduced.

First operation.—Start up condensing plant and obtain 20 ins. to 24 ins. vacuum.

Second operation.—To warm up turbine, open stop valve sufficiently to start it rotating, then close it down until a steady speed of not more than 10 per cent. of the normal speed is obtained. Maintain this condition for ten to fifteen minutes during which vacuum is increased.

Third operation.—Increase the speed by gradual and steady increments at a rate of from 5 per cent. to 10 per cent. of normal speed per minute.

Fourth operation.—Synchronise set and switch on to bus bars.

Fifth operation.—Build up load on set at a steady rate of from 5 to 10 per cent. of normal load per minute.

The procedure here outlined is a safe one to be recommended in normal practice; in an emergency the time intervals can, of course, be reduced.

BRITISH STANDARD SPECIFICATION FOR STEAM TURBINES.*

No. 132 (Abstract).

RATING.

A steam turbine shall be capable of giving continuously a specified output when running under specified conditions as regards speed, steam pressure, temperature and exhaust pressure.

OUTPUT OF THE TURBINE WHEN WORKING NON-CONDENSING.

If the specified steam pressure is not less than 165 lbs. per sq. in. absolute, and the rated output of the turbine is not more than 3,000 kw., the turbine shall be capable of developing 30 per cent. of its rated output when exhausting to the atmosphere. For turbines of larger capacity the output must be settled between the purchaser and the manufacturer.

Note.—It is undesirable to work high-pressure steam turbines exhausting to the atmosphere, and this should only be done in cases of emergency. Even under this condition, the turbine should not operate in this way for a longer time than is absolutely necessary.

MAXIMUM SPEED.

The turbine shall be capable of withstanding for five minutes, without injury, a speed 15 per cent. in excess of the rated speed.

GOVERNING CHARACTERISTICS.

When the rated load is taken off or on the permanent speed variation shall not exceed 4 per cent.

SPEED ADJUSTMENT.

Means shall be provided to enable the speed to be adjusted by plus or minus 5 per cent. when working under synchronising conditions.

EMERGENCY CUT-OFF SPEED.

The emergency governor shall be set to shut off steam if the speed exceeds 9 to 11 per cent of the rated speed. When operating at rated speed and the rated load is thrown off the maximum variation in speed shall not be sufficient to bring the emergency governor into operation.

ECONOMIC RATING OF THE TURBINE.

The turbine shall be designed to give a minimum steam consumption at 80 per cent. of its rated output.

STEAM (OR HEAT) CONSUMPTIONS.

If steam (or heat) consumptions are guaranteed for various outputs, the average steam consumption shall be obtained by

Multiplying the consumption at the rated output by b_1.
" " 80 per cent. of the rated output by b_2.
" " 60 " " " b_3.
" " 40 " " " b_4.

* By permission of the British Standards Institution.

and dividing the sum by $b_1 + b_2 + b_3 + b_4$, the multiplying factors to be specified by the purchaser or mutually agreed.

The steam (or heat) consumption may be requested and furnished either for some specified load or for a series of loads, but the guarantee shall be confined to the one specified load or to the weighted consumption as above.

Tolerance.

For the purpose of comparing test consumption with the guarantee performance a tolerance calculated as follows shall be allowed :—

Percentage tolerance (+ or −) =

$$\frac{\text{Arithmetical sum of percentage corrections} - 5}{4} + 2\tfrac{1}{2} \text{ per cent.}$$

Hydraulic Test for Parts Exposed to Full Boiler Pressure.

All parts exposed to the rated steam pressure shall be tested with a hydraulic pressure 50 per cent. in excess of the rated pressure. The machine as a whole shall be tested with a hydraulic pressure of 15 lbs. per sq. in. gauge.

Critical Speed of Rotating Element.

The critical speed of the turbine shaft shall not be within 20 per cent. above or 20 per cent. below the rated speed.

TURBINE SPEED-REDUCTION GEAR.

The speed at which it is most desirable to run a turbine is limited only by the strengths of the materials used in its construction. Unfortunately this limit of speed cannot be approached when the turbine is employed for direct driving of propellers, D.C. generators of large output, rolling mills, line shafting, etc., and in such cases where comparatively low speeds are essential not only does the turbine become heavy, costly and inefficient, but direct turbine drive is often quite out of the question.

For this reason several systems of speed reduction have been introduced, and are being successfully applied.

Double Helical Gears.

Speed reduction by means of double helical gears was employed in the earliest days of turbine engineering by Dr. Delaval, whose turbines, running at speeds of the order of 10,000 r.p.m., could not be directly coupled to electric generators. The gearing of these turbines gave remarkably little trouble, and although the outputs were low it is notable that even to-day this is the most successful system of speed reduction.

Double helical gearing for large turbines was introduced by Sir C. A. Parsons, who with characteristic courage first introduced the system in the famous case of the propelling machinery of the s.s. 'Vespasian,' and the drive of a rolling mill at Calderbank Steel Works.

In the earlier installations difficulties were experienced through minute inaccuracies of the gear teeth, which gave rise to very noisy working. This was traced to errors in the gearing of the table of the gear-cutting machine by Sir C. A. Parsons, who has very materially reduced the trouble by arranging a 'Creeping' table giving the gear blank an advance of 1 per cent. relatively to the main table of the machine. The improvements in modern gear-cutting machines have eliminated the inaccuracies referred to, so that the creeping device is no longer essential.

The helix angle adopted varies from 20° to 45°, the usual angle in modern practice being 30°.

The teeth of modern double helical gears are of the standard involute type, the pressure angle adopted by different makers varying from $14\frac{1}{2}$° to 20°.

If p = average pressure in lbs./inch tooth face.
 L = total length of tooth face in inches.
 D = diam. of pitch circle of pinion in inches.
 R = revs. per minute of pinion.

$$V = \text{pitch speed in feet per second} = \frac{\pi D}{12} \times \frac{R}{60} = \frac{\pi D}{720} \times R.$$

Then

$$\text{Horse-power transmitted} = P = \frac{pLV \times 60}{33{,}000} = \frac{pLV}{550}$$

Dr. Stoney states that V should be kept between the limits 100 and 130 ft./sec. He also gives the maximum permissible pressure $p = 120d$ for pinions smaller than 5 inches diameter, and $p = 175\sqrt{d}$ for larger pinions.

Fig. 31 shows curves co-ordinating p and d, published by Mr. F. H. Hodgkinson *Engineering*, Jan. 10, 1919).

With pinions having two bearings where E = Young's modulus of elasticity = 30×10 lbs./sq. inch

$$I = \text{the moment of inertia of the pinion} = \frac{\pi D^4}{64}$$

$$\text{Transverse deflection at the centre} = \delta_1 = \frac{5}{384} \cdot \frac{pL^4}{EI}.$$

This is one of the determining factors in design, and, according to Stoney, should not exceed 0·0002 inch.

For pinions having three bearings, the maximum deflection on each portion of the pinion between two bearings should not exceed 0·0002 inch, and in most cases this is not the condition determining the diameter.

Another condition fixing the design of the pinion arises from the fact that the torsional deflection opens the teeth, thus preventing uniform distribution of pressure on them. This torsional opening, δ_2, according to Stoney, must be limited to less than 0·0001 inch, and is given by the following expressions:

$$\text{For two or three bearing pinions } \delta_2 = \frac{CL}{10 \times 10^4} \text{ inches over one helix.}$$

In the Melville-McAlpine gear, manufactured and developed by the Westinghouse Machine Co., the pinion is supported in a floating frame which permits its axis to take up a slight inclina-

DOUBLE HELICAL GEAR CURVES.

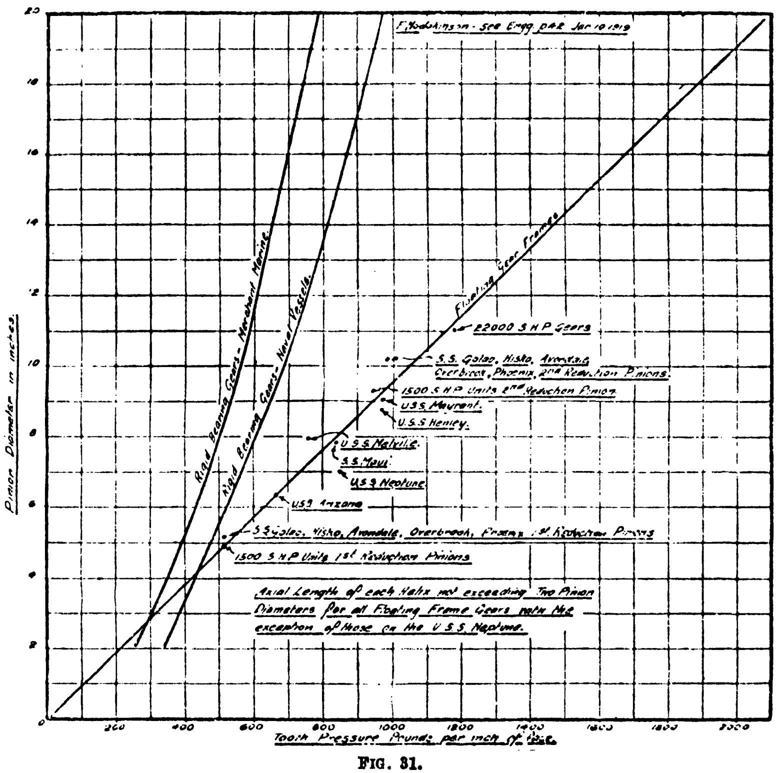


FIG. 31.

tion in the vertical plane relatively to the slow speed shaft. The torque on the pinion tends to increase the helix angle of one half of the pinion and decrease that of the other half, so that if the helix angle of the pinion is the same as that of the wheel, as load comes on, the helix angles become different. If the axis of the pinion is inclined in a vertical plane relatively to the wheel, the change in the helix angle with load can, it is claimed, be very largely compensated, so that with the floating pinion larger torques can be transmitted through a given size pinion than through a pinion supported in rigid bearings. According to the inventors, the floating frame also corrects to a certain extent errors in the teeth and in alignment of the gears.

The action of this gear is well explained, together with some authoritative notes on double helical gearing, in a series of articles by Mr. J. H. McAlpine, which appeared in *Engineering*, commencing May 5, 1916.

In these articles Mr. McAlpine suggests the use of the 'Power Constant' for comparing different gears; these he defines as follows:

If

D = pitch diameter of pinion in ins.; L = length of tooth face in ins.; p = average pressure in lbs. per in. of tooth face; R = r.p.m.; O = power constant; P = horse-power transmitted:

Then

$$P = \frac{\pi D R p L}{33000 \times 12}.$$

For correctly designed gears, p and L vary as D, therefore

$$P = O (D^3 R),$$

and

$$O = \frac{P \times 1000}{D^3 R}.$$

For gears having the pinion supported in two rigid bearings, the power constant 'O' should not exceed 1·5.

'O' is usually from 1·2 to 3·0, for gears having three bearings on the pinion.

Gears having the pinion supported in a floating frame have worked very successfully with a power constant of 4.

The ratio $\frac{L}{D}$ should preferably be equal to 4 in gears having the floating frame.

For best results the pinions for all sizes should have from 26 to 30 teeth; if the number is appreciably reduced interference takes place.

The efficiency in double helical gearing varies from 98·5 per cent. in large gears to 97·5 per cent. in small sets, so that from 1·5 per cent. to 2·5 per cent. of the heat equivalent of the power transmitted has to be dissipated in the lubricating oil cooling system.

The gears are usually lubricated by spraying the oil on the teeth at the point of contact.

The success of a gear depends to a larger extent than is generally supposed on the rigidity and correct design of the gear-box and lubricating system, and where experienced engineers are responsible for these details every confidence can be placed in a turbo-geared proposition.

Rapid advances have been made in the application of steam turbines to the mercantile marine, by the introduction of double reduction gearing. Single reduction gears having ratios larger than 15 to 1 are scarcely practical for ship propulsion, but a reduction of, say, 42 to 1 can be commercially obtained by a double reduction gear having two trains of wheels, the first having a reduction of, say, 6 to 1 and the second 7 to 1.

Such double reduction gears are now being used having efficiencies at full load of about 97 per cent.

Now that such large ratios of reduction are commercially practicable, both the turbine and the propeller can be run at the speed most suitable for each respectively.

See also p. 996 (Vol. I), and 'Reduction Gear for Propulsion,' p. 290.

The possibilities and limitations of this type of transmission are illustrated by the following figures given by Prof. Fottinger.*

No. of Stages.	Gear Ratio.	Estimated Efficiency.
1	5 to 1	90 per cent.
2	7½ to 1	90 ,,
3	10 or 12 to 1	88

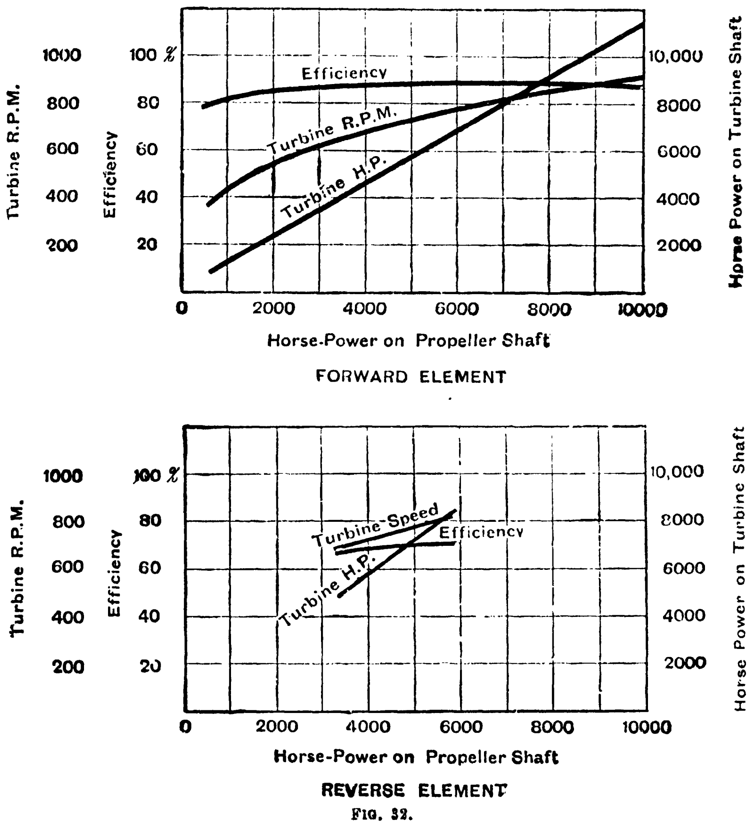


FIG. 32.

Unfortunately the ratios usually required for marine purposes are from 10 to 1 up to 20 to 1 and as it is doubtful if more than single stage transmitters could be commercially adopted, it will be seen the field open to this type is limited. A distinct advantage lies in the fact that the discharge from the turbine pump element can be diverted to a reverse water turbine, by which means reversal can be easily and economically effected. The water used in the transmitter is usually the boiler feed, and in this way some of the losses can be recovered in feed water heating, but it is difficult to see how the gain can possibly exceed 1 per cent. measured at the driven shaft.

Fig. 32 shows the results of tests on a 10,000 B.H.P. transmitter.†

* *Engineering*, March 14, 1913.

† *Z. V. d. I.*, May 16, 1913.

In Table XIV. are given some particulars of machines typical of modern large turbine practice. The table is based largely on that given by Gibb in 'Post-War Land Turbine Development,'* and it is pointed out by the author that in most cases the heat consumption must be regarded as approximate only, since in a large number of cases insufficient information is available and certain assumptions have had to be made.

A comprehensive analysis of particulars of large turbines by the leading manufacturers is given by Baumann in 'Neuere grosse Dampfturbinen.' (*V.D.I.*, June 14, 1930.)

No.	Year.	Station.	Output maximum Continuous kW.	Speed. R.p.m.	Type.	Stop Valve Pressure. Lb. per sq. in. g.	Stop Valve Temperature. ° F.	Vacuum. Ins. Hg.	Heat Drop S.V.—Vac. ° F. B.Th.U.	Feed Heating ° F.	Heat Consumption with Feed Heating. B.Th.U. per kW. hr.	Maker.	Remarks.
1	1917	Carville . .	10,000	2,400	Double-ended L.P.-cyl. tandem.	250	650	29·0	481	120	14,020	Parsons . .	Reheating to 400° F.
2	1918	Lots Road .	18,000	2,000	Double-ended L.P.-cyl. tandem.	185	500	29·0	399	—	14,197†	,, . .	—
3	1922	Barton . .	27,500	1,500	Single-cyl. impulse .	350	700	29·1	486	200	11,958	Metropolitan-Vic.	—
4	,,	Barking .	40,000	3,000	Cross-compound, H.P. and L.P., I.P. and L.P.	350	700	29·0	568	250	10,750	Parsons . .	Reheating to 700° F.
5	1923	Chicago .	50,000	1,800 720	Cross-compound, 3 cyls.	550	750	29·25	604	315	10,026	,, . .	Reheating to 700° F.
6	1924	N. Tees . .	25,000	2,400	Double-ended L.P.-cyl. tandem.	450	700	29·0	490	300	11,020	,, . .	—
7	,,	Philo . .	40,000	1,800	Single-cyl. impulse .	560	725	29·0	590	314	10,360	G.E. . .	Reheating to 725° F.
8	1925	Nankai . .	12,500	3,600	2-cyl. impulse, Baumann exhaust.	300	722	28·0	443	150	13,278	Metropolitan-Vic.	—
9	,,	Boston . .	35,000	1,800	—	1,200	700	350 lb. per sq. in. g.	—	—	—	G.E. . .	—
10	,,	Hudson Ave. .	80,000	1,800	Cross-compound, H.P. and L.P., I.P. and L.P.	376	700	29·0	483	361	11,000	Amer. West. .	—
11	1926	Leghorn, Italy	14,000	3,000	Single-cyl., radial flow.	330	842	29·0	519	—	12,370†	Ljungstrom .	—
12	,,	Crawford Ave.	77,000	1,800 1,200	2-cyl. cross-compound impulse.	550	750	29·0	515	—	11,790†	G.E. . .	—
13	,,	Klingenburg .	80,000	1,500	Cross-compound, H.P. and I.P. impulse, L.P. reaction.	478	752	28·8	501	280	10,740	A.E.G. . .	—
14	,,	Utrecht .	20,000	3,000	4 cyls. in line, 2 impulse, 2 reaction.	455	752	28·65	492	176	12,243	Stork . .	—
15	,,	Lakeside .	7,000	3,600	Single-cyl. impulse .	1,200	720	332 lb. per sq. in. g.	—	—	—	G.E. . .	—
16	,,	Barking .	20,000	3,000	Double-ended L.P.-cyl. tandem.	350	700	29·0	480	280	11,080	Parsons . .	—
17	1927	Barton . .	40,000	1,500	2-cyl. Baumann exh. impulse.	350	700	29	480	300	10,920	Metropolitan-Vic.	—

18	„	Hudson Ave. .	110,000	1,800	Cross-compound, H.P. and I.P. and 2 L.P.	400	700	29·0	486	270	10,620	Amer. West. .	—
19		Waukegan .	50,000	1,800	2-cyl. tandem reaction	600	725	29·0	510	—	11,700†	Allis Chalmers .	—
20	„	Witkowitz Coal Co.	19,500	3,000	4-cyl. tandem impulse	1,425	915	28·8	641	—	10,900†	Erste Brunner .	Reheating to 662° F
21	„	Powerton, Ill.	55,000	1,800	2-cyl. impulse . .	600	725	29·0	583	350	9,650	G.E. . .	Reheating to 725° F.
22	1928	Hell Gate .	165,000	1,800	2 lines, 4-cyl. imp. reaction.	266	700	29·0	467	252	11,300	Amer. West. .	—
23	„	„ „ .	160,000	1,800 1,200	Cross-compound, 2-cyl. pure reaction, double ended L.P.	265	610	29·0	442	208	11,500	Brown Boveri .	—
24	„	Moges . .	17,600	3,000	Single-cyl. impulse, Baumann Triple Exhaust.	355	707	28·5	461	—	13,527†	Metropolitan-Vic.	—
25	1929	I.C.I., Billingham.	12,500	2,400	Single-cyl. impulse .	630	833	275 lb. per sq. in. g.	—	—	—	„ „	—
26	„	Calcutta .	20,000	3,000	Single cylinder . .	195	590	28·0	389	170	13,300	Parsons . .	—
27	„	Liverpool .	50,000	1,500	2 cylinders . .	400	800	29·0	516	300	10,600	Metropolitan-Vic.	—
28	„	State Line .	208,000	1,800	3-cyl. cross-compound, 1 H.P., 2 2-flow L.P.	600	730	29·0	539	380	10,250	G.E. . .	Reheating to 500° F.
29	„	Delray, Detroit	10,000	3,600	2-cyl. tandem impulse.	365	1,000	29·0	575	300	10,260	B.T.H. . .	—
30	„	Vic. Falls, Vereeniging.	32,500	3,000	2-cyl. impulse . .	200	626	28·2	403	198	12,960	Metropolitan-Vic.	—
31	„	Sanyo Chuo .	38,500	1,800	„ „ . .	600	725	29·0	533	295	10,470	„ „	Reheating to 480° F.
32	1930	Dunstan .	50,000	1,500	Single-ended L.P. cyl. tandem.	600	800	29·0	618	340	9,280	Parsons . .	Reheating to 800° F.
33	„	Copenhagen .	36,000	1,500	Double-ended L.P. cyl. tandem.	323	743	29·0	489	284	10,845	„ . .	—
34	„	State Line	150,000	1,800	3-cyl. impulse . .	1,200	832	29·0	675	—	9,900†	G.E. . .	Reheating to 832° F.
35	„	Pekin, Ill. .	105,000	1,800	3 cylinders . .	600	725	29·0	589	—	10,550†	„ . .	Reheating to 750° F.
36	1931	State Line .	125,000	1,800	2 cylinders . .	1,200	825	29·0	672	—	9,900†	Allis Chalmers .	Reheating to 825° F.
37	„	St. Denis, Paris	50,000	3,000	4 cylinders . .	770 980	842 880	29·07	582	360	10,300†	Brown Boveri .	—
38	„	Schelle, Ant. .	60,000	3,000	3 cylinders . .	500	797	28·88	519	330	10,680	Siemens-Schuck	—
39	„	Barking .	75,000	1,500	3 cylinders . .	600	800	29·0	533	340	9,850	B.T.H. . .	—
40	1932	Thermo-Tech. Inst.‡	24,000	3,000	Single-cyl. impulse .	1,763	878	370 lb. per sq. in. g.	—	—	—	Metropolitan-Vic.	—
41	1933	Battersea .	105,000	1,500	3-cyl. double-ended L.P.	570	800	29·1	534	340	—	„ „	—

TABLE XV.—PARTICULARS OF SOME NOTABLE INSTALLATIONS OF GEARED TURBINES.

Name of Installation.	Kind, Maker.	Face Width.	Angle of Helix.	Pressure in Lbs. in W.D.	Pitch Line Speed. Ft. per Sec.	Power Constant.	Face Width ÷ Pinion. Diam.	Efficiency of Gear.	Reference.
Hope Natural Gas Company	American Westinghouse Co.	40	30°	715	65·5	2·26	4·0	—	'Engineering,' May 26, 1916.
Cleveland Elec. Illuminating Company	,,	40	30°	925	81·5	2·72	3·85	—	,,
U.S.S. Melville	,,	32	30°	720	47·5	3·02	4·12	—	,,
Cruising Gears, U.S.S. Arizona	,,	24	30°	662	69·2	3·12	3·78	—	,,
S.T.D. 1000 Gear Kw.	,,	20	30°	625	81·5	3·66	3·85	—	,,
U.S.S. Pennsylvania	,,	22	30°	526	37·75	3·99	4·57	—	,,
Swedish Cruisers T.S.	,,	31	30°	825	113·0	3·61	4·13		,,
Prop. for U.S. Torpedo Boat Destroyer W.M. Co.	,,	39	30°	1,255	95·0	3·88	3·94		,,
	,,	27	30°	530	104·0	4·03	4·07		,,
	Rigid Gear	27	—	710	129·0	1·32 H.P. 0·497 L.P.	2·51 H.P. 1·54 L.P.	—	,,
T.S. Vespasian	C. A. Parsons	24	20°	394	31·8	3·00	4·8	98·5%	'The Engineer,' April 7, 1911.
S.S. Transylvania and Tuscania	,,	About 48	—	442	70·7	1·80	5·05	—	'Engineering,' May 26, 1916.
S.S. Ciudad de Buenos Aires and Ciudad de Monte Video	,,	About 28	—	450	63·0	2·58 H.P. 1·29 L.P.	4·5 3·18	—	,,
Rolling Mill, Calderbank Steel Works	,,	—	23°	270	62·4 24·7	8·75 4·2	—	—	'The Engineer,' April 7, 1911.
Melville-McAlpine	American Westinghouse Co.	—	30°	—	92	—	—	98·99%	'Engineering,' Dec. 3, 1909.
Melville-McAlpine Geared D.C. Gen.	,,	—	30°	—	79	2·85	—	—	'Engineering,' Jan. 31, 1913.
T.S. King Orry	C. A. Parsons	—	44° 22½′	—	—	—	—	—	'Engineering,' June 27, 1913.
—	Power Plant Co.	—	—	325	100	1·49	—	—	'The Engineer,' Oct. 1, 1915.
Melville-McAlpine	American Westinghouse Co.	26·5	30°	404	110	1·86	3·93	98·5%	'Steam Turbines for Land Purposes,' Manch. Assoc. of Eng., Jan. 27, 1917.
Messrs. C. & J. Hirst	D. Brown	11	32° 12′	275	65·5	0·96	2·2	98%	
Birmingham Corporation	Power Plant Co.	12	23°	132	46·3	0·696	2·83	—	

SECTION XXVII

PART V

STEAM CONDENSERS — HEAT TRANSMISSION IN SURFACE CONDENSERS—AIR PUMPS—ROTARY AIR PUMPS—STEAM JET EJECTOR AIR PUMPS—FEED PUMPS—COOLING TOWERS—STEAM ACCUMULATORS.

(Revised by R. H. Parsons, M.I Mech E)

STEAM CONDENSERS.

The principal types of condenser are:—

(1) Jet condensers, in which the condensing water mixes with the steam.

(2) Surface condensers, in which the cooling water and the steam are kept separate.

The jet type involves much less initial cost than surface condensing plant, and requires less attention and cleaning, but requires more power to operate it. The choice of type depends to a very large extent on the nature and cost of the water available for cooling and for boiler feed.

Jet condensers are preferred to surface if the cooling water is fit for boiler feed or if a suitable boiler feed, other than the condensed steam, is available.

If ample water is available, but not suitable for boiler feed, the surface condenser is advisable. In some cases the water may be too bad even for use in surface condensers, causing choking of tubes or serious corrosion; jet condensers may then be used and boiler feed obtained from a limited supply of good water, even though the price of the good water is high.

Jet Condensers.

With low-level jet plants it is necessary that the cooling water should be drawn in by virtue of the vacuum, and an efficient vacuum breaker be fitted whereby, in the event of the pumps connected to the condenser suddenly shutting down, the vacuum will be broken, thus stopping the flow of the injection water, and preventing it from entering the L.P. cylinder.

The mixture of cooling water and condensed steam is preferably withdrawn at the bottom of the condenser, and the air is taken off near the top. This involves separate air and water pumps, but in the case of engines of moderate power having their own jet condenser one pump for both air and water is commonly used. In the barometric or elevated type of jet condenser no water extraction pump is required, as the condenser being fixed at the barometric height is self-draining. Usually, however, an injection pump is required to lift the water partly up to the condenser, as the vacuum can be utilised to a certain extent for drawing the water into the condenser the same as in the low-level plant. These condensers are more expensive than the low-level type, and, due to a long exhaust pipe being necessary, require to maintain a slightly higher vacuum for an equivalent exhaust pressure at the turbine or engine exhaust flange.

The capacity of a contra-flow jet condenser should not, according to *Sim*, be less than one-quarter of the volume of the cylinder or cylinders exhausting to it, and need not be more than one-half. For quick-running engines one-third is generally sufficient. For turbine work or continuous steam flow, 0·008 to 0·011 cub. ft. per lb. of steam per hour is reasonable.

Surface Condensers.

Cooling water should be arranged to pass through the tubes and steam to flow over the tubes. It is usual to divide the tube element in two, three, or four passes according to the design of the plant, the passes being arranged so that the hottest water meets the hottest steam.

Large spaces should be allowed between the tubes at the condenser inlet to obtain a good flow of steam into the heart of the condenser. The steam flow should always be as direct as possible, changing the direction of flow should be avoided, and to ensure an evenly distributed flow over the top of the tubes a dome should be formed on the top of the condenser body.

Internal guide plates are used by some makers, in order to distribute the entering steam uniformly over the tubes.

It is important that the drop of pressure across the groups of tubes should be reduced to a minimum. Baffles in the tube space are not recommended, as their total effect is harmful.

The pitch of the tubes should be such that they will allow sufficient area for the flow of steam without an excessive pressure drop through the condenser.

The number of tubes which can be fixed in a square foot of plate is,

$$N = \frac{166}{p^2};$$

where,

N = number of tubes per sq. ft. of plate; p = pitch of tubes in ins.

Air in the condenser must not be allowed to remain stagnant round the tubes.

Frequently a section of the condenser next the air pump suction branches is partitioned off to act as a cooler for the air passing to the air pump. The first water pass takes place through the tubes in this section. The cooling of the air reduces the volume to be dealt with by the air pump.

The condensate should be as hot as possible, and in good normal condensers, it is delivered at a temperature almost equal to that corresponding to the vacuum.

The drop of vacuum through the condenser should not exceed 0·2 in mercury.

The practice in power stations of extracting steam from the turbine for feed-heating reduces the steam passing to the condenser, and for turbines of a given output, the size of the condenser may be reduced.

A good rule for power station condensers is to allow a cooling surface in sq. ft. equal to the maximum rated capacity of the generator in K.W. This is equivalent to a heat transmission of about 5,600 B.Th.U. per sq. ft. per hour through the tubes at the economical rating of the machine.

The factors underlying the design of condensing plant were fully examined by Mr. W. T. Bottomley in a paper read before the N.E. Coast Institute of Engineers and Shipbuilders in 1941. He concluded that it was uneconomical to employ a higher velocity than 5 ft. per sec. for the water through the tubes on account of the increased pumping costs, and he maintained that, contrary to a prevalent opinion, experience showed that trouble from deposits in the tubes was not mitigated by increasing the quantity of circulating water. The only way of counteracting the formation of such deposits was by increasing the cooling surface and reducing the water velocity. The economical load on a condenser for dirty conditions should be about 5,300 B.Th.U. per sq. ft. per hour, instead of the 7,000 B.Th.U. generally recommended. When the size of the power house building was fixed, it paid to provide the largest possible cooling surface in the space available, not by reducing the pitch of the tubes but by increasing the overall dimensions of the condenser shells. Condensers working with tidal water or with cooling towers, and fitted with 1 in. tubes should have at least three passes, with four passes for the smaller machines. Only in cases where the circulating water pipes were very short, and the machines very large, could a single pass condenser be economically justified. Considerations of overall economy also indicated that the greater the length of the circulating water pipes, the less should be the quantity of cooling water, and the lower should be the designed vacuum.

The most practical operating check on the efficiency of a surface condenser is to note the difference between the temperature corresponding to the vacuum, and the outlet temperature of the circulating water. This difference is not affected by the temperature of the water at inlet and will not be affected by more than a fraction of a degree by large variations in the quantity of circulating water. Hence if the temperature differences are recorded at various loads when the condenser is known to be in perfect condition, the same differences should always be obtained at the stated loads. Any increase of the temperature difference at a given load is a sign that the condenser is getting dirty.

In order to prevent the formation of slime and growth of organisms in the tubes, chlorination of the circulating water has been introduced. It results in saving of cleaning and increasing the 'availability factor' of the plant.

Condensing Plant in Power Stations.

According to a 'Progress Review of Power Station Equipment,' by I. V. Robinson, (*Jour. Inst. Elect. Eng.*, March 1935), there is a tendency to specify a vacuum of which the corresponding temperature is only 21° F. above the inlet temperature of the circulating water.

While cast iron is still used for condenser shells, welded steel shells are very popular, with, as a rule, cast-iron water heads.

Single pass condensers remain the favourite in America but not in Europe.

Tube vibration has been largely overcome. Fine clearance in the holes of the supporting plates and uneven pitching of these plates are beneficial in preventing vibration.

Air ejectors are always of steam jet type, and except in few cases have surface inter- and after-condensers. Except for vacua below 29 in., 3-stage ejectors are generally used.

De-aerators are not much used as condensate is so well freed from air at the base of the condenser with the latest form of hotwell. Plants are now very airtight, 'a pot of paint is worth as much as a de-aerator.' In Great Britain it is established practice to instal air extractors having a dry air capacity of 1 lb. per 2,000 lb. steam, though this basis is admittedly unduly ample.

Surface Condenser Practice.—For a critical survey of practice in surface condensers and auxiliaries, see *The Engineer*, August 24 to November 23, 1934.*

Tube Plates.

These are usually of rolled brass alloy of copper 62 per cent., zinc 37 per cent., and tin 1 per cent., and are generally made of a thickness equal to tube diameter + 0·25 in.

Tubes.

These are of solid drawn brass alloy, of copper 67 per cent., zinc 32 per cent., tin 1 per cent., or where salt or corrosive water is used, of Admiralty mixture alloy, of copper 70 per cent., zinc 29 per cent., tin 1 per cent. A still better tube to use where corrosion is liable to occur is copper of 99 5 per cent. purity, but the initial cost of these is often found prohibitive. Tinned tubes should be avoided, since unless the tinning is absolutely perfect, serious pitting is liable to occur. The Admiralty practice is to adopt untinned tubes of the alloy referred to.

The unsupported length of tubes should not exceed 100 times the diameter, and where the length of the condenser exceeds this dimension, tube-supporting plates should be fixed. The tubes should be secured in the tube plates by tape packing and ferrules, the latter having an internal lip to prevent 'creeping' of the tubes.

Ferrules should allow a space for free movement of the tube ends, and should be rounded, so as to reduce to a minimum the loss of head of the water at its entry into the tubes.

According to Stanton, in a condenser of a given surface for maximum efficiency, the tube length should be as great as possible. As regards tube diameter he says that condensers will be equally efficient in which the ratio $\frac{l n^{-1}}{d}$ = constant, where l = length in feet, n = number of tubes, d = external diameter of tube in feet. Tube diameters should be chosen which offer the least aggregate resistance to flow. Resistance is, he finds, proportional to l^{4-2n}, and is therefore least when the tubes are the smallest practical diameter. He considers the best position for tubes is vertical and the water movement downward.

Tube diameters may vary from ½ in. to 1 in., but are usually ⅝ in. or ¾ in., in fact ¾ in. diameter and 18 S.W.G. (0·48 in.) thickness are practically standard.

Unannealed hard-drawn condenser tubes tend to become corroded more readily than tubes that have been annealed.

Cleaning Condenser Tubes.

A method of cleaning condenser tubes which has given good results is to shoot 'bullets' through the tubes by means of a compressed air pistol.

At Barton power station (see Guy and Lamb, Inst. Mech. Engineers, 1930) the bullets used for condenser tubes ·904 in. bore consist each of seven leather washers, giving a total thickness of 1¼ in., held together by a central pin. The washers are [illegible] in. diameter and are slightly staggered, the central holes having clearance for that purpose. Four rows of copper gauze are inserted between the washers and bent over their edges. The air pressure used is 70 to 80 lbs. per sq. in.

The condensers are cleaned, as a rule, when the vacuum has fallen 0·2 in. below the datum value.

The wages cost for cleaning by shooting as compared with brushing was reduced from £42 to £8.

The extra fuel cost of operating a 26,000 K.W. turbo generator at 0·2 in. loss of vacuum, with coal at 14s. 3d. per ton, was £2·96 per day.

* Reprinted as *The Surface Condenser*, by B. W. Pendred (Sir Isaac Pitman & Sons, Ltd.)

British Standard Specification for Condenser Tubes and Screwed Glands for Condensers for Marine Purposes.

(No. 3000—1931.)*

(*Abstract.*)

1. The material of the shells from which the tubes for condensers and for screwed glands are drawn shall be an alloy of copper and zinc as follows :—

***For Condenser Tubes :* 70/30 Alloy.—Containing not less than 70 per cent. of metallic copper**

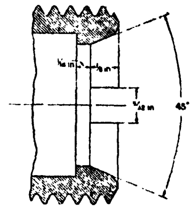


FIG. 1.—View of Outer End of Screwed Glands for ⅝ in. and ¾ in External Diameter Condenser Tubes.

and not more than a total of 0·75 per cent. of materials other than copper and zinc (except that, if so specified, it may contain 1 per cent. of tin or 1½ to 2 per cent. of lead).

***For Screwed Glands :* 60/40 Alloy.—Containing not less than 60 per cent. of metallic copper and not more than a total of 0·75 per cent. of materials other than copper and zinc (except that, if so specified, it may contain 1 per cent. of tin or 1½ to 2 per cent. of lead).**

3. The condenser tubes shall be of ⅝ in. or ¾ in. external diameter as may be specified—and they shall not be larger than, nor more than 1½ per cent. smaller than, the specified external diameter. The tubes shall be of uniform diameter throughout and the thickness shall be 18 S.W.G. within the usual manufacturing margins.

5. For ⅝ in. external diameter condenser tubes the average weight per 100 ft. run shall be not more than 32¾ lbs. and not less than 31¼ lbs., and for ¾ in. external diameter condenser tubes the average weight per 100 ft. run shall be not more than 40 lbs. and not less than 38 lbs., *i.e.* 2½ per cent. above or below the theoretical weight.

6. Each condenser tube shall be tested internally by water pressure to 600 lbs. per square inch.

Condenser Tube Corrosion.

See 'Eighth Report to the Corrosion Research Committee of the Institute of Metals.' Abridged Report, '*Engineering*,' Sept. 7, 1928.

Heat Transmission in Surface Condensers.

Let K = coefficient of heat transmission from steam to cooling water, *i.e.* the number of B.Th.U. transmitted per hour per sq. ft. of cooling surface per degree F. of mean temperature difference.

$\frac{1}{K}$ represents the resistance to the flow of heat. This resistance is made up of :—

(1) The resistance to the flow of heat from the steam to the external surface of the condenser tube, $\frac{1}{k_1}$ say ;

* See also B.S.S. 378—1941.

(2) resistance to conduction through the tube $= \frac{t}{k_2}$, where t = thickness of tube in inches and k_2 = conductivity of metal in B.Th.U. per hour per sq. ft. per temperature gradient of 1° F. per inch ; and

(3) the resistance to flow of heat from the tube to the water $= \frac{1}{k_3}$ say.

The total resistance $\frac{1}{K} = \frac{1}{k_1} + \frac{t}{k_2} + \frac{1}{k_3}$.

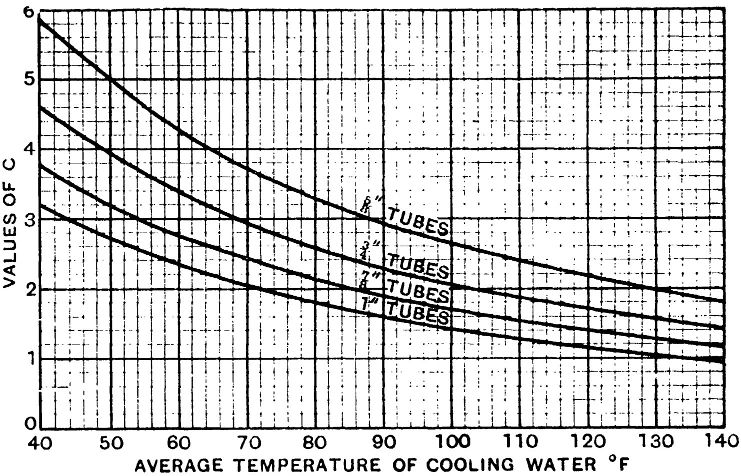


FIG. 2.

For brass tubes, 18 S.W.G. thick, the middle term $\frac{t}{k_2}$ is negligible in comparison with the others.

k_1, the coefficient for flow from steam to metal, depends on the velocity with which the steam flows over the tubes, the presence of air in the condenser, and the cleanliness of the tubes.

k_3, the coefficient for transmission from tube to water, is the most important term. It depends upon the water velocity, the diameter of the tube, cleanliness of tube surface, and also on the mean temperature, there being evidence that the heat transmitted is greater at higher temperatures even though the temperature *difference* is the same.

For the usual tube sizes and usual temperatures, it will be seen that the chief variable factor influencing the overall transmission rate is the water velocity.

Data for k_1 and k_3 are not yet thoroughly established, and the data given by various experimenters for the coefficient K also vary widely. This is due not only to experimental error but to the fact that frequently some of the factors influencing the transmission rate are not taken into account at all.

The resistance to heat flow may be expressed as—

$$R = R_a + R + R_t + R_w,$$

where

R_a is due to the effect of air blanketing the surface of the condenser tubes ;

R depends on the viscosity and density of the film of condensate clinging to the tubes;

R_t is the resistance to conduction through the metal ;

R_w depends on the velocity of the condensing water in the tubes, its density and viscosity, and the tube diameter.

Sim uses units such that $R = \frac{1{,}000}{K}$, where K is defined above.

In these units $B_a + B_f$ together may be taken as 0·6 to 1·2 for turbine plants, and 1·2 to 1·5 for reciprocating engines.

B_t for clean tubes 18 S.W.G. thick may be taken as 0·08. It increases considerably with dirty tubes.

B_w for a particular tube diameter $= \dfrac{C}{v^{0\cdot 82}}$, where v is the water velocity in feet per sec.

Values of C are shown by fig. 2.*

K is usually about 400 to 600 B.Th.U. per hour per sq. ft. per ° Fah. in typical designs, though in very high class condensers with clean tubes and small air leakage higher values are realised.

With certain recent regenerative condensers K = 600 to 800, or even up to 1,000.

An approximate formula (due to *Orrok*) is

$$K = \cfrac{1{,}000}{\cfrac{1{,}000}{140V + 250} + a}$$

where V is the velocity of the cooling water through the tubes, in ft. per second
and a = 1·1 for reciprocating engines
0·8 for turbines
0·6 for high class power station turbines.

The increased transmission rate—and consequently reduced cooling surface necessary—due to high water velocities is not all gain, however, for high velocity means increased power to circulate the water. The final design must be decided with reference to cost of pumping as well as initial cost of plant.

Fig. 3 shows the head lost in friction per 10 foot of tube element for ⅝ in., ¾ in., and 1 in. external diam. by 18 S.W.G. smooth brass tubes. For application it is necessary to take the total length of the tube element (the length of the tubes multiplied by the number of water passes and add the 'loss at entry (including loss at exit also) per pass,' values for which are given in the bottom curve.

The heat transmission, coefficient K_w, for heat flow *from condenser tube to water only*, may be taken from fig. 4 which is due to Guy and Winstanley (*Inst. Mech. Eng.*, 1934) and is based upon researches of Eagle and Ferguson (*Inst. Mech. Eng.*, 1930). The graph applies to tubes ¾ in. outside diameter, 18 I.W.G. thick, and to a heat transmission rate of 8,000 B.Th.U. per hour per sq. foot. Corrections for other rates are negligible, and corrections for other sizes of tube are not of great significance.

According to Guy and Winstanley (reference as above) the *overall* rates of heat transmission applicable to good condensers with clean tubes, may be based upon the above for the transmission from tube to water, and on 1,350 B.Th.U. per hour per sq. ft. per deg. F. for K_s, the coefficient for heat flow from steam to tube, and these values lead to an overall coefficient K expressed fairly closely by the expression

$$K = 650 \sqrt[3]{\frac{V_t}{5}} \times \sqrt[4]{\frac{t}{100}}$$

where V_t is the water velocity in the tubes in feet per sec.
and t is the arithmetic mean water temperature between inlet and outlet in deg. F.

Mean Temperature Difference.

The true mean temperature difference for condensers, heaters, coolers, etc., in which heat is transmitted between two fluids separated by a metallic wall is not the difference of the arithmetic mean temperatures.

If one of the fluids is cooled from T_1 to T_2 and the other heated from t_1 to t_2, then,

$$\text{mean temperature } t_m = \frac{(T_1 - t_1) - (T_2 - t_2)}{\log_e \dfrac{T_1 - t_1}{T_2 - t_2}}$$

Fig. 5 enables t_m to be readily determined from the maximum and minimum temperature differences $T_1 - t_1$ and $T_2 - t_2$.

Cooling Surface.

If

t_m = mean temperature difference between steam and water; K = heat transmission B.Th.U. per hour per sq. ft. per F.°; then,

$$\text{Sq. ft. of Cooling surface per lb. steam per hour} = \frac{1{,}050}{K \times t_m}$$

(1,050 is the number of B.Th.U. given by 1 lb. steam in the condenser, under usual conditions).

* Redrawn from Sim's *Steam Condensing Plant* (Blackie & Son, Ltd.), which work provides a full exposition of this important subject.

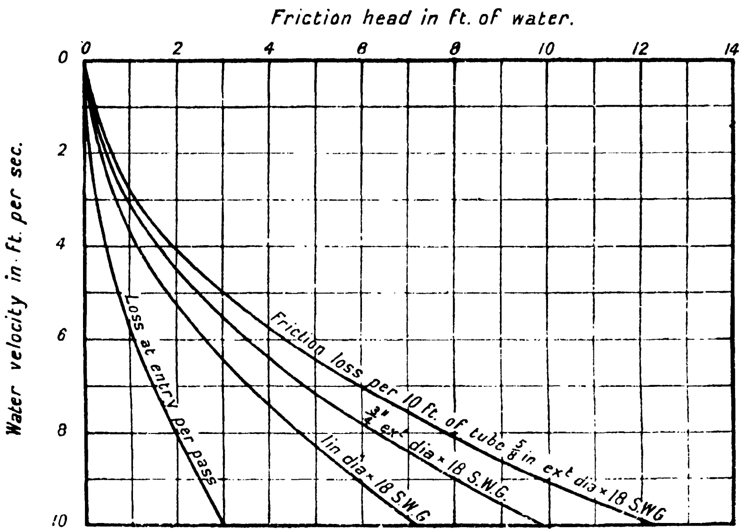

Friction head in ft. of water.
0
2
4
6
8
10
12
14
Water velocity in ft. per sec.
0
2
4
6
8
10
Loss at entry per pass
Friction loss per 10 ft. of tube 5/8 in ext dia × 18 S.W.G
3/4" ext dia × 18 S.W.G.
1 in dia × 18 S.W.G

FIG. 3.

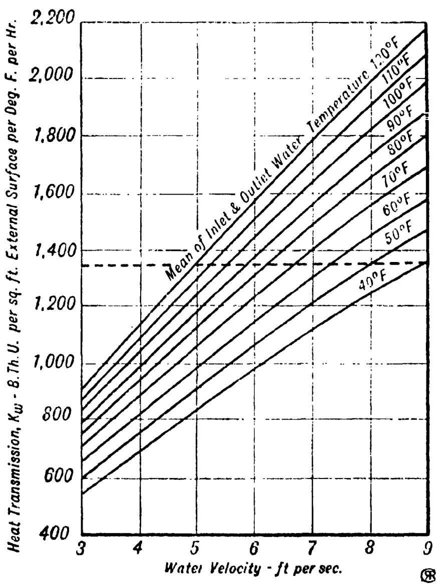

Heat Transmission, Kw - B.Th.U. per sq. ft. External Surface per Deg. F. per Hr.
2,200
2,000
1,800
1,600
1,400
1,200
1,000
800
600
400
Mean of Inlet & Outlet Water Temperature 120°F
110°F
100°F
90°F
80°F
70°F
60°F
50°F
40°F
3
4
5
6
7
8
9
Water Velocity - ft per sec.

FIG. 4.

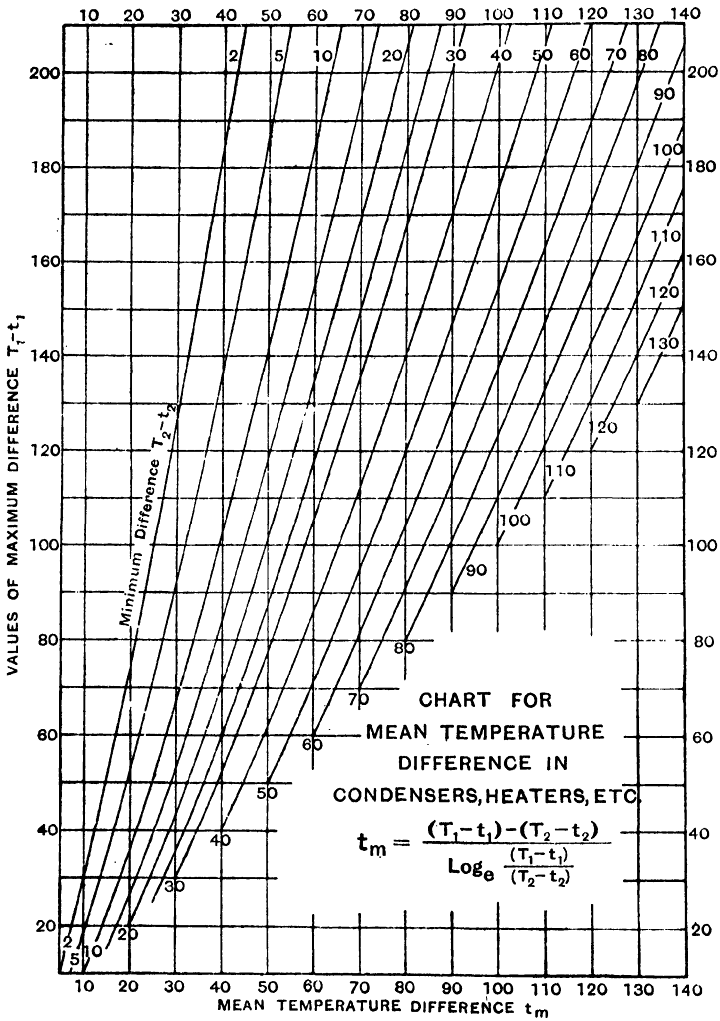


CHART FOR
MEAN TEMPERATURE
DIFFERENCE IN
CONDENSERS, HEATERS, ETC.
$t_m = \dfrac{(T_1 - t_1) - (T_2 - t_2)}{\mathrm{Log}_e \dfrac{(T_1 - t_1)}{(T_2 - t_2)}}$
VALUES OF MAXIMUM DIFFERENCE $T_1 - t_1$
MEAN TEMPERATURE DIFFERENCE t_m
Minimum Difference $T_2 - t_2$
10 20 30 40 50 60 70 80 90 100 110 120 130 140
20 40 60 80 100 120 140 160 180 200
2 5 10 20 30 40 50 60 70 80 90 100 110 120 130

FIG. 5.

QUANTITY OF COOLING WATER.

If T_1 = temperature of cooling water at inlet to condenser; T_2 = temperature of cooling water at outlet of condenser; T_3 = temperature of steam at inlet, corresponding to the vacuum; T_4 = temperature of the condensed water at outlet; h_3 = sensible heat at inlet; h_4 = sensible heat of the condensed water at outlet; L = latent heat in B.Th.U. per lb. of steam; W = steam weight; Q = water weight; R = ratio of water and steam quantity; x = dryness fraction; then,

Heat (in B.Th.U.) given up by exhaust steam $= W\left\{xL + (h_3 - h_4)\right\}$

Heat (in B.Th.U.) gained by cooling water $= Q(T_2 - T_1)$.

In practice $h_3 - h_4$ may be neglected, therefore

$$WxL = Q(T_2 - T_1), \quad \text{and,} \quad \frac{Q}{W} = R = \frac{xL}{T_2 - T_1}.$$

To allow a small margin, x—which generally varies between 0·8 and 1·0—can be neglected, and as L only varies slightly with the vacuum, it is generally assumed to be a constant for all vacua at 1,050 B.Th.U. per lb. of steam. Therefore the general equation for all vacua can be taken as

$$R = \frac{Q}{W} = \frac{1050}{T_2 - T_1}.$$

For surface plants, T_3—T_2 ranges from 8° to 20° F., and for jet plants, 4° to 8° F.

The ratio of the amount of cooling water to that of the steam to be condensed usually varies from 35 to 40 for jet, and from 60 to 70 for surface condensers.

It is important that the condensed steam should be at as high a temperature as possible, since this water is usually returned to the boilers, therefore it is essential that all the heat possible should be recovered.

Every 8°-10° F. difference in the hot-well temperature means approximately 1 per cent. difference in fuel consumption. The gain obtained with an increase in hot-well temperature can be calculated as follows :—

$$\text{Percentage gain in B.Th.U.} = \left(\frac{100\,T}{H_T + T - T_4 + 32}\right) \times \frac{100}{B_p}$$

where,

T = difference in hot-well temperature (F.°); T_4 = final hot-well temperature (F.°) H_T = total heat of steam at boiler pressure (B.Th.U. per lb.), reckoned from 32° F.; B_p = percentage boiler efficiency.

In surface condensers for turbines, $(T_3 - T_4)$ is about 5° to 8°, and for reciprocating engines about 10° to 15°.

Vacuum Efficiency may be expressed as the ratio of the vacuum at the condenser inlet to vacuum corresponding to the temperature of the condensed steam; it is dependent both upon the design of the condenser and the air pump.

The *Efficiency of a Condenser* is best stated as a 'Coefficient of Performance,' which measures the extent to which the two conditions are fulfilled that (1) the condensate should be cooled as little as possible below the steam temperature, and (2) the quantity of cooling water should be a minimum, and hence its rise of temperature a maximum.

Sim takes as 'Coefficient of Performance' the ratio $(T_2 - T_1) \div (T_3 - T_4 + 10)$, and states that it should be 1·0 to 1·2 for good surface condensers.

Degree of Vacuum.—The maximum vacuum obtainable is dependent upon the inlet circulating water temperature. For engine work 26 ins. to 27 ins. is the maximum vacuum desirable, owing to the limitations in size of the low-pressure cylinder and ports, etc. For turbine work high vacua are necessary. The most economical vacuum depends upon individual installations, and is generally decided by the power required to be expended by pumping the cooling water, price of coal, available water supply, etc.

The following may be taken under average conditions to be the maximum commercial vacua.

Temp. of inlet water, F.° .	55	60	65	70	75	80	85
Vacuum in ins. (baro. 30 ins.)	28·8	28·6	28·5	28·3	28·0	27·8	27·5

PROPERTIES OF SATURATED STEAM AT LOW PRESSURES.
Per Increment of Vacuum reduced to 30 ins. Barm.

Inches of Vacuum.	Absolute Pressure Lbs. per Sq. In.	Temp. °F.	Spec. Vol. Cub. Ft. per Lb.	Inches of Vacuum.	Absolute Pressure Lbs. per Sq. In.	Temp. °F.	Spec. Vol. Cub. Ft. per Lb.
0	14·69	212	26·79	25	2·45	133·8	143·5
1	14·20	210·3	27·64	25·2	2·35	132·1	149
2	13·71	208·5	28·54	25·4	2·25	130·5	155
3	13·22	206·7	29·52	25·5	2·21	129·8	158·5
4	12·73	204·9	30·61	25·6	2·16	128·9	161·5
5	12·24	202·9	31·75	25·8	2·06	127·2	169
6	11·75	201	32·95	26	1·96	125·3	177
7	11·26	198·9	34·37	26·1	1·91	124·4	181
8	10·77	196·8	35·80	26·2	1·86	123·4	185·5
9	10·29	194·6	37·35	26·3	1·81	122·4	190
10	9·80	192·3	39·1	26·4	1·76	121·4	195·5
10·5	9·56	191·1	40	26·5	1·71	120·4	200·5
11	9·31	189·9	41	26·6	1·67	119·6	206
11·5	9·07	188·7	42	26·7	1·62	118·4	212
12	8·82	187·4	43·15	26·8	1·57	117·3	218·5
12·5	8·58	186·1	44·25	26·9	1·52	116·2	225
13	8·33	184·8	45·5	27	1·47	115·1	232
13·5	8·09	183·4	46·8	27·1	1·42	113·9	239·5
14	7·84	182	48·2	27·2	1·37	112·6	247
14·5	7·60	180·6	49·6	27·3	1·32	111·3	255
15	7·35	179·1	51·2	27·4	1·27	110	265
15·5	7·11	177·6	52·8	27·5	1·23	108·7	276
16	6·86	176·1	54·6	27·6	1·18	107·3	286
16·5	6·62	174·4	56·4	27·7	1·13	105·8	298
17	6·37	172·8	58·3	27·8	1·08	104·3	310
17·5	6·13	171·1	60·5	27·9	1·03	102·7	324
18	5·88	169·3	63	28	0·98	101·1	339
18·5	5·64	167·4	65·5	28·1	0·93	99·4	356
19	5·39	165·5	68·4	28·2	0·88	97·6	374
19·5	5·15	163·5	71·4	28·3	0·83	95·7	395
20	4·90	161·4	74·7	28·4	0·79	93·8	418
20·5	4·66	159·3	78·4	28·5	0·74	91·7	444
21	4·41	157	82·5	28·6	0·69	89·5	474
21·5	4·17	154·7	86·9	28·7	0·64	87·1	508
22	3·92	152·2	92·1	28·8	0·59	84·6	548
22·5	3·68	149·6	97·7	28·9	0·54	81·9	597
23	3·43	146·8	104·3	29	0·49	79	653
23·5	3·19	143·9	112	29·1	0·44	75·8	721
24	2·94	140·7	121	29·2	0·39	72	806
24·5	2·70	137·4	131	29·3	0·34	68	913
24·6	2·65	136·7	133·5	29·4	0·29	63	1060
24·8	2·55	135·2	138·3	29·5	0·25	59	1230

Resistance to Water Flow in Condensers.

Guy and Winstanley (*Inst. Mech. Eng.*, 1934) propose the following formula for the total friction loss of head in the water passes of a condenser, including the losses due to changes of direction in the water heads :—

$$H = n\left(C_1 \frac{L}{D}\frac{V_t^2}{2g} + C_2 \frac{V_t^2}{2g}\right) + C_b \frac{V_b^2}{2g}$$

where n is the number of flows;
- L length of tube in feet;
- D diameter of tube in *feet*;
- V_t mean velocity of water through tubes in ft. per sec.;
- V_b velocity in outlet branches of water box.

C_2 depends upon the type of tube fixing and has following values :—

Tubes ferruled at both ends	1·5
Tubes bellmouthed and expanded at inlet, and ferruled at outlet . .	1·25
Tubes bellmouthed at inlet and expanded at outlet	1·00

It is suggested that C_b be taken as unity in view of the great difficulty of determining the water box loss.

For C_1 the value 0·30 is suggested for tubes cleaned by wire brushing, and for cooling water at 70° F. at velocity 5 ft. per sec. in ¾ in. external diameter tubes 18 I.W.G.

For other thicknesses and tube diameters, correction factors by which O_1 is to be multiplied are given (see original paper) which vary from 1·067 for ⅝ in. diam., 16 I.W.G., to 0·918 for 1 in. diam., 19 I.W.G. The paper gives a graph for correction factors for temperature and velocity varying from 1·3 at 40° and 3 ft. per sec., to 0·67 at 240° and 8 ft. per sec.

Air in Condensers.

Air enters the condenser in the following manner :—

1. With the steam entering from the boiler, having been dissolved in the feed water; this is usually a small amount in the nature of 1 to 3 per cent. of the total volume of feed water.

2. Air leakage from the atmosphere through the glands of the prime mover or other sources; this cannot easily be measured, and an allowance has to be made accordingly to cover contingencies.

3. In jet condensers air in solution with the cooling water which is released when in the vacuum space; this varies from 1 to 5 per cent. of the total volume of water.

Due to air contained in the cooling water in the case of a jet condenser, a much larger air pump is required than for a surface plant of similar duty.

For turbine work to cover cases (1) and (2) it is usual to allow for 6 to 8, and, in the case of low-pressure turbines, up to 11 lbs. of air per 10,000 lbs. of steam.

For engines, about two or three times this amount is required according to the type of engine.

These figures are for one unit only; if two or more turbines or engines exhaust to the one condenser an extra margin of 10 or 15 per cent. per extra turbine or engine, respectively, should be allowed.

The B.E.A.M.A. specifies a fixed allowance to cover air in steam due to leakage and entrainment. For land plants

$$W_a = \left(\frac{0\cdot 5W_s}{1{,}000} + 3\right) \text{ lbs. per hour}$$

where W_s = lbs. steam condensed per hour. It is probable that the actual air may be about half this allowance.

The Ejector Condenser.

This condenser is at once an air-pump and a condenser. It acts on the principle that steam will flow into a vacuum at a velocity of many hundred feet per second. The water supply to the ejector should flow to it under a head of several feet. The steam condenses in the jet of water, and the combined jet has a velocity which is a mean of the velocity of the steam and of the water. Thus let the water approach at a velocity 30 feet per second and the steam at 1,200 feet per second. Let the combined jet be assumed to have a temperature of 100° F. and the water to

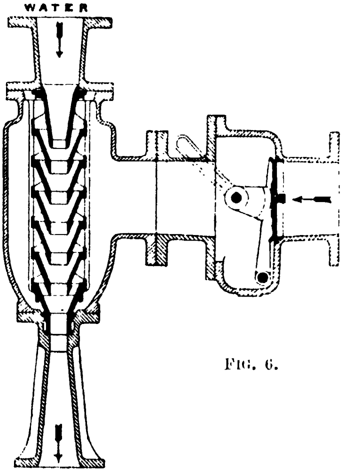


FIG. 6.

be supplied at 60° F. The water gains 40° F. The steam losses, say 1,100 B.Th.U. Then 1,100 ÷ 40 = 27 = ratio of water weight to weight of steam. The total weight of combined jet is 28. The velocity difference is 1,200 − 30 = 1,170, to be divided over a mass of 28, or $\frac{1{,}170}{28}$ = 42 feet per second nearly. Add the initial velocity of the water 30, and the final velocity is 72 feet per second

This corresponds with a head of $H = (72 + 8)^2 = 81$ feet; and since 1 foot of water column equals 0·43 pounds, 83 feet will represent 36 pounds nearly. Since a perfect vacuum is only 14·7 pounds, it follows that with ordinary ratios of water and steam an efficiency of less than 50 per cent. should give an excellent vacuum.

The apparatus (fig. 6) consists of a converging water nozzle into which water is directed from a centrifugal pump or other pressure source, or from an overhead source of 15 to 20 feet head. This jet flows downwards through a series of steam cones to which the exhaust steam has admission at their upper wide portion. The steam flows down each of these short cones and imparts further velocity to the jet as it condenses, and finally the jet flows into a divergent discharge cone at a velocity as above indicated, carrying with it any air that has come in with the steam, and discharging it. The degree of vacuum is of course that proper to the jet temperature, and the proportion of water can, as seen, be sufficiently great to admit of a moderate final jet temperature and therefore a good vacuum. The ejector condenser occupies very little space and for jet condensing is very simple. The centrifugal water supply pump is conveniently direct driven by an electric motor. While the jet final temperature is amply low for good vacua, the velocity of the combined jet is still more than sufficient to run freely against atmospheric external pressure; but it is always desirable to give the water a fair initial velocity, and there is nothing lost by giving the discharged jet a good fall in a long tail pipe.

The Barometric Condenser is a simpler form of ejector condenser, in which the water enters the head of the apparatus as a spray about the steam entrance, and the vacuum is produced by the fall of the water down a tail pipe of at least 30 feet in vertical drop. By suitable formation of the head the air which enters with the steam is entrained with the water, or, by means of a special surface condenser at the side of the head the air is drawn over cold tubes and carried down to a lower point in the draught tube, whereby it is withdrawn from the head at a minimum temperature, with the object of securing a maximum degree of vacuum.

Air-Pumps.

The essential feature of a plant having a high vacuum efficiency is an air-pump capable of operating without cooling the condensed steam. Excepting in small plants where the heat loss is not so appreciable, all surface condensers should be arranged with two pumps working on the vacuum side, *i.e.* an air-pump capable of dealing with all the air entering the condenser and a water-pump or extraction pump capable of dealing with all the condensed steam. This is what is known as a 'dry' system. Cases where only one pump is employed for the combined duty are known as 'wet' systems. In order to cut down the size of the air-pump in the latter system to reasonable proportions, the condensed steam is usually cooled by means of the last few rows of tubes in the condenser being arranged so that they are always flooded, or by an intercooler.

CAPACITY OF AIR-PUMP IN ORDER TO REMOVE GIVEN AMOUNT OF AIR.

The diagram (fig. 7, p. 179) shows the properties of mixtures of air and steam such as exist at the air-pump suction. Thus, if the temperature is 100° F., and the vacuum 26 ins., the volume of 1 lb. of air (together with the associated vapour) is 204 cu. ft. and the weight of air per lb. of steam is 1·75, *i.e.* 204 cu. ft. of the mixture contains 1 lb. of air and 0·57 lb. vapour.

The scale on the right provides a basis for calculating the air-pump capacity if the air leakage and the temperature and vacuum are known. For example, assume a surface condenser installation for a reciprocating engine in which allowance is to be made for 16 lbs. air per 10,000 lbs. steam, vacuum 27 ins., temperature at air-pump suction 100° F. From the right-hand scale the volume of air per lb. of condensate, measured at the conditions of the air-pump suction, is 0·395 cu. ft. for an air proportion of 10 lbs. per 10,000 lbs. steam. Thus in the present case the volume is

$$0 \cdot 395 \times \frac{16}{10} = 0 \cdot 63 \text{ cu. ft. per lb. steam.}$$

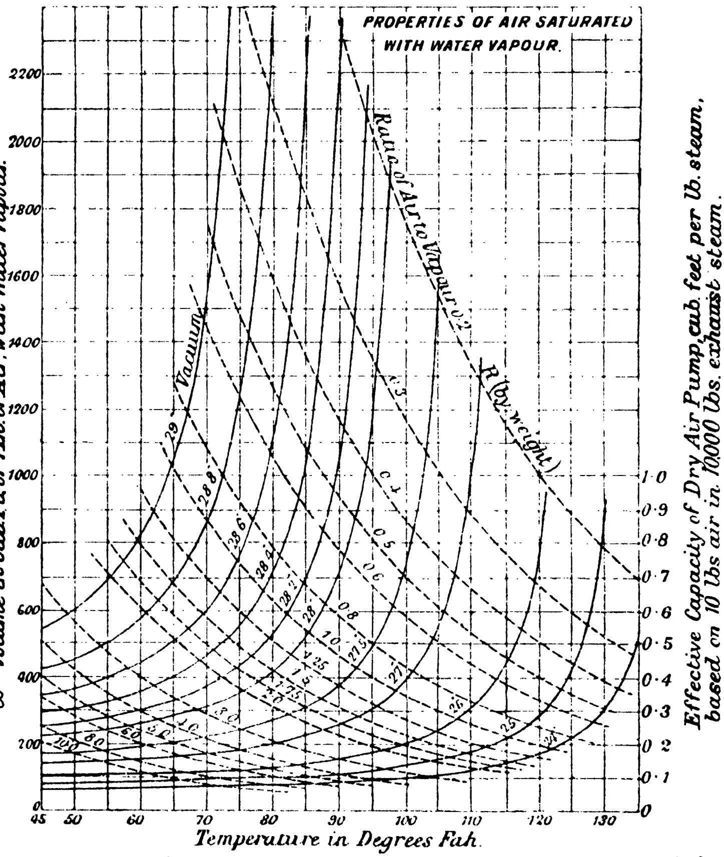
If the steam condensed is 15,000 lbs. per hour, the *effective* capacity of the 'dry' air-pump will then be

$$\frac{15{,}000}{60} \times 0 \cdot 63 = 157 \text{ cu. ft. per minute.}$$

For a 'wet' air-pump the volume of the condensate must be added. The actual capacity of the pump must be greater than the capacity thus found, on account of losses of volumetric efficiency due to clearance, slip, leakage, and valve or port resistance.

The diagram shows the reduction in volume due to cooling of the air-steam mixture before it passes to the air-pump.

DIAGRAM SHOWING THE PROPERTIES OF MIXTURES OF AIR AND STEAM, SUCH AS EXIST AT THE AIR PUMP SUCTION.



(Morley.)

FIG. 7.

In a jet condensing plant the air introduced by the injection water must also be allowed for.

Leakage of air into the exhaust pipe and condenser should be reduced to a minimum by xercising the greatest care to keep all joints, etc., in good condition.

The amount of air entering the exhaust pipe due to leakage at joints and glands is obviously not proportional to the amount of exhaust steam, nevertheless air-pumps are frequently designed to give an effective displacement proportional to the amount of exhaust steam, and in fact air-pumps for piston engines are frequently proportioned upon the low-pressure cylinder.

Vertical air-pumps are usually single-acting, and low-pressure engine cylinders double-acting. On this basis for ordinary bucket pumps the piston displacement of the low-pressure cylinder is about 6·5 times that of the pump for jet condensing compound engines in this country and 5·5 times for India and hot tropical climates. For surface condensers this air-pump ratio varies from 10 to 12, the lower figure for the smaller and less efficient engines.

Wet-air pumps for jet condensers are frequently proportioned so that their effective displacement is about five times the volume of the water (injection plus condensed steam) which they have to discharge.

Horizontal air-pumps are double-acting, and their volume is therefore one-half that of single-acting pumps.

Air-pumps of the Edwards type usually have about 75 or 80 per cent. as much capacity as the ordinary pumps.

These notes apply particularly to air-pumps for reciprocating engines, and it is tacitly assumed that the vacuum aimed at is about 26 ins. Considerably greater capacity is necessary for higher vacua.

Air-pump bucket speeds rarely exceed 200 ft. per minute unless the buckets are of the solid plunger type. A usual value is 180 ft. per minute.

The vacuum in a condenser ought to maintain itself an hour with a drop of only 2 lbs.

The smallest air-pump found practicable by Sir R. W. Allen had 0·5 cu. ft. of displacement per pound of steam condensed.

With circulation water at 65° to 70° F. a vacuum of 28½ ins. at full load with a single-stage air-pump can be maintained without special accessories. It is not advisable to allow less than 0·75 cu. ft. of air-pump volume per 1 lb. of steam condensed, and it ought to be possible to maintain a 98½ per cent. vacuum.

Circulating Pump.

The pump for delivering the required water quantity to the condenser is nearly always of the centrifugal type. The head against which this pump has to deliver is made up of the static difference in water levels between the inlet and discharge points, plus the pipe and condenser friction. In cases where water is drawn from and returned to a river or canal, the discharge pipe end can be sealed, thus forming a closed or syphon system. In arranging syphon systems care should be taken to ensure that the highest point in the system is never above a vertical distance of 25 ft. from the water level, otherwise the air contained in solution with the water is liable to be released and accumulate, thus breaking the syphon and throwing extra head on the pump.

Jet or Ejector Air-Pumps.

Of recent years great developments have taken place in the design and application of air-pumps in which the vacuum is produced entirely by means of a jet or jets of fluid, which may be water or steam. The air from the condenser mixes with the jet, and the mixture, which just after mixing is at high velocity and low pressure, passes through a 'diffuser' tube of increasing area in which its kinetic energy is as far as possible transformed into pressure energy, so that the mixture may be finally discharged against atmospheric pressure.

Such types of air-pump have the advantage that there are no moving parts in the complete condensing plant, with the exception of a pump for removing the condensate; also they occupy very little space, have little weight, can be located in any convenient situation, and require little attention and maintenance.

The types using steam jets have superseded those using water jets.

It is found advantageous not to carry out the compression of the air from vacuum to atmospheric pressure in one stage, but in two or more stages. In order to reduce the work in the later stages, and hence to reduce steam consumption, frequently auxiliary condensers are used between the stages

Measurement of Condensed Steam.

The condensate may be measured by passing over a V-notch in a suitably arranged tank, the height of the water surface above the apex of the notch being measured at a point at such a distance from the notch that the water is nearly still.

If h is the height in inches, Q the flow in cub. ft. per minute, and the angle of the notch is 90°, then $Q = ch^2\sqrt{h}$, where c is approximately 0·305, but, more accurately, varies from 0·31 at $h = 2$ ins., to 0·301 at $h = 9$ ins. or over. Less heads than 2 ins. should not be used, it being better to increase the head by using a notch of narrower angle; the discharge is approximately proportional to the tangent of half the notch angle.

If one notch will not pass the water, two may be used, and the reading of the instrument doubled.

In the 'closed feed' systems, which are now frequently adopted in power stations, the condensate and the make-up feed-water may be measured by means of Venturi meters included in the pipe system.

Injection Cocks

are usually of the hollow plug type, in gunmetal. One per condenser is sufficient. Two or three standard sizes are generally available to choose from. The minimum size can be checked as in the following example:—

Maximum load under normal conditions	1,000 i.h.p.
Steam per i.h.p. hour	15 lbs.
Steam per second	4·17 lbs.
Condensing water per pound of steam	40 lbs.
,, ,, per second	167 lbs.
,, ,, ,,	2·67 cub. ft.
Vacuum in condenser, worst	24 ins.
,, ,, in feet of water	27 ft.
Vertical height water level to injection cock	11 ft.
Head of water on injection service, 27-11	16 ft.
Assume spent in friction	50 per cent.
Net head at injection cock	8 ft.
Velocity of water at cock $= 8\sqrt{8}$	22·6 ft. sec.
Hence area of opening	0·118 sq. ft.
,, ,,	17 sq. ins.

If in doubt add a margin of 25 to 30 per cent. It is, of course, understood that the above level, steam consumption, etc., must be modified to suit the actual conditions. Thus in tropical countries 60 should be substituted for 40 lbs. of condensing water per lb. of steam.

Force or Feed Pumps.

When the boiler feed-pump is driven off the engine it is usual to allow sufficient capacity for other works purposes.

The following are frequently adopted:

Textile spinning mills	25 lbs. water per i.h.p. hour
Textile weaving sheds	40 ,, ,, ,,
Mixed spinning and weaving	30 ,, ,, ,,
Flour mills	35 ,, ,, ,,
Bleach works	50 ,, ,, ,,

It will be seen that an ample allowance is made, and for economical superheated-steam engines a reduction of about 5 lbs. can be made. The above are based on actual pump displacement. The plunger speed should not exceed about 200 ft. per minute.

Pump Openings.

The velocity of water in pump openings should not exceed 500 ft. per minute.

Cooling Ponds.

Where the supply of water for condensing purposes is limited, it is necessary to cool the condensing water and use it again and again. One method of cooling is to pass the hot water into an open pond and draw off water for condensing from the opposite side of the pond, the water being cooled simply by evaporation from the surface.

Spray Ponds.

An improvement upon the simple pond is to lay a system of piping over the pond and let the water issue from nozzles in a fine spray. A large amount of water is, however, carried away by the wind and lost.

A spray pond should have 2¼ sq. ft. of area per gallon of water to be cooled per minute. This includes the marginal straps necessary for windage, etc.

Water-Cooling Towers.

Water-cooling towers are of two principal types—chimney type and open type. Both act by bringing the water into intimate contact with air currents, and to expose as large a surface of water to the air as possible the water is either broken up by splash bars into fine drops, or made to run down in thin films. In chimney type towers air is admitted at or near the bottom only, and passes up inside the casing usually by natural draught, though sometimes forced draught is used. Forced-draught towers are smaller than natural-draught for the same duty, but require a large expenditure of power in the fans.

Open type coolers have water-distributing arrangements similar to those in the chimney type, but have no external closed casing, the flow of air across the water is dependent upon the action of the wind. To prevent excessive loss of water, the stack is usually surrounded by louvre boarding, but even then the loss of water is large compared with that in the chimney type.

In chimney type towers the water is usually pumped to distribution troughs at a height of 16 to 25 ft.; the height of the chimney is about 70 ft. if of timber, 100 to 120 ft. if of steel (which is less usual). Towers are often of reinforced concrete, up to 150 ft. in height. With wide towers, special means of introducing the cool air to the central parts are necessary. With such means towers may be up to 40 ft. in width, and should not be narrower than about 15 ft., in order that they may have an adequate margin of stability.

In all cooling towers the heat is extracted from the water partly by (1) direct heating of the air, partly by (2) evaporation of part of the water. Under typical atmospheric conditions in this country evaporation accounts for about 75 to 85 per cent. of the cooling.

The amount of evaporation possible depends upon the amount of moisture already contained in the air entering the tower and its temperatures at inlet and outlet.

It is important, in specifying or judging the capacity of cooling towers, to take into account the humidity or the wet and dry bulb readings.

The 'relative humidity' of the atmosphere is the ratio of the weight of water vapour contained in a certain volume of air to the weight required to saturate this volume at the same temperature, or it may be expressed as the ratio of the vapour pressure at the dew-point to that at the actual air temperature, the dew-point being the temperature to which the air must be cooled in order that it may just become saturated, so that further cooling would cause condensation to begin. The greater the humidity of the atmosphere, the less, of course, is its capacity for taking up moisture by evaporation. The humidity is ascertained by reading wet and dry bulb thermometers and referring to humidity tables.

The quantity of fresh 'make-up' water required to compensate losses by evaporation, etc., in cooling plants is about 3 to 5 per cent. of that in circulation.

It is stated that the loss by evaporation only in natural draught chimney type cooling-towers is approximately 1 per cent. for each 12° F. by which the water is cooled in the tower.

(*I. V. Robinson*, '*B.E.A.M.A. Journal*,' *March* 1921.)

The amount of air for which the cooling tower should be proportioned may be calculated as in the following example:—

Let the atmospheric temperature and humidity be 65° F. and 60 per cent. respectively, the atmospheric pressure 14·7 lbs. per sq. in., and let the air leave the tower at 85° F. (saturated with vapour).

From steam tables, the vapour pressure at 65° F. is 0·31 lb. per sq. in., thus the air pressure is 14·7−0·31 = 14·39 lb. per sq. in.; 1 lb. air at this pressure and at 65° F. occupies 13·45 cu. ft.; 1 lb. vapour at 85° F. occupies 990 cu. ft.

$$\therefore \text{ weight of vapour per lb. air if saturated} = \frac{13\cdot45}{990} = 0\cdot0136 \text{ lb.}$$

$$\therefore \text{ weight of vapour at 60 per cent. humidity} = 0\cdot0136 \times 0\cdot6 = 0\cdot00816 \text{ lb.}$$

At 85° F. the vapour pressure = 0·6 and air pressure 14·1 lb. per sq. in.; 1 lb. air at these conditions occupies 14·3 cu. ft.; 1 lb. vapour at 85° occupies 541 cu. ft.

$$\therefore \text{ vapour per lb. air} = \frac{14\cdot3}{541} = 0\cdot0264 \text{ lb.}$$

Hence, for each lb. of air passing through the cooling tower, 0·0264−0·00816 = 0·0182 lb. water is evaporated.

Heat to evaporate 1 lb. water from water at 65° F. to vapour at 85° F. = 20 + latent heat = 1·061 B.Th.U

$\therefore$ heat to evaporate 0·0182 lb. = 0·0182 × 1·061 = 19·3 B.Th.U.
Heat absorbed in heating air = 0·237 × 20 = 4·74 B.Th.U.
Total heat taken from water per lb. air = 19·3 + 4·74 = 24·04 B.Th.U.

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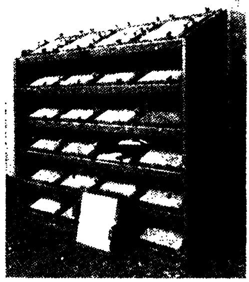

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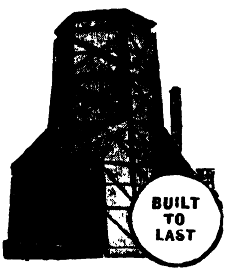

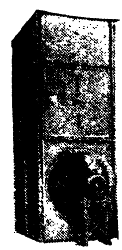

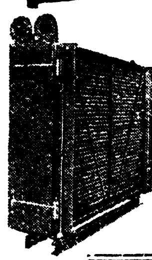

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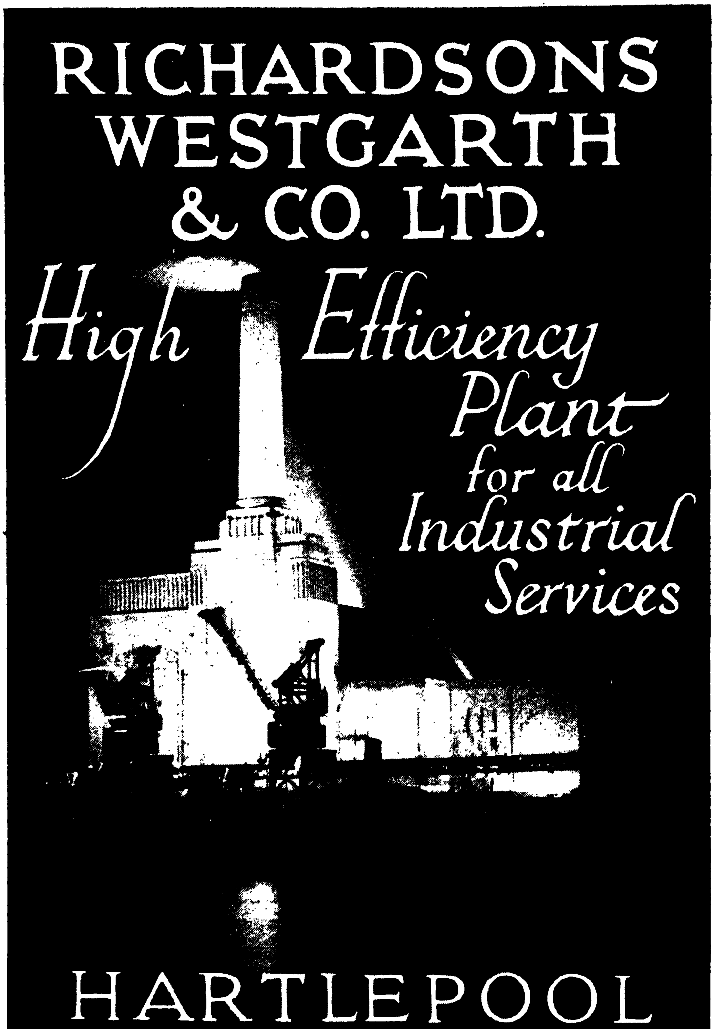
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Ratio of cooling by evaporation to total cooling

$$= \frac{19 \cdot 3}{24 \cdot 04} = 0 \cdot 8 \text{ or } 80 \text{ per cent.}$$

If the water is cooled, say 30°, the weight of water cooled per lb. air $= \frac{24 \cdot 04}{30} = 0 \cdot 801$ **lb.**

Percentage of water lost by evaporation

$$= \frac{0 \cdot 0182}{0 \cdot 0182 + 0 \cdot 801} \times 100 = 2 \cdot 22 \text{ per cent.}$$

Weight of air to pass through cooler per lb. water

$$= \frac{1}{0 \cdot 801} = 1 \cdot 25.$$

It is stated by Mr. W. T. Bottomley that for a typical base-load power station with cooling towers, the economical loading of the towers should be 14,000 B.Th.U. per hour per sq. ft. of base area, with a cooling range of 14° F. The economical rainfall at the tower base should therefore be 100 gallons per hour per sq. ft. of base area.

STEAM ACCUMULATORS.

The function of a steam accumulator is to absorb heat during periods when there is an excess of live or exhaust steam, and to restore the heat in the form of a constant supply of steam at some lower pressure. The employment of accumulators enables a boiler plant to be operated more efficiently because of the greater uniformity of its load, and the economy thus brought about in many kinds of factories amply justifies the comparatively high cost of the apparatus. Accumulators can, of course, only deliver saturated steam.

When engines such as those of collieries or rolling mills, work intermittently, stopping and starting at short intervals, their exhaust steam may be usefully discharged into an accumulator from which a steady supply may be drawn for operating a low pressure turbine or any other purpose. In a few instances accumulators of the gas-holder type have been employed for this purpose, but these are generally too large and costly to be advisable. They have, however, the advantage of great simplicity, and they also work with an extremely slight pressure drop, while the heat loss from their exposed surfaces is not excessive on account of the low ratio of the surface to the volume. Another good feature is the constancy of their discharge pressure, which remains unchanged so long as there is any steam in the accumulator. In one British colliery a steam accumulator of the gas-holder type has been in constant use for more than thirty years.

A much more common type is exemplified by the Ruths accumulator. This consists of a large pressure-tight cylindrical container with spherical ends and nearly filled with water. The charging steam is delivered to the accumulator by a pipe from which it escapes into the water by submerged nozzles. On the top of the container is a steam dome from which steam at some lower pressure is taken continuously or as desired. Automatic valves in the inlet and outlet pipes control the steam flow in accordance with the pressures in the respective mains.

When such an accumulator is installed for the purpose of enabling boiler to be worked at constant load, it is charged directly from the live steam main at times when excess steam is available. When no live steam is entering it, the supply of low pressure steam is maintained by the evaporation of the water at the expense of its own heat. The accumulator therefore works at variable temperature and pressure. The limits of pressure between which it will have to supply steam have therefore to be decided on, for these determine the amount of storage it can provide.

The amount of steam an accumulator can store, or conversely, the weight of water required for a given storage capacity can be calculated in the following manner.

Let the accumulator before charging (*i.e.* when at its lowest permissible working pressure) contain W_0 lbs. of water having a heat content of h_0 B.Th.U. per lb. When charged up to full boiler pressure, let it hold W_1 lbs. of water having a heat content of h_1 per lb. Then, if H is the total heat per lb. of the charging steam, the heat brought in by the charge will be $H(W_1 - W_0)$, and this will increase the heat in the accumulator by $(W_1 h_1 - W_0 h_0)$. By equating these two expressions and simplifying them we obtain

$$\frac{W_1}{W_0} = \frac{H - h_0}{H - h_1}$$

whence we find the weight of steam absorbed by each lb. of the original quantity of water to be

$$\frac{W_1 - W_0}{W_0} = \frac{h_1 - h_0}{H - h_1} \text{ lbs.}$$

The values of H, h_0, and h_1 for any given pressures and temperatures are to be found in Callendar's Steam Tables.

To calculate the quantity of steam that an accumulator can supply before its pressure has dropped to the lowest permissible point is not a difficult matter.

If there are W lbs. of water in the accumulator at any time when it is discharging, and no steam is entering it, a small discharge of dW lbs. of water in the form of steam will abstract an amount

of heat equal to LdW, where L is the latent heat of steam at the pressure concerned. This amount must be equal to Wdh, where h is the heat in a pound of the water, so that by equating these values we have

$$\frac{dW}{W} = \frac{dh}{L}.$$

If, at the beginning of the discharge, the accumulator contains W_1 lbs. of water with h_1 heat units per lb., and at the end of the discharge it contains W_2 lbs. with h_2 heat units per lb., we find by integrating the above equation between these limits, that

$$\log_\epsilon W_1 - \log_\epsilon W_2 = \frac{h_1 - h_2}{L_m}$$

where L_m is the mean value of the latent heat of steam over the range of temperature concerned.

Denoting, for brevity, the expression on the right hand side of the above equation by x the equation can be put into the form

$$W_2 = \frac{W_1}{e^x}$$

in which e^x can be taken as equal to $\left(1 + x + \frac{x^2}{2}\right)$ with more than sufficient accuracy for all practical purposes.

From the above equation we can calculate W_2 the amount of water remaining in the accumulator after a discharge through any given range of pressure, and so get $(W_1 - W_2)$ which is the weight of steam that will have been delivered during the discharge.

It is interesting to observe that, if an accumulator is charged through a given range of pressure and then discharged through the same range, it will contain slightly more water than at first. This is because the low-pressure steam has a somewhat greater latent heat than the charging steam, so that to produce an equal temperature difference in the water, the weight of charging steam must exceed the weight of steam discharged.

Thermal Storage at Constant Pressure.

The system of thermal storage at constant pressure was developed by the late Druitt Halpin to enable boilers to cope with heavy demands for steam by the provision of ample feed-water at the full boiler temperature. A large cylindrical storage vessel above the boiler drum contained water to which the boiler steam had free access. The water was thus raised to boiler temperature and pressure to serve as feed-water at times of the greatest load. The storage vessel thus virtually increased the water capacity of the boiler and so enabled it to deal better with variable demands for steam.

In the Kiesselbach accumulator the same principle is carried out in a more elaborate manner. The storage vessel is placed below the level of the boiler drum, to which its steam and water spaces are both connected. A circulating pump draws water from the bottom of the storage vessel and delivers it to the boiler drum, whence an equivalent quantity flows back to the storage vessel so long as the level in the boiler is above the inlet to the return pipe.

When the demand for steam is small the ordinary boiler feed pump delivers an excess of water to the boiler, with the result that the storage vessel becomes filled by means of the return pipe. At times when the demand for steam is abnormally large the feed pump is stopped, the drum being only supplied then by the circulating pump with fully heated water from the storage vessel.

See also Descriptive Section XXVII, Part V.

SECTION XXVIII

THE INTERNAL COMBUSTION ENGINE

(By R. J. W. Cousins, M.I.Mech.E.)

PART I

UNITS AND EQUIVALENTS—GAS LAWS, etc. (pp. 187–196)

PART II

GENERAL DESCRIPTION OF TYPES OF ENGINES AND FUELS — CONSTRUCTIONAL DETAILS PECULIAR TO LARGE ENGINES—NOTES PARTICULARLY APPLICABLE TO HIGH-SPEED ENGINES (pp. 197–214)

PART III

DESIGN OF I.C. ENGINES—TORSIONAL VIBRATIONS OF CRANKSHAFTS—SCAVENGING AND SUPERCHARGING—MOUNTINGS — FOUNDATIONS — MAINTENANCE, etc.—STANDARD SPECIFICATIONS — B.S.I. AND LLOYD'S RULES (pp. 215–251)

PART IV

GAS TURBINES

(By B. Wood, M.A. (Cantab.), A.M.I.Mech.E.) (pp. 253–267)

SECTION XXVIII

THE INTERNAL COMBUSTION ENGINE
(By R. J. W. Cousins, M.I.Mech.E.)

PART I

UNITS AND EQUIVALENTS—GAS LAWS, etc.

Units and Equivalents.

Length
1 metre = 100 centimetres = 1,000 millimetres.
1 metre = 39·37079 inches.
1 inch = 25·4 millimetres.

Area
1 square inch = 6·4516 square centimetres.
1 square centimetre = 0·155 square inch.

Volume
1 cubic inch = 16·387 cubic centimetres (c.c.'s).
1 litre = 1,000 c.c.'s = 61·03 cub. inches.
1 gallon = 8 pints = 277·3 cub. ins. = 4·5435 litres.

Weight
1 gramme = the weight of 1 c.c. of water at maximum density, *i.e.* at 4° C.
1 gramme = 15·432 grains.
1,000 grammes = 1 kilogramme = 2·2046 lbs. (lbs. av.).
1 pound avoirdupois = 7,000 grains = ·45359 kilogramme.
1 British ton = 2,240 lbs.
1 metric ton = 1,000 kilogrammes = 2,204·6 lbs.

Pressure
1 atmosphere = 14·7 lbs. per sq. in. = 1·03375 kilogrammes per sq. centimetre.
1 atmosphere = 30 ins. of mercury (approx.).
1 atmosphere = 760 millimetres of mercury (approx.).
1 atmosphere = 34 ft. of water (approx.).
1 kilogramme per sq. cm. ($kg/cm.^2$) = 14·22 lbs. per sq. inch.
1 inch of mercury = 0·49 lb. per sq. inch.

Mass
1 gravitational pound = 32·2 lbs. weight.
That is the mass (M) of a body equals its weight (W) in pounds divided by the acceleration due to gravity (g) which is 32·2 ft. sec. sec. (approx.).

$$M = \frac{W}{g} \text{ pounds (grav.).}$$

Velocity
1 foot per second = 30·48 centimetres per sec.
1 foot per second = 1·097 kilometres per hour.
1 foot per second = 0·6818 mile per hour.

Acceleration
1 foot per second per second (1 $ft./sec.^2$) means that the velocity increases at the rate of 1 ft. per sec. in every second.
1 $ft./sec.^2$ = 30·48 centimetres per sec. per sec. ($cms./sec.^2$).

Angle
1 complete circle = 4 right angles = 360 degrees = 2π radians.
1 radian = 57·3 degrees.

Angular velocity
1 radian per second = 9·549 revolutions per minute.

Temperature
Freezing point of water = 32° Fahrenheit = 0° Centigrade.
Boiling point of water at 14·7 lbs. sq. in. = 212° F. = 100° C.
Zero absolute = −460° F. = −273° C.
1° Centigrade = 1·8° Fahrenheit.

Heat
1 British Thermal Unit (B.T.U.) = heat necessary to raise 1 pound weight of water through 1° F.
1 B.T.U. = 778 ft. lbs. of work.

Power
1 British Horse Power = 33,000 ft. lbs. per minute.
1 British Horse Power = 42·416 B.T.U.'s per minute.
1 Metric Horse Power = 75 kilogrammetres per second.
1 Metric Horse Power = 0·9863 British Horse Power.
1 Watt = 1 ampere current at an E.M.F. of 1 volt.
1 Kilowatt = 1,000 Watts = 1·34 British Horse Power.

Gas Laws, etc.

The internal combustion engine is a prime mover in which fuel is burnt in the working cylinder or in a chamber directly communicating therewith.

The expansion of the gases resulting from the combustion then acts on the piston or its equivalent, and thus mechanical work is produced. It is therefore essential to know the way in which gases are affected by heat, pressure, etc., in order to understand approximately what goes on in the cylinder, and to make the required calculations.

It will be assumed, unless otherwise stated, that the working fluid behaves as a perfect gas.

***Pressure and Volume.*—Boyles' Law states that the volume of a mass of gas varies inversely as the pressure, provided the temperature is maintained constant.**

$$V \propto \frac{1}{P} \text{ or } PV = c \text{ (a constant).}$$

***Temperature and Volume.*—Charles' Law states that the volume of a mass of gas varies $\frac{1}{273}$ of its volume at 0° C. for every 1° C. change of temperature if the pressure be kept constant.**

$$V_T \text{ (volume at T° C. abs.)} = \frac{T}{273} V_0 \text{ when } V_0 = \text{volume at 0° C.}$$

i.e. the volume is proportional to the absolute temperature T and $\frac{V}{T}$ is constant.

If, however, the volume is maintained, the absolute pressure alters in proportion to the absolute temperature, *i.e.* $\frac{P}{T}$ is constant.

Combining the two $\frac{PV}{T}$ = a constant for any one gas.

If P is measured in lbs. absolute per sq. ft., V in cubic ft. per lb. weight at 0° C. and T in degrees C absolute then $\frac{PV}{T}$ = 96 for dry air.

When P = 760 millimetres of mercury (mm.s Hg) and T = 0° C., the gas s said to be at 'Normal Temperature and Pressure' (N.T.P.).

In practice, air forms the bulk of the working fluid in internal combustion engines, and therefore the behaviour of air under conditions of changing pressure and temperature forms a useful reference, and approximates to that of the charge in the cylinder.

***The Specific Heat* of dry air at constant pressure,**

$$k_p = 0 \cdot 2374 \text{ B.Th.U. per lb. weight.}$$

When allowed to expand as the temperature rises—the pressure being kept constant—work is done against the pressure of the atmosphere. The volume of 1 lb. wt. of dry air at N.T.P. is 12·387 cub. ft. and when raised in temperature 1° F. at constant pressure it expands $\frac{1}{492}$ of its volume against atmospheric pressure, *i.e.*, the external work is :—

$$\frac{12 \cdot 387 \times 14 \cdot 7 \times 144}{492} = 53 \cdot 29 \text{ ft. lbs.} = \frac{53 \cdot 29}{778} = 0 \cdot 0685 \text{ B.Th.U.}$$

When the gas is *not* allowed to expand, *i.e.*, when the *volume* is maintained constant while the temperature is raised 1° F., the pressure rises but the external work is not performed, and the heat required is then less by the amount calculated above, *i.e.* :—

$$0 \cdot 2374 - 0 \cdot 0685 = 0 \cdot 1689 = k_v.$$

This is the specific heat of air at constant *volume*.

The relationship between k_p and k_v is very important in internal combustion engines and compressors. It is usually written γ.

$$\frac{k_p}{k_v} = \frac{0 \cdot 2374}{0 \cdot 1689} = 1 \cdot 406 = \gamma.$$

Compression and Expansion of Gases.

When gases are compressed or expanded in an actual engine, the conditions are somewhat complex, but for simplicity two basic modes are recognised.

(*a*) Isothermal Compression or Expansion.

(*b*) Adiabatic Compression or Expansion.

Isothermal expansion means that the gas changes volume without changing temperature. This is the simple case assumed in Boyles' Law, so that when P alters to P_1 and V to V_1,

$$P_1V_1 = PV = c. \quad \text{(Fig. 1)}$$

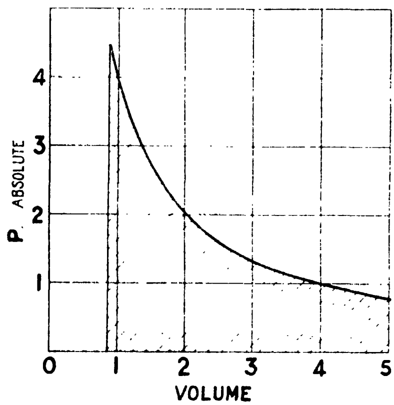

FIG. 1.

This only obtains in practice when the change is made very slowly, and adequate means are provided to maintain a constant temperature. In this mode of expansion, since the temperature is constant, the internal energy of the gas remains constant, as it is proportional to the absolute temperature.

The external work must therefore be supplied from the outside in the form of heat, and conversely in isothermal compression the heat which flows from the gas is equal to the work done in compressing it. This work, represented by the hatched part of fig. 1, is :—

$$144\ PV \log e \frac{V_1}{V} = W \text{ ft. lbs.}$$

P = lbs. per sq. in. V = volume cub. ft.

Adiabatic Expansion or Compression obtains when no heat flows either to or from the gas during the change in volume.

This is a hypothetical condition almost impossible to realise, since it requires a retaining cylinder and piston opaque to radiation and of zero conductivity. It is, however, a convenient and calculable standard.

Since no heat passes, it is clear that the gas must lose or gain internal energy by the amount of external work done by or put into it. This raises the temperature on compression and lowers it on expansion, and so affects the pressure directly ($P = cT$).

The equation then becomes

$$PV^n = P_1V_1^n \text{ where } n = \frac{k_p}{k_v} = \gamma = 1{\cdot}406 \quad \text{(See Specific Heat of Air.)}$$

With Isothermal compression or expansion $n = 1$, and the pressure is modified only by the volume change, but in most practical cases the value of n is between 1 and 1·4, and varies with the conditions governing the heat flow to or from the gas.

The action may be pictured mentally as taking place in two stages, first the pressure rise or fall due to volume change, and second, the additional pressure alteration due to the effect of changing temperature.

If R is the ratio between V and V_1 so that $\frac{V}{V_1} = R$; then

First $P_1 = PR^1$ volume effect only (isothermal change).
Second $P_1 = PR^{0.406}$ temperature effect;
Result $P_1 = PR^{1.406}$ for adiabatic change.

The External work involved is :—

$$\frac{144PV}{n-1}\left[1 - R^{n-1}\right] \text{ ft. lbs.}$$

added on compression and given out on expansion.

The absolute temperature change is :—

$$T_1 = TR^{n-1}$$

Actual Values of 'n.' When air is admitted to the cylinder of an engine, which has been running long enough to warm up, it receives heat from the walls, and even during the early stages of compression this process may continue for a short period. This raises the value of n above $\gamma(1 \cdot 406)$ and gives a more steeply rising pressure and temperature curve.

As the temperature of the charge rapidly approaches that of the cylinder walls the heat flow ceases and then reverses; 'n' then falls below the adiabatic value more and more as heat is lost to the walls, at a rate dependent on the temperature difference between charge and cylinder valves, etc., the ratio between exposed surface and volume, the amount of movement of the gases over the surfaces, and the density of the charge.

An examination of a large number of indicator cards has shown that the mean value of 'n' on compression and expansion varies greatly not only in engines of different design but in the same engine at different speeds and loads.

An extreme case gave a compression pressure rising constantly with speed even after the volumetric efficiency had begun to fall. These results were obtained on a special cell type of oil engine, where heat was given up by the hot throat on compression and taken in when combustion started. The expansion exponent was naturally affected in the other sense.

The difficulty experienced in getting sufficiently accurate cards at the low pressure end, and fixing T.D.C. on the diagrams introduces a source of error, but the recorded pressures and values of 'n' are given below.

Type of Engine.	R.P.M.	Comp. Ratio.	Comp. Pressure abs.	'n' Comp.	'n' Expansion.
Petrol sleeve valve	1500	7·0 to 1	205 lbs. sq. in.	1·35	
" " "	2100	7·0 " 1	212 "	1·36	1·28 approx.
" " "	2600	7·0 " 1	218 "	1·365	
Petrol poppet valve	1500	5 " 1	122 "	1·35	1·28 "
Cell type oil engine	650	16·8 " 1	550 "	1·30	
" " " "	1500	16·8 " 1	640 "	1·34	1·2 "
" " " "	2000	16·8 " 1	693 "	1·36	
Slow speed oil engine	450	15·5 " 1	550 "	1·34	1·24 "
" " " "	400	15·0 " 1	452 "	1·327	—
Poppet valve alcohol engine	1500	6·7 " 1	180 "	1·33	1·26 "
Cell type oil engine, hand turned at 0° C. with brine circulated in the jacket and air intake	100 approx.	19·0 " 1	520 "	1·22	—

The last example illustrates the great difference in the 'n' due to slow speed and a cold engine. This accounts for the difficulty sometimes experienced in starting when :—

1. The speed is slow and the time for heat loss consequently great.
2. The initial temperature is low as both air and inlet passages are cold.
3. There is no heat pick up in the early part of the stroke, but loss all the way.

Owing to the difficulty of determining when burning ceases, the values cannot be given accurately for the early part of the expansion curve.

Compression Ratio.

In spark ignited engines the ratio R between the volumes at bottom and top centres ranges from 4 or less up to 10 to 1, while in compression ignition engines (where the terminal compression pressure is sufficient to ignite finely divided fuel without any external aid) the ratio may be as low

as 11 to 1 and as high as 20 to 1. A rough guide to the ratio suitable for various cases is given in the table below:—

TABLE OF APPROPRIATE COMPRESSION RATIOS.

Note.—The compression ratio is the volume of the enclosed space at bottom centre divided by that at top centre.

Type of Engine.	Speed R.P.M.	Fuel.	Octane No.	Comp. Ratio.
Small commercial or boat engines	Low 750–1,000	Kerosene (paraffin)	45 to 50	3·6 to 3·8
Small commercial or boat engines	Low to Medium 1,000–1,500	Special 'Power Kerosene	55 to 65	4·0 to 4·8
Commercial vehicles (large)	Medium 2,000	Commercial grade petrols	65 to 70	5·0 to 5·3
Commercial vehicles (small)	Moderately high 2,500	Commercial grade petrols	65 to 70	5·3 to 5·5
Car engines (large)	3,500	No. 1 Petrol	75	5·6 to 6·0
,, ,, (small)	4,000	No. 1 Petrol	75	6·0 to 6·5
,, ,, (sports)	5,000	Leaded Petrol	80	6·5 to 7·5
Track motor cycles	6,000 to 7,000	Special alcohol and benzol fuels	100 to 105	up to 12
Gas engines	Low or medium speeds	Town gas	—	6·0 to 6·5
,, ,,	Low or medium speeds	Natural gas	—	7·5
,, ,,	Low or medium speeds	Producer gas	—	7·0
Large engines	Low speed	Furnace gases	—	7·0 to 8·0
Oil engines (large)	Low speed	Residual oil (air injected)	0·89 sp. gr.	12 to 14
,, ,, ,,	Low speed	Distillates (airless injection)	0·86 to 0·87	12 to 14
Oil engines (medium)	medium	Distillates (airless injection)	0·86 to 0·87	14 to 17
Oil engines (small) (open chamber or cell types)	High speeds 2,000 to 3,500	Gas oil (airless injection)	0·86 sp. gr.	14 to 19 or even 20 in very small engines.

SPECIFIC VOLUME.

Dalton showed that at the same conditions of temperature and pressure all gas molecules occupied the same volume.

When a charge of air and fuel is burned the atoms rearrange themselves and combine with oxygen to form new molecules which may be more or less in number than those in the original charge.

The nitrogen, carbon-dioxide and water in the air will not be changed, so only the oxygen need be considered at this point.

$$2\,H_2 + O_2 = 2\,H_2O$$
2 volumes + 1 vol. = 2 vols. (at the same T. and P.).
3 vols. = 2 vols.

Benzene
$$2\,C_6H_6 + 15\cdot O_2 = 12\,CO_2 + 6\,H_2O$$
2 vols. + 15 vols. = 12 vols. + 6 vols.
17 vols. = 18 vols.

Nonane
$$CH_3(CH_2)_7CH_3 + 14\cdot O_2 = 9\,CO_2 + 10\,H_2O$$
1 Vol. + 14 vols. = 9 vols + 10 vols.
15 vols. = 19 vols.

Oxygen forms 21 per cent. of the atmosphere by volume, the other constituents being merely diluents as far as combustion is concerned.

The alteration in specific volume of the whole charge is therefore not so great as that of the active constituents :—

e.g.

Nonane	1 vol.		
Oxygen	14 vols.	21 per cent.	} 100
Remaining air . .	52·7	79 per cent.	
Volume . . .	67·7 before burning.		
Carbon dioxide . .	9 vols.		
Water Vapour . .	10 vols.		
Unchanged parts of the air	52·7		
	71·7 after burning.		

Increase, 67·7 to 71·7 = 5·7 per cent.

Water Vapour in the Air.

When vapour is in an enclosed space in contact with the liquid from which it is produced, the pressure of the vapour rises to a point which is dependent solely on the temperature, and this is the same whether the space was originally vacuous or filled with some gas such as air. This pressure is generally called the 'vapour tension.' When fresh air contains the maximum amount of water vapour for the temperature, it is said to be 'saturated,' and as the total pressure remains the same, it is clear that the pressure due to the air alone is reduced by the value of the partial pressure due to the vapour tension. Therefore, in saturated air the weight of oxygen available is reduced below that in dry air at the same temperature and pressure.

Taking dry air at N.T.P. = 0·0807 lb. per cub. ft. as the standard, the weight of dry air per cubic foot of saturated air at temperature T° C. and pressure P is as under :—

$$\frac{P - P_w}{14\cdot7} \times \frac{273}{T^\circ \text{ C. (abs.)}} \times 0\cdot0807 \text{ lb. per cub. ft.}$$

when P_w is the vapour tension for temperature T from the tables. To find the total weight per cub. ft. add the weight of the water vapour (see table, p. 193).

Data relating to the Vapour Tension of Water from 32°F. to 212°F.

At °F.	Absolute Pressure, in Lbs. per Sq. In. P′.	Cu. Ft. Occupied by 1 Lb. u.	Latent Heat, in B.Th.U. per Lb.	At °F.	Absolute Pressure, in Lbs. per Sq. In. P′.	Cu. Ft. Occupied by 1 Lb. u.	Latent Heat, in B.Th.U. per Lb.
Col. :	1	2	3	Col. :	1	2	3
32	0·089	3,285	1070·5	130	2·209	158	1017·9
40	0·122	2,455	1066·3	140	2·871	123	1012·3
50	0·177	1,708	1060·9	150	3·699	97·0	1006·7
60	0·253	1,210	1055·7	160	4·719	77·5	1001·1
70	0·359	870	1050·3	170	5·970	62·5	995·3
80	0·502	640	1045·1	180	7·492	50·5	989·5
90	0·691	466	1039·7	190	9·310	41·0	983·7
100	0·939	353	1034·3	200	11·496	33·6	977·8
110	1·264	268	1028·9	210	14·09	27·8	971·6
120	1·681	203	1023·3	212	14·70	26·75	970·4

Vapour Tension of Liquid Fuels.

The remarks on vapour tension of water apply also to liquid fuels used in carburettor engines, *i.e.*, drawn in along with the air. These uels are very complex and tend to become more so owing to modern methods of treatment. The boiling points of their constituents cover a wide range, but one of the most important points to the ordinary user, is that the fuel should contain ingredients having a sufficiently high vapour tension at ordinary temperatures to produce an explosive mixture in the cylinder. When the engine is hot the fuel will evaporate, and if special heating means are provided a heavy kerosene will function, but it is impossible to start on it from cold.

DATA RELATING TO AIR SATURATED WITH MOISTURE AT ATMOSPHERIC PRESSURE. ESTIMATED FROM 32°F.

Temperature. t°F.	Per 1,000 cu. ft. of Saturated Air.			Cu. ft. of Saturated Air, per Lb.	Cu. ft. of Dry Air, per Lb.	Density ratio of Saturated to Dry Air.	Heat of 1,000 cu. ft. of Sat. Air from 32°F. at t°F.				B.Th.U. per Lb. of Saturated Air.
	Lbs. of Dry Air.	Lbs. of Water Vapour.	Total Weight in Lbs.				B.Th.U. of Heat of Dry Air.	B.Th.U. of rise of Temp. of Water.	B.Th.U. of Vaporisation of Water.	Total B.Th.U. per 1,000 Cu. Ft.	
Column	1	2	3	4	5	6	7	8	9	10	11
32	80·241	0·3046	80·546	12·415	12·387	0·998	0	0	326	326	4·05
40	78·779	0·4073	79·186	12·628	12·56	0·995	149	3·26	434	586	7·4
50	76·943	0·5855	77·529	12·898	12·84	0·995	329	10·54	621	961	12·4
60	75·068	0·8264	75·894	13·176	13·09	0·993	499	23·14	872	1,395	18·4
70	73·112	1·1494	74·261	13·466	13·34	0·991	660	43·7	1,207	1,911	25·7
80	71·042	1·5625	72·605	13·773	13·60	0·987	810	75·0	1,633	2,518	34·7
90	68·821	2·146	70·967	14·091	13·85	0·983	948	124·5	2,231	3,304	46·6
100	66·396	2·833	69·229	14·445	14·10	0·976	1,072	192·6	2,930	4,195	60·6
110	63·691	3·731	67·422	14·832	14·35	0·968	1,179	291·0	3,839	5,309	78·7
120	60·650	4·926	65·576	15·250	14·60	0·957	1,267	433·5	5,041	6,741	102·8
130	57·204	6·329	63·533	15·740	14·85	0·943	1,331	620·0	6,442	8,393	132·1
140	55·269	8·130	61·399	16·287	15·11	0·928	1,366	878	8,230	10,474	170·6
150	48·729	10·309	59·038	16·938	15·36	0·907	1,365	1,217	10,378	12,960	219·5
160	43·498	12·903	56·401	17·730	15·61	0·880	1,322	1,652	12,917	15,891	281·8
170	37·442	16·000	53·442	18·712	15·86	0·848	1,227	2,208	15,925	19,360	362·2
180	30·431	19·802	50·233	19·917	16·11	0·809	1,069	2,931	19,594	23,594	470·0
190	22·406	24·390	46·796	21·369	16·37	0·766	840	3,854	23,992	28,686	613·0
200	13·117	29·762	42·879	23·322	16·62	0·713	523	5,000	29,101	34,624	807·5
210	2·460	35·971	38·431	26·021	16·87	0·648	104	6,403	34,949	41,456	1078·7
212	0·000	37·383	37·383	26·750	16·92	0·632	0	6,729	36,277	43,006	1150·4

EFFICIENCY.

The thermal efficiency of an engine is the relationship between the energy delivered in mechanical work and the total heat value of the fuel used.

The Air Standard Efficiency,

$$E = 1 - \left(\frac{1}{R}\right)^{\gamma - 1}$$

assumes no heat transfer to or from the gas, complete and instantaneous combustion at constant volume, no change in specific volume, and constant specific heat over the whole temperature range.

None of these conditions obtains in practice, but this ideal figure forms a useful comparison and is a guide to the variation which may be expected by change in compression ratio R. It will be clear that the higher the ratio, the greater the efficiency will be, but the rate of increase falls off rapidly and several practical considerations tend to put a limit on the figure which may usefully be employed. Fig. 2 gives the values for ratios 2 to 22.

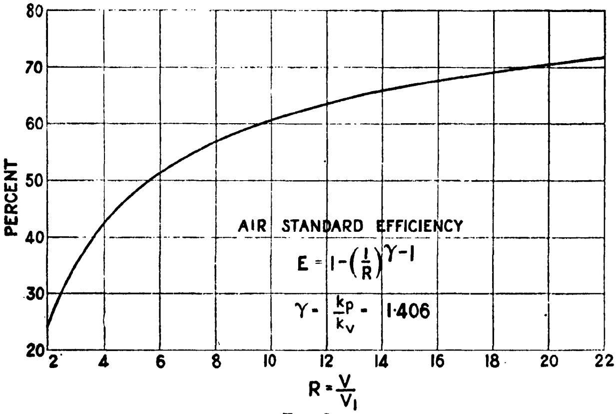

FIG. 2.

THE CONSTANT VOLUME CYCLE.

This cycle follows as far as possible the above conditions, and if they could be realised the diagram would be as fig. 3, with temperatures at the important points as indicated.

The fuel heat is assumed to be added instantaneously without loss at T_2 raising it to T_3. $T_3 - T_2$ then = 100 per cent.

The rejected heat at the end of the cycle = $T_4 - T_1$ and is lost, therefore the efficiency may be written—

$$E = \frac{(T_3 - T_2) - (T_4 - T_1)}{(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \left(\frac{1}{R}\right)^{\gamma - 1}$$

This is the cycle aimed at in the ordinary gas and petrol engines working on the Otto cycle, but there are many quantitative departures from the ideal as under:—

1. Heat is lost in the later stages of compression making T_2 lower.

2. The heat energy is not added instantaneously (the nearest approach to this is violent detonation which soon leads to pre-ignition and cannot be tolerated in practice). The fuel begins to burn before top centre and continues for an appreciable time during expansion.

Since it has been shown that the efficiency is dependent of the expansion ratio, it is evident that the fuel which burns later is not used as economically as that burned at top centre.

3. The specific heat is actually higher at higher temperatures, so that the addition of a certain quantity of heat does not produce the rise in temperature (and therefore pressure) which would result if specific heat were constant.

4. Dissociation of the products of combustion CO_2 and H_2O into CO, H_2 and O_2 occurs at high temperatures. This absorbs heat in the early stages, and recombination later on raises the terminal temperature and therefore the amount of rejected heat.

5. Heat loss, with consequent falling off in pressure, occurs during burning and expansion and as it is greatest at the highest temperature, this loss produces its effects throughout the working stroke.

6. Since the process of evacuating the cylinder takes some time, the exhaust valves or ports must be opened well before the end of the stroke, thus lowering the real expansion ratio.

This all sounds dreadfully disheartening, but the efficiencies obtained are still so good that only the best and largest steam engines can approach the poorest kind of internal combustion engine in fuel consumption.

The Constant Pressure Cycle **consists of an adiabatic compression, addition of heat energy at constant *pressure*, adiabatic expansion and rejection of waste heat at constant volume. The ideal diagram for this cycle is as fig. 4 :—**

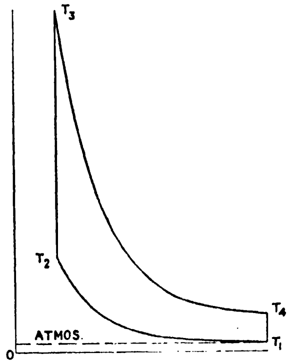

FIG. 3.

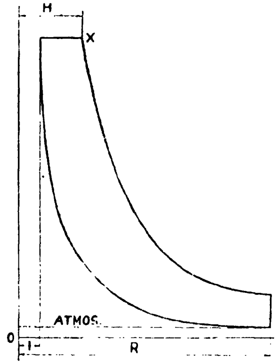

FIG. 4.

The compression ratio is R, the end of heat input X at a volume H (where the clearance equals 1).

The efficiency is $E = 1 - \left(\dfrac{1}{R^{\gamma} - 1} \times \dfrac{H^{\gamma} - 1}{\gamma(H-1)}\right)$

Apart from the inevitable heat losses to the jacket, etc., previously mentioned, this cycle is closely approached in large slow-speed Diesel engines, where the quantity of fuel is sufficiently great to permit the rate of injection to be controlled throughout the burning period, but in the higher speed smaller engines, the departure from the cycle becomes more marked, and in the modern vehicle oil engine there is a rapid rise of pressure above compression pressure, so that the cycle is really a compromise between the two—constant volume and constant pressure.

In the larger engines the maximum pressure and temperature are deliberately kept down for practical constructional reasons, but in the smaller sizes more liberty may be taken, and it is found, as may be expected, that efficiency is improved if the injection timing and rate are such as to give higher peak pressures for the same total fuel per charge.

Naturally the fuel burnt later in the stroke is not used so efficiently, but the compression ratios used in engines working on this cycle, where the fuel is ignited by the heat of compression alone, are so high that even the last of the burning charge is expanded more than in spark ignited engines, and consequently the efficiency of oil engines in general is higher, particularly on lighter loads, where the mean temperature is lowered by reducing the fuel while still admitting a full charge of air.

APPROXIMATE MAXIMUM BRAKE THERMAL EFFICIENCIES.

(At the most favourable speed and load.)

Small petrol 2-strokes	14 to 16 per cent.
Small petrol car engines	23 to 26 ,,
Aero engines	30 to 32 ,,
Small high-speed oil engines	34 to 36 ,,
Medium size open chamber engines . . .	36 to 38 ,,
Large oil engines—airless injection . . .	37 to 41 ,,
Producer and furnace gas engines . . .	25 to 28 ,,
Coal gas engines	30 ,,

VOLUMETRIC EFFICIENCY.

The power developed by an engine is in every case dependent on the weight of oxygen it can handle in unit time.

This is particularly the case in petrol engines, which often run slightly 'rich,' *i.e.* with a fuel to air ratio in excess of true chemical balance.

Care must be taken therefore that the valves or ports for admitting air are of sufficient area to pass the required volume without 'wiredrawing' and correctly timed so that the maximum amount of air is retained in the cylinder at the beginning of the compression stroke. Light spring diagrams are very revealing on this point, and should always be taken on engines of new design. In addition to the pressure obtaining, there is also the question of temperature, and in engines where air and fuel are drawn in together the latent heat of the fuel (petrol, benzol or alcohol) has an important effect, as the cooling of the charge offsets to some extent the temperature rise during the suction stroke and thus causes a greater weight of air to be retained when the inlet valve closes. For example, actual tests have shown a rise of from 7 to 10 per cent. in volumetric efficiency when alcohol has been substituted for petrol, thus demonstrating the importance of the temperature effect.

While it is common practice in carburettor engines to heat the incoming charge in order to provide a more homogeneous mixture, and so distribute the fuel evenly to the several cylinders, it is essential that this heating should be kept to a minimum, if maximum power is important.

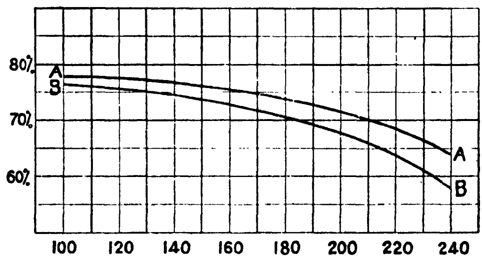

FIG. 5.

Volumetric Efficiency in Terms of Standard Pressure and Temperature.
A. Average of a large number of tests with valves in head.
B. Average of a large number of tests with valves in side pockets.

The following table, giving weights, volumes and latent heats, may be found useful.

Substance.	Symbol.	Specific Gravity as a Gas. Hydrogen = 1.	Weight of 1 Cub. Ft. as a Gas at N.T.P.	Latent Heat of Evaporation B.Th.U.'s per lb. (G = gas at normal temp.)
Hydrogen	H_2	1	0·00559	G
Methane (Marsh gas)	CH_4	8	0·04472	G
Water	H_2O	9	0·05031	971·7
Carbon monoxide	CO	14	0·07826	G
Carbon dioxide	CO_2	22	0·12298	G
Ethylene	C_2H_4	14	0·07826	G
Normal benzene	C_6H_6	39	0·21801	172
Hexane	$CH_4(CH_2)_5$	43	0·24057	156
Heptane	$CH_4(CH_2)_6$	50	0·27950	133
Octane	$CH_4(CH_2)_7$	57	0·31863	128
Acetylene	C_2H_2	13	0·07267	G
Ether (Ethyl)	$C_2H_5OC_2H_5$	37	0·20683	—
Ethyl alcohol	C_2H_5OH	23	0·12857	32
Methyl ,,	CH_3OH	16	0·08944	51
Cyclo hexane (Naphthene series)	C_6H_{12}	42	0·23478	156
Oxygen	O_2	16	0·08944	G
Nitrogen	N_2	14	0·07826	G
Air	mixture $O_{22}N_{77}$ (approx.)	14·44	0·08073	G

SECTION XXVIII

PART II

GENERAL DESCRIPTION OF TYPES OF ENGINES AND FUELS — CONSTRUCTIONAL DETAILS PECULIAR TO LARGE ENGINES—NOTES PARTICULARLY APPLICABLE TO HIGH SPEED ENGINES.

(By R. J. W. Cousins, M.I.Mech.E.)

General Description of Types of Engines and Fuels

Internal Combustion Engines may be classified in several ways of which the broadest are as follows :—

(1) The Heat Cycle.—Constant Volume or Constant Pressure.

(2) The Mechanical Cycle.—Four-stroke, Two-stroke, etc.

(3) Single-Acting or Double-Acting.

(4) Trunk Piston type or Piston Rod and Crosshead.

(5) Method of Cooling.—Air, water, etc.

(6) Cylinder Arrangement.—Line, vee, fan, radial, horizontal, vertical, opposed piston and H types.

(7) Type and arrangement of valves used.—Overhead or side poppet valves, sleeve valves, piston valves or ported cylinders. Also special types using rotary or sliding valves.

(8) Method of Firing.—Spark ignited or compression ignition.

(9) Fuels used.—Gas (town producer or furnace), volatile liquids (petrol, alcohol, benzole and kerosene); non-volatile fuels (heavy oils and distillates).

(10) Method of injection of fuel oil.—Air blast or 'solid' injection.

(11) Design of the combustion chamber for compression ignition engines.—Open chamber, hot bulb low compression, and recent high compression cell types.

(12) High and low speed types.—The former generally being of lighter and more refined design, while the latter are heavy and either of cheaper and simpler construction, or intended for a very long working life.

(1) *Heat Cycle.*

(*a*) *Constant Volume Cycle.*—This includes most engines running on town, producer, furnace or natural gases, petrol and paraffin, but as previously explained the cycle is not followed strictly.

(*b*) *Constant Pressure Cycle.*—The larger oil engines approximate to this cycle when the fuel is injected by an air blast, but the 'solid' injection types, including all the small engines, have a cycle which is a compromise, the pressure of combustion rising above that of compression.

The operations in either case may be concluded in four-strokes (two revolutions) or two-strokes (one revolution).

(2) *Four-Stroke Cycle.*

Engines draw in air on the outward movement of the piston, compress on the next stroke, burning occurs at or near the inner dead centre and expansion follows on the third stroke near the end of which the exhaust valve is opened, the burnt gases being swept out on the fourth stroke.

The question of valve areas to attain the best results is dealt with later under the headings 'Inlet' and 'Exhaust Valves.'

In some large engines working on low value fuel, such as furnace gases, air from a low pressure blower (1½ to 2 lbs. per sq. in.) is admitted towards the end of the exhaust stroke, to assist in clearing the combustion space of dead gases, thus preventing further dilution of the next charge.

The Four-stroke Cycle, first used by Dr. Otto, after whom it is named, is by far the most popular, practically all car, truck and aero engines, as well as the bulk of small and medium commercial engines, being of this class.

The Two-stroke Cycle.—In this the charge is compressed on the inward stroke, burning occurs near the inner centre and expansion follows for a large part of the next stroke. Near the end of this second stroke, large ports or valves are opened and the used gases are allowed to escape, so that the pressure in the cylinder falls to a low figure before the stroke is completed. Other ports are then opened, allowing air from a blower to enter, and the time of opening, size, and direction

of the ports must be such that the residual exhaust is largely swept out and replaced by a clean charge before the piston has travelled far on the next (third) stroke, which then becomes the compression stroke for the second cycle.

The relationship of swept volume, speed and port areas is dealt with later.

Curiously enough, two-stroke engines are usually found either in the very small class, such as is used in light motor cycles, lawn mowers, etc., using the underside of the piston and the crankcase as an air pump and running on a petrol-oil mixture, or in the very large class, mostly marine engines running on heavy oil and with elaborate scavenging arrangements. The reason, no doubt, is that the very simple ported cylinder crankcase compression two-stroke engine, while very convenient in small sizes, is not economical in fuel, and introduces many problems, thermal and otherwise, when made in larger sizes, and it is usually not worth while providing the blower equipment, piston cooling, etc., until such dimensions are reached that a considerable overall saving in weight, etc., is effected by the doubling of the working strokes. The two-stroke engine is now very popular for marine work in both single- and double-acting forms.

(3) *Single- or Double-Acting.*

While most small and medium size engines are single-acting, *i.e.*, use the upper side of the piston only, a large number of marine engines are made with piston rods and lower cylinder covers fitted with glands, so that the underside of the piston may also be used. The glands are formed of nests of contracting rings of the same material as piston rings, each one contained in a rebated distance ring and arranged to have lateral clearance so that they do not carry any force radial to the piston rod. For reasons of accessibility, double-acting engines are generally only found in the larger classes.

(4) *Trunk Pistons.*

which are attached to the connecting rod by the gudgeon or piston pin and carry the side thrust on their skirts are the only form in use on small engines, but many large land engines and most early marine units used piston rods with single-acting engines, the crank chamber being closed by an oil stripping gland and the lower end of the piston rod attached to a crosshead to take the side forces.

This makes a very high, heavy engine, and in consequence there is a swing towards trunk pistons for some marine sets, where the saving of weight and head room are considered of more value than the lower oil consumption of the piston-rod type.

In general, trunk piston engines are of shorter stroke in proportion to the bore, and run at higher revolutions. Many are now under construction in Continental yards for use with reduction gears so as to increase the engine speed without sacrificing propeller efficiency.

(5) *Cooling.*

It is necessary to cool the cylinder walls so that they may be at a sufficiently low temperature to retain the lubricant, and the head and exhaust valve seat must also be cooled to prevent undue thermal stresses and distortion, softening, or burning of the parts.

Air cooling is used on almost all motor-cycle engines, aero radials, and the smaller aero line engines. It is found that the increased powers obtainable from modern fuels necessitate a proportional increase in the fin surfaces, which are now usually fully machined in the form, depth, and pitch found most effective.

Some aero engines use a high boiling point liquid like ethylene glycol on account of the reduction of radiator area rendered possible by the higher temperature (boiling point 197° C.). Steam cooling in radiators fitted with loaded safety valves is also used. The water and steam are separated at the engine outlet, the steam going through to the cooling surfaces so as to be condensed, while the water is circulated rapidly through the jackets again.

Most engines, however, are water-cooled.

Radiators on aero, vehicle and small commercial engines return the same water to the jacket repeatedly, and are to be preferred to the method employed in other cases of supplying new water continuously, for two reasons—

The new water tends to deposit more and more lime, etc., in the jackets if 'fresh.' If sea-water it must be kept below about 55° C. (120° F.) to prevent deposition of salt.

In either case the new water usually enters the cylinder jacket first and keeps it cold, the worst possible condition for bore wear, as it causes the condensation of small quantities of acid products on the cylinder bore.

It is better to use 'fresh' (non-salt) water in a closed circuit with the sea-water passing through a 'heat interchanger.' In this way a very rapid circulation may be kept up in the cylinder head where the greatest differences of heat input exist between the inlet and exhaust valve seats, etc., and where the directing of a large quantity of fairly hot water in the right places prevents great temperature differences in the various parts of the metal.

The jacket round the cylinder should be kept as hot as possible without boiling. This reduces wear by preventing deposition of water vapour and acid products on the walls, and also reduces the viscosity of the cylinder lubricant and thus the frictional losses.

This method is receiving increasing attention as speeds are rising, and wear becoming more important. In large engines the pistons are also cooled, sometimes by water, but increasingly by oil circulation, as corrosion fatigue of piston rods has been experienced with water.

(6) and (7) *Cylinder Arrangement and Types of Valves.*

The first step from single horizontal cylinder engines to vertical cylinder type to save space, was followed naturally by increasing the number of cylinders on the same line. This vertical in-line type now covers most vehicle, marine and power station engines as well as the small commercial class, the number of cylinders ranging from two to ten or even twelve. The crankshaft is one of the main problems, and care must be taken that forces and couples are balanced and that, over the running range, the engine is free from dangerous synchronous vibrations (see crankshafts).

The 'Vee' engine is well entrenched in aero work, and is coming into the field for cars, medium size high-speed generator drives and railcars. It provides two lines of cylinders in practically the same length and very little more width. The crankshaft is of course shorter for the same power, thus simplifying the torsional problem. The member requiring the most attention is the big end, most other parts are as in the line engine.

The fan engine—three banks of cylinders on one crankshaft—is not so well represented The big end, which must be split on a multi-crank engine, presents a difficult problem in design and manufacture. Radial engines of five, seven and nine cylinders on one crank, or fourteen and eighteen on two cranks, dominate the air-cooled aero engine business, except in the smaller powers.

The big advance in power due to the better fuels has been met in the air-cooled radials by more liberal finning and the use of the unsplit big end and built-up crankshaft. One master connecting rod carries a banjo end with side-flanges between which work the wrist-pin connections of the articulated rods. The frontal area of these engines is greater than that of the 'Vee' type for the same output, but by careful design the overall diameter has been reduced while powers have risen, and advanced methods of cooling the cylinders and exhaust ring, and controlling the flow of cooling air, have cut down the head resistance to a very low figure.

The H engine is an aero type of comparatively new development, and consists of four banks of cylinders, usually six in each line. Two banks, one above and one below, act on each of the two crankshafts which are geared to a slower speed propeller shaft in the centre. The basic idea is to get more cylinders of small size on to one engine and thus raise the speed at which the required power is developed, so that the power weight ratio is kept down. In addition to this, small pistons and exhaust valves do not present such formidable thermal problems at high outputs as large ones.

Opposed Piston Engines.—These are made with two pistons back to back forming the combustion chamber. They work on the two-stroke cycle, and one piston is given a slight lead over the other, so that the exhaust ports, which it uncovers towards the end of the working stroke, will open in advance of the scavenge or air inlet ports, which are controlled by the second piston. The Junkers aero engine of this type, has two crankshafts geared together by a vertical train of spur gears, while the Doxford marine two-cycle engine has the upper piston connected to the same crankshaft as the lower by side rods. Swash-plate engines have also been made on this principle. The type in general has its own particular design problems, which have called for a good deal of development work—notably the exhaust piston and the connecting of the two pistons to the point at which power is taken off, but it has two very attractive features, first the elimination of the cylinder heads, which means a reduction of the heated surfaces, and second the long combined stroke to bore ratio which makes for very good scavenging. Excellent efficiencies are obtainable.

Valves.—The majority of engines use the well tried poppet valve. The old T-head engine with the exhaust valve in one side pocket, and the inlet in the other, was a bad detonator and required two camshafts. It is dead and not likely to be resurrected. Both valves in one compact chamber at the side are popular for petrol engines in cars and lorries, also for small commercial engines. Overhead valves operated by push rods are common on vehicle oil engines, and overhead camshafts on aero and fast car engines.

Reduction of deflection and weight of moving parts, careful cam and spring design and the cooling of the exhaust valve seat are the important points.

Large engines use removable valve cages, but these are not found good on small engines, where the separate exhaust seat cannot be properly cooled.

The single sleeve valve engine has many attractive points, not the least being the absence of local hot spots and the high compression ratio rendered possible by this. At least one aero engine is a definite success, and this system may come to the front for particular applications. Wear of the sleeve is remarkably low and high speeds are obtainable.

Rotary valve engines have attracted designers for many years, but their seeming simplicity is deceptive—they usually leak or seize if made in elementary form, and success has only been reached by certain proprietary designs where very thorough work has been put into the elimination of their troubles.

Most two-stroke cycle engines use ports in the cylinder uncovered by the piston with, in some cases, supplementary valves in the exhaust or scavenge belt to give correct timing. Certain large two-stroke engines, both single and double-acting, use piston valves at the end of the cylinder for the exhaust release. These are controlled by very robust gear, as they are subject to full gas pressure and in fact produce quite a respectable percentage of the total horse-power (about 9 per cent. to 10 per cent.). In smaller engines poppet valves may be used in the same way for the exhaust. Centrally placed scavenge ports open alternately above and below the piston in

the double-acting engines of this type, and as the whole circumference is available for air, high outputs and good piston speeds are possible. In all valve systems the first essential requirement is that the area of opening multiplied by the time, shall be enough to pass the necessary weight of gas with the available pressure difference across the port or valve. If this is not satisfied, the engine will be handicapped. Methods of approximating to the areas needed will be given under valve details.

(8) *Method of Firing.*

There are broadly three methods of firing :—

(*a*) Electric ignition by the passing of a spark between two points in the combustion chamber. This is now done by a momentary high voltage of the order of 10,000 volts applied to the points of a sparking plug by a magneto, or by a coil and distributor working from a battery. The latter gives a better spark at starting speeds. At one time a low voltage was applied to a robustly made 'make and break' device in the cylinder, the contact being broken and an arc thus produced by a trip gear working from the camshaft. Before that, an incandescent tube was so arranged that some explosive mixture entered it at the right moment, but this device disappeared very soon.

(*b*) Surface Combustion or Hot Bulb. In this system a comparatively low compression engine is fitted with an uncooled bulb or chamber forming part of the combustion space. For starting, the bulb is made red hot by a blowlamp, and afterwards the heat of combustion keeps it hot enough to fire the charge, although the compression pressure is only about 200 lbs per sq. in. These engines are of low efficiency but cheap, as the low pressures and speeds do not call for advanced design or expensive materials. They are disappearing before the more efficient and higher speed high compression oil engine.

(*c*) The third method is to design the engine with a compression ratio so high that the temperature of the air at the end of the stroke is high enough to fire the fuel when the latter is injected by pump or air blast. A variety of fuels may be used in this way, since their tendency to detonate does not impose a limit as it does in the spark ignited engine.

The long expansion range gives high efficiency and this too on cheap fuels.

Fuels not volatile at ordinary temperatures are used. Engines firing in this way are now made in sizes from about 400 c.c. capacity per cylinder (a four-cylinder taxi engine has recently been produced in this size) up to marine and power station plants with cylinders about 36 bore and several thousand horse-power per line.

(9) *Fuels Used.*

This is perhaps one of the most important ways of classifying engines, since the design must start from this point, and an engine arranged for high duty on a particular fuel may simply not work on a lower grade, while one designed for low-grade fuel will not use the better stuff economically.

In spark ignited engines the fuel and air are mixed on the suction stroke and a homogeneous explosive charge is compressed. When the spark passes, flame spreads through the charge. The combustion of the first part of the charge heats and expands it, thus compressing and further heating the unburnt part. If the shape of the chamber is not compact, or if the unburnt part is squeezed into a hot pocket, for example over the exhaust valve, it may explode spontaneously throughout its mass, causing a detonation wave to go through it at the high speed appropriate to its density and pressure. This causes a sharp noise like the blow of a light hammer and is called 'knocking' or 'pinking' (the noise sounds something like the word 'pink' repeated quickly). This quickly leads to real preignition, rapid loss of power and probably piston seizure from overheating. Therefore, persistent knocking cannot be tolerated. Much can be done by designing the combustion chamber as compactly as possible (a sphere without hot zones and a spark appearing magically in the centre is the unattainable ideal), by firing the charge so that the flame travels towards cooler and not hotter zones, and by giving a certain amount of movement to the gases to facilitate flame spread, but when this has been done, the fact remains that some fuels are more stable chemically and will work under control (*i.e.* without spontaneous detonation) up to very high compression ratios, while others (notably the heavy paraffins) being what are called 'chain' compounds (the construction of the molecule is like that of a long train of heavy waggons held together by weak couplings) break up easily and 'knock' on the smallest provocation. If pre-heating is necessary (as for kerosene) this makes matters worse. The compression ratio in spark-ignited engines must therefore be low enough to prevent the particular fuel from knocking, but if high-grade fuel is used in a low-compression engine, practically no advantage is gained. The heat value of the available hydrocarbons does not vary much. *The real value of a fuel for spark ignition engines lies in its ability to work at a high ratio and therefore at a high efficiency*, and in support of this it is found that greater power may be obtained from alcohol blends, although the heat value is actually less (due to the combined oxygen). This result comes about thus :—

(*a*) The very high latent heat of alcohol cools and contracts the air, so that a greater weight of air is retained at the beginning of compression.

(*b*) A very high compression ratio may be used and the extra air is used at a high efficiency. (See tables of appropriate compression ratios in Part I.)

The 'lower' heat values of some fuels are given below. These do not include the latent heat of the water vapour produced as it is not available at the working temperature.

	Per lb.	B.Th.U.s Per gall.	Sp. Gr. at 60° F.
Petrol (mean of several samples)	18,890	136,600	0·725
Benzene, pure	17,300	153,000	0·884
Heavy aromatics	17,900	158,500	0·885
Pentane (normal)	19,600	122,300	0·626
Kerosene (approx. value)	19,000	154,400	0·812
Cracked spirit (approx. value)	18,400	139,400	0·760
Cyclo hexane	18,800	146,600	0·780
Ethyl alcohol 98 per cent.	11,480	91,600	0·796
Methyl alcohol	9,630	79,900	0·830
Methylated spirits	10,200	83,700	0·820
Gas oil 0·86 sp. gr. (for compression ignition)	18,300	157,200	0·860

It will be seen from this table that if fuel is bought by the gallon, a high specific gravity may be an advantage—provided the knock rating of the fuel is high enough for the engine.

The knock rating was first quoted as the percentage of toluene required to be mixed with a standard non-aromatic petroleum spirit in order to give the same behaviour in a variable compression engine as the sample in question. The system of 'Octane numbers' has since been adopted. In this system pure iso-octane has been taken as 100 and fuels are classified as 73–87–100, etc., according to their approach to this fuel in detonation resistance. The higher the number the greater the compression ratio at which the fuel will work, and consequently the better the efficiency. For compression ignition engines, on the other hand, *a readily disrupted molecule is not only desirable but to some extent necessary.* For example, tar oil and creosote are very difficult to burn well in engines, as their stability is great, and their ignition temperatures very high. Kerosene—a poor fuel in spark-ignition engines, in fact quite at the bottom of the class—is good in a compression ignition engine, but a slightly heavier petroleum distillate—gas oil—is preferable in many ways, being cheaper, less volatile, and a better lubricant for the injection pump, as well as a better starter. Very large engines designed to use tar oils are usually provided with pilot injectors, using from 5 per cent. to 10 per cent. of a more readily ignited petroleum. The delay in starting combustion with certain fuels is followed by a sudden rise in pressure, due to the fact that a considerable proportion of the charge is then in the cylinder whereas if burning starts promptly, the rise is more gradual and the combustion noise is less.

Fuels for compression ignition are sometimes classified according to their 'cetene value' which may be regarded as the direct opposite to the 'Octane number,' but universal standardisation has not yet been attained. Another system aims at defining the ignition delay in crankshaft degrees, and another in compression ratios.

Engines using gases or light volatile fuels are spark ignited, and the compression ratio is normally between 3·6 to 1, and 7 to 1—the figure varying with the fuel. Among the gases used are furnace gases and producer gas, both of low calorific value and much diluted with nitrogen.

Producer gas engines at one time had a great vogue. Plants have been built to make gas from a variety of waste substances, such as wood, sawdust, bark, cocoanut shells, sugar cane refuse, mealie cobs, etc., as well as charcoal, coke, bituminous coal and anthracite, but in spite of the economy of the engines, they have made little progress in this country.

An analysis of a sample of producer gas is given below :—

Out of 100 volumes				
CO_2	4·2 vols.	requiring	0	vols. of oxygen.
C_nH_{2n} (various)	0·6 ,,	,,	2·55	,, ,,
O_2	0·9* ,,	,,	0	,, ,,
CO	20·1 ,,	,,	10·05	,, ,,
H_2	6·8 ,,	,,	3·4	,, ,,
CH_4	3·7 ,,	,,	7·4	,, ,,
N	63·7 ,,	,,	0	,, ,,
Total	100·0 ,,		23·4	
			−0·9*	
			22·5	,, ,,
			=107·0	,, air.

Blast furnace gas has the approximate composition by volume as under :—

CO_2	11 per cent.
CO	27 ,,
H_2	2 ,,
N_2	60 ,,

Heat values for some of the gases used in engines and the air required for complete chemical balance are :—

Gas.	B.Th.lbs. per cub. ft.	Air for 1 volume
Blast furnace gas	84	0·69
Producer (coke) gas	126	1·07
Coal gas	500	4·90
Water (blue) gas	270	2·22
Water (carburetted) gas	458	4·27
Natural oil gas	1,000 to 1,100	11·00 (approx.)

Town gas—once the favourite for small plants—is now being ousted by electric motors or small oil engines, while in large plants this fuel is too dear.

In special cases, for example municipal electric plants operated in conjunction with a gas works, it may pay to use gas for engines driving generators for the evening lighting load, and so level up the demand on the gas making plant, which has its peak for cooking, etc., in the daytime, but the greatest call nowadays in the gas engine trade is for vertical multicylinder machines running on natural gas in the oil-fields, or furnace gas engines which are in general very large, and have horizontal cylinders.

Volatile Liquid Fuels.—The heaviest of this class is paraffin or kerosene, which while not volatile at ordinary temperatures will work in a carburettor engine if an exhaust heated vapouriser is provided.

Its greatest use was in small boats, particularly fishing boats, where the reduction in fire risk was in its favour. Owing to the fact that this fuel detonates readily, low compression ratios must be used. The thermal efficiency is therefore low, and as the preheating in the vapouriser reduces the volumetric efficiency, the power of the engine is also low in relation to its size and weight. Better kerosenes with higher knock values have been produced, but the vapouriser engine is being ousted in favour of the oil engine which is much more efficient, works on a cheaper and still safer fuel, and gives higher mean effective pressures. The really volatile fuels—petrol or gasoline, benzole and alcohol or mixtures of these—have been subjected to intense study and development, so that their qualities as represented by the compression ratio they will stand, freedom from undesirable impurities, ease of starting from cold, etc., have advanced enormously. Cracking and recombination of the simple petroleum distillates is practised extensively, while special fuels capable of working at very high compression ratios and 'boosts' are produced by hydrogeneration, *i.e.* the combination with hydrogen under very high pressures, and the use of small quantities of materials which have been found to inhibit detonation—notably, lead tetra ethyl, $Pb(C_2H_5)_4$.

These fuels enable suitably designed engines to operate at the maximum power/weight ratio yet attained, the remarkable figure of 11 ozs. per horse-power including blower and reduction gear being the dry weight of one racing aero engine produced for a special event. The fuels used in compression ignition engines of the light high-speed class, such as bus and truck engines, are light petroleum distillates of the grade generally known as 'gas oils' with a specific gravity of about 0·86 to 0·87. On account of the very small nozzles sometimes used (holes as fine as 0·008 in.) no sediment can be tolerated, and the fuel must be of consistently low viscosity and free from water and corrosive constituents. Fine filters are commonly fitted.

In the large air-blast injection engines, much cruder fuels of specific gravity 0·89 and higher can be used, but in cold conditions, means may have to be provided for heating the service pipes, as these fuels are very viscous at low temperatures.

The quality and cleanliness of fuel oils available for ships vary in different ports, and cases are reported of erosion of nozzles, etc., by fuels containing water and sediment. This is particularly bad for airless injection engines, which means almost all marine engines, and hence there is a move towards establishing some standards for these oils.

On the smaller plants distillates are generally regarded as practically essential, but experiments now in progress show that centrifugally filtered residuals work well in certain cell type engines of small size, though heating has been necessary to reduce the viscosity.

(10) *Injection of Fuel.*

The early oil engines used the method, still adopted in some large machines using residual oils, of injecting the fuel by an air blast of considerably higher pressure than that of compression. (Compression pressure about 500, blast pressure 800 to 900 lbs. per sq. in.) The fuel is metered into the injector by a controlled plunger pump and the needle valve lifted mechanically at the correct time. The high pressure air (from a two- or three-stage compressor built into the engine) then blows the fuel into the cylinder through the 'flame plate' breaking it up into a fine mist which fires from the heat of compression.

'Solid Injection' engines dispense with the air blast and rely on the very high pressure from pecial fuel pumps (from 3,000 up to 7,000 lbs. per sq. in.) and the fine nozzles to give sufficient

subdivision of the fuel. This system has gained ground very rapidly, as its simplicity makes it the only one for small engines and it has also been applied to the largest sizes. On marine engines it is now universal, and the fuel economy is in some cases as much as 10 per cent. better than blast injection engines driving their own compressors.

(11) *Design of Combustion Chamber.*

In both air blast and direct injection engines there are several ways of ensuring that the fuel shall be intimately mixed with the air.

(*a*) The open chamber (used also with blast air injection) relies on the several jets in the nozzle or flame plate to distribute the fuel, but in modifications of this system the air inlet ports or valves are arranged to give some rotation to the air so that the radial jets from the injector may be swept circumferentially by clean air and so use a larger proportion of the oxygen present.

(*b*) This rotation may be speeded up as much as possible, everything being done to create a rapid 'swirl' and thus maintain a good performance at higher speeds. Both (*a*) and (*b*) usually have the combustion chamber formed by a central cup in the piston head.

(*c*) The air may be compressed into a separate chamber in such a manner as to produce a rapid circulation past the injector. In these cases the nozzle may be of the single hole or preferably the 'pintle' type, which will run for long periods without attention.

(*d*) The piston may be made with a projecting piece on the head, known as a 'turbulence block.' Near the end of the compression stroke this enters and restricts the passage to the combustion chamber, thus setting up high velocity air currents which assist in the distribution of the fuel.

In general the best starting and the lowest consumption are obtained on open chamber engines with low velocity air movement, as the heat losses are lower, but the highest speeds and outputs have been obtained from certain engines of (*c*) type, which also give clean exhausts free from visible smoke and objectionable aldelydes up to high mean pressures.

(12) *High and Low Speed Types.*

In all classes of engines, speeds tend to increase from year to year.

In aero engines the struggle for greater power combined with propeller efficiency, forced a decision in favour of geared engines, even for large radials, and this at once opened the way for much higher rotational as well as piston speeds.

The 1914–18 piston speeds of 1,400 to 1,800 ft. per minute have now been raised to 3,000.

Car engines were originally rated on 1,000 ft. per minute, but present-day speeds reach 3,000 for cars and 2,500 for commercial vehicles.

In these fields there are degrees of speed but no broad division, whereas in marine work there is a definite step from the direct drive where speeds exceeding 120 r.p.m. are not usual, to the geared or electrically driven propeller, where the engines may run 250 r.p.m., in cargo vessels up to 450 r.p.m., as in the German 'pocket battleship' class, which have double acting two-strokes 420 mm. bore × 580 mm. stroke. The piston speed is approximately 1,700 ft. per minute, which is about twice that of the long stroke piston rod type, common in direct drives. In spite of the fact that the blowing engines develop 13 per cent. of the power of the main engines in providing the 5 lbs. scavenge air pressure required, the overall fuel consumption per b.h.p. at the couplings is approximately 0·385 lb., *i.e.*, 0·34 net apart from the blowers.

These engines are of wrought steel welded construction and the bare weight without gear boxes is the somewhat remarkable one of 18 lbs per b.h.p. There are now many ships in commission or building abroad in which single acting two-stroke geared engines with trunk pistons are fitted.

The elimination of the piston rod and the shorter stroke used, make these engines much lower, lighter, and easier to handle. The gear allows the designer to choose the highest engine speed consistent with life and economy, while keeping the propeller speed even lower than in the direct drive, and the gain in propeller efficiency goes to balance the small gear loss. The fast running trunk piston engine uses rather more lubricating oil, and the slow-speed crosshead type has an excellent record behind it, but judging by all the other fields the increase in speed in marine engines will go on. A number of ships' auxiliaries have been installed, running at 900 r.p.m., and 1,800 ft. per min. piston speed. These have given consumptions as low as 0·365 lbs. per b.h.p. hour on long runs, and used in generator drives they save a great deal in weight and cost of the dynamos.

As pointed out in the notes on cooling, cylinder wear is a matter of corrosion as well as erosion, so that increasing the speed does not increase the wear in proportion. In other words, a piston running fairly fast does more work before reboring is necessary than if run slowly under the same temperature conditions. The increase in speed has led more and more to the use of aluminium alloys, both cast and forged, and recently magnesium alloys have also been tried experimentally by several makers. In marine engines wrought and cast steel are taking the place of cast iron

as they are not only stronger and therefore lighter for the same loads, but have a stress strain relationship (Young's modulus) twice as favourable, in other words, stiffness does not suffer by the reduction in section.

Dynamic forces which increase as the square of the rotary speed have now become more important, and every effort must be made to use the advantages offered by the stronger or lighter materials now available to reduce the weight of moving parts.

Constructional Details Peculiar to Large Engines.

While many things are common to all engines, the larger classes have problems of their own, and in particular use constructions not readily applicable to small machines.

General Construction.

Practically all marine and most other large oil engines now have vertical cylinders. Crankshaft bearings are usually carried in cross members of the bedplate to which is attached the column supporting the cylinders. In some cases the cylinder heads form a stiff entablature attached to the bedplate by the transverse bulkheads of the column, and from this entablature the cylinders depend, the water jackets being of light section and unstressed. The lower ends of the cylinders spigot into a deck which takes the side thrust in the case of trunk pistons. This scheme has the great merit of giving a cylinder free from axial stresses, and thus reduces the risk of distortion in the top flange and in the neighbourhood of the piston rings at top centre when gas pressure is greatest.

In another system the water jackets are of strong section and either cast in one piece or have their end faces machined and firmly bolted together to form a stiff upper beam. The main bolts pass up to the top of this beam and may extend farther through the cylinder heads, or the latter may be separately studded to the top face of the jacket section.

The liners in this case spigot top and bottom into the jacket, the bottom joint being made with rubber rings while the top is a slight interference fit.

The cylinder liners are turned all over to ensure uniformity, and are easy to renew if necessary.

In some very large engines forged mild steel has been used for the cylinders, with a thin dry liner of hard cast iron shrunk in. This reduces the thickness required, and in consequence the thermal stress is less in proportion to the ultimate strength of the material. In this connection it must be remembered that in two cylinders of the same thickness—one of cast iron and the other of steel—having the same temperature difference between the inner and outer skins, the stress in the steel is greater because the relationship between stress and strain is greater, *i.e.* 30×10^6 in steel and about 14×10^6 for cast iron (in lbs./sq. in. units), while the thermal expansion is the same approximately. The steel, however, will stand a tensile stress even higher in proportion, thus giving it a considerable advantage. The extreme upper part of the cylinder is sometimes formed with an external helical rib, which, when the jacket is in place, forms a narrow passage through which the circulating water is forced at a high velocity to take away the heat from this part, where the temperature and density of the gases are much higher than lower down, and where in consequence the heat stresses are greatest, particularly as the liner is thickest here. The lower part of the cylinder should be kept as hot as possible, having regard to the cooling system used (fresh or salt water). The cylinder thickness should be calculated for the maximum gas pressure at the top and may be reduced considerably below when the liner is not subject to axial load. For good cast iron the metal may be $0 \cdot 07\ D + 0 \cdot 25$ in. at the top, tapering to $0 \cdot 04\ D + 0 \cdot 25$ in. at the half-stroke and continuing parallel. If steel barrels are used, the stress due to gas pressure may be at least doubled, but it is sometimes advisable to turn ribs on the outside to increase the ring stiffness and preserve the roundness of the working barrel. Due to the temperature difference of the inner and outer faces, a very thick cylinder may be more heavily stressed by heat than a thinner one by mechanical forces, and a stronger material is then the solution.

The maximum gas pressure in the old air blast engines was about 500 lbs./sq. ins., but with airless injection the peak is now about 700 in large cylinders. This only affects the upper part of the cylinder, and the pressure drops very rapidly as the piston moves down the stroke.

Thermal and Mechanical Stresses in Liners, etc.

The percentage heat losses to water do not vary very much with piston speed or size, for the same general design of engine, but different designs and systems differ widely in the amount and distribution of the heat flow. This suggests that the horse-power per square inch of piston may be taken as a rough measure of the rate of heat flow in corresponding parts of engines of the same class.

Local gas velocities together with the temperature and density obtaining at each instant determine the heat flow at any point. The cycle occurs so rapidly, however, that it is unlikely that anything beyond the merest skin is affected by the cyclic temperature change, and the mean value is the pertinent figure, even in comparatively slow engines.

For any given heat flow the temperature difference between the inside and outside surfaces of a containing wall varies directly as the thickness, and inversely as the conductivity. If the wall is constrained to keep its shape, the outer part is intension, and the inner in compression. The thermal stress at the face is therefore—

$$\pm \frac{R \times t \times E \times a}{2C} \text{ lbs. per sq. in.}$$

where R = rate of heat flow in B.Th.U.'s per hour per sq. in.
t = thickness in ins.
E = direct modulus of elasticity.
a = coefficient of thermal expansion per 1° F.
C = conductivity B.Th.U.'s per hour per in. cube per 1° F.

Approximate values for the various factors are as under:—

Cylinder heads high output cell type oil engines .	$R = 700$	B.Th.U.'s per sq. in. per hour
Cylinder heads high output supercharged petrol engines	$R = 700$	,, ,, ,,
Liners for either of the above—top zone . .	$R = 300$ max.	,, ,, ,,
Liners for oil engines, 2-stroke . . .	$R = 150$	for 1,000 ft. min. piston speed.
,, ,, ,, ,, 4-stroke . . .	$R = 125$	

Material.	E.	a.	C.
Cast iron	14×10^6	$6{\cdot}7 \times 10^{-6}$	3·0
Aluminium (normal high expansion alloys, cast or forged)	11×10^6	$12{\cdot}5 \times 10^{-6}$	9·6
Mild steel	30×10^6	$6{\cdot}2 \times 10^{-6}$	2·15

It is clear that the thermal stress increases in proportion to the thickness, while the stress due to gas pressure varies inversely as the thickness, being $\frac{DP}{2t}$ lbs. per sq. in. (D = cylinder diameter, ins., P = pressure lbs. per sq. in., t = thickness inches). Therefore there is in each case an optimum thickness for liners, etc., which gives the lowest combined stress.

Example:—

20-in. cylinder, 4-stroke, cast-iron liner, 1,200 ft./min. piston speed, 700 lbs. maximum pressure, say 600 maximum on liner below the spigot.

Heat flow 150 B.Th.U.'s per hour per sq. in. approximately.

Thickness (inches).	Stress.		
	Thermal.	Mechanical.	Combined.
1	2,350	6,000	8,350
1¼	2,940	4,800	7,740
1½	3,520	4,000	7,520
1¾	4,110	3,430	7,540
2	4,700	3,000	7,700

The addition of the two stresses shows that between 1½ in. and 1¾ in. there is little difference, and allowance for reboring, etc., would suggest that 1¾ in. would be suitable in this case for the upper part of the liner.

Through bolts are often used to tie the heads down to the bedplate. They pass through distance pillars, preferably formed in the bulkheads or A frames of the column, and these pillars have an important bearing on the fatigue stress of the bolts. Since both bolt and pillar are springs in parallel, the variation in bolt stress when each explosion occurs depends on the pillar as well as on the bolt.

Let A_1 A_2 = Areas in sq. ins. of bolt and pillar respectively.
,, L_1 L_2 = Length in ins. ,, ,, ,,
,, E_1 E_2 = Young's Modulus for the materials in bolt and pillar (30×10^6 for steel, 14×10^6 for cast iron).

Then the axial stiffness of the bolt is $\frac{A_1 \times E_1}{L_1} = S_1$ lbs. per inch., and the axial stiffness of the pillar is $\frac{A_2 \times E_2}{L_2} = S_2$ lbs. per inch. Therefore the stiffness of the two together is $S_1 + S_2$ lbs. per inch deflection. And if the variation of the *external force* (in this case the gas force) coming on to each bolt and pillar is F, then the deflection is $\frac{F}{S_1 + S_2}$ inches, the variation in bolt tension is only $\frac{F \times S_1}{S_1 + S_2}$ and the variation in bolt stress is $\frac{F \times S_1}{A_1(S_1 + S_2)}$. This is a very important point and should be applied to connecting rod big end bolt bosses, etc. It shows that the stiffness of the metal round tension bolts carrying varying loads is as important as the bolts themselves, and that the failure of a bolt from fatigue may be due to the flimsiness of its abutment.

The superiority of cast steel over cast iron on account of its higher E value is also clear.

Wrought Steel Construction.

The use of this material for engine columns and bedplates is increasing rapidly. Long through bolts may be dispensed with so that there are no opposed stresses, pattern-making is cut out, and size becomes of less importance as the foundry question never arises. Greater rigidity or less weight, absence of 'wasters' cooling stresses and blow-holes are further advantages. Cast steel bearing housings or ribbed cross members of small individual size and of open section easily cast without cores, may be employed and attached by tennons to the plates forming the bulk of the structure. Electric welding is used to hold all the parts together and fill and key the tennons, but the design is so arranged that direct tension on the welds is not relied upon. Double bulkheads may be used, curved round at the top to bring the stress lines close to the cylinder flange, or separated at the bottom to enclose the bearing housings. Strips are welded on where required for machining. The main bulkheads may be made as U-shaped slings, carrying the bearing housing in the bend, and attaching to the cylinder beam at the two upper ends, thus eliminating the long bolts. Side plates notched and welded in position extend right round under the frames, and project right and left to the cylinder centre line, where they are bolted together on a vertical joint and complete the crankcase. This construction has resulted in a really astonishing reduction in weight. The whole structure should be annealed after welding.

The cost compares very well with the heavier designs, and in ship work it means less dead weight to carry.

In addition to the direct gas forces in line with each crank, the whole engine must be considered as a beam subjected to bending moments set up by the varying dynamic forces along the length of the engine.

These couples depend on the amount of balancing, the crank arrangement, the reciprocating masses, etc., and are given in the crankshaft section, but the engine frame must withstand them without undue stress or deflection, and therefore it is necessary to make an approximation to its strength as a beam. The welded structure lends itself very well to this need for beam strength, as good longitudinal sections can be provided at top and bottom, while quite light side plates, suitably attached, form what is in effect the web of the girder.

Crosshead.—This feature is only found in large engines. The area of the thrust face may be equal to that of the piston or down to 0·75 of that figure. In single guide designs the sliding shoe is T section, and retained by keep plates each side which take the horizontal component of the compression force. This is much less than the working thrust and does not need half the area of the main slide face. Means should be provided in the design for adjusting the face of the fixed guide for both distance and alignment. If distance plates are used for the adjustment, they should be of ample area to prevent any subsequent settling down, and should be screwed or riveted in place. Cast iron forms the best surface for the fixed guide. The shoe may be of cast iron or steel, the latter faced with white metal.

There seems no reason why aluminium, which works so well in aero and car engines (where the piston forms its own crosshead) should not be used for the shoe, and in one experimental engine this has been done with a useful saving in reciprocating weight.

Piston Cooling.—Liquid cooling of pistons becomes necessary as the size increases, because the heat gradient in the piston head becomes too great and the rings will not live. Two-stroke engine pistons are hotter, size for size, than four-strokes.

Salt water is still used in a few cases, but fresh is better, and oil is much to be preferred to either, for two reasons :—

(1) leakage into the crankcase from the telescopic or hinged pipe connections does not matter provided enough oil is reaching the piston.

(2) Corrosion fatigue may be set up by the action of the water, particularly at the edges of holes (in the piston rod for example) through which it passes.

Piston Rods.—Prolonged tests on samples unstressed on the one hand, and subjected to alternating loads on the other, show that the rate of corrosion in weak salt, acid or alkaline

solutions is much more rapid in the case of stressed parts, and the piston rods, especially of double-acting engines, should be made with as little stress concentration as possible, radial holes should be avoided, and oil cooling employed to eliminate this danger.

Internal bushings of cast iron are sometimes fitted to reduce the chances of corrosion fatigue in the rod when water cooling is used. External sleeves of pearlitic cast iron are also in common use in double acting engines to reduce the temperature stress in the rod, and an arrangement used in one very large engine is shown in fig. 1 where the piston cooling oil passes up through the annular space between the rod and the protecting sleeve, thus keeping the former free from any marked temperature differences, while the thermal stress in the sleeve is small owing to its thin section. The outlet is down through the centre of the hollow rod. The sleeve is attached to the top and free to expand below.

The minimum section where the rod is screwed into the crosshead should not be less than 0·04 of the piston area in a double acting engine, and every possible care must be taken to avoid sudden changes of section, sharp corners, etc., which would cause concentration of stress.

Oil Pipes.—Most slow-speed engines employ sliding telescopic pipes for flow and return of the piston coolant, but at high rotary speeds the pulsations and pumping effects become marked, even when large air vessels are fitted. 'Walking pipes' then become preferable, if not absolutely necessary.

Liberal surfaces should be allowed at the hinged joints, and light but stiff *straight* weldless steel tubes used with end fittings welded or brazed on. Any tube having a considerable amplitude of movement should be calculated for strength in bending, and curves should be avoided in these members.

W lbs. = weight of tube per inch (including the oil in it); S = stroke in inches at end.

$R = \dfrac{\text{crank}}{\text{con. rod}}$ ratio.

R.P.M. = revs. per minute.

Then the force per inch varies from zero at the fixed end to—

$$+ 14{\cdot}1\ SW \left(\frac{\text{R.P.M.}}{1{,}000}\right)^2 \times (1 + R) \text{ at Top Dead Centre} = + F_T$$

and

$$- 14{\cdot}1\ SW \left(\frac{\text{R.P.M.}}{1{,}000}\right)^2 \times (1 - R) \text{ at Bottom Dead Centre} = - F_B$$

If L inches equals the length of the pipe the greatest bending moments are—

$$\frac{+F_T L^2}{9\sqrt{3}} = + M_T \text{ at top, and } \frac{-F_B L^2}{9\sqrt{3}} = - M_B \text{ at bottom centre.}$$

The resistance of the tube to bending is $\left(\dfrac{D^4 - d^4}{D}\right) \times \dfrac{\pi}{32} = B$ ins.^{3} (when D and d are diameter and bore of pipe) and the stresses positive and negative are $\dfrac{+M_T}{B}$ and $\dfrac{-M_B}{B}$ lbs. per sq. in. respectively. They should not total more than 18,000 lbs./sq. ins. To lower the stress, increase the diameter D.

Pistons.—These are now often made in the larger sizes with forged steel heads machined all over without ribs or other irregularities, in order to avoid distortion and cracking. Cast steel may also be used, and many are still of cast iron, but the strength of the latter being much inferior it must be thicker for the same load, and hence the thermal stresses are higher.

The older designs with ribs under the head and made of ordinary cast iron, develop cracks in the higher duty airless injection engines now being built. Ribs should be avoided in the head.

The cooling liquid should be constrained to follow the heated surface closely and at a good velocity. Skirts are of cast iron and piston bearings of bronze on case hardened pins.

A special double-acting piston design is shown in which the comparatively thin crowns (upper and lower) are supported mechanically near the edge and at an intermediate diameter by a strong forged steel centre. The piston rod has no radial holes in highly stressed sections. Light baffles of aluminium are used to cause the cooling oil to pass right across the heated surfaces and behind the rings, which are fitted with wear rings. The crowns are of forged Y alloy (aluminium), while the split bolted-on guide ring is of cast iron and registered on the steel centre.

Fairly large four-stroke trunk pistons up to about 22 ins. diameter have been made of cast aluminium with bronze interiors to carry the pin. These are not liquid cooled, but rely on a deep nest of narrow rings. The speed is nearly 1,600 ft./min.

Starting.—The usual method is to admit compressed air from a distributing valve through non-return valves in the heads of a sufficient number of cylinders, but small compressed air engines running fast and geared to a toothed ring on the flywheel are more efficient, and are common on medium sizes.

Engines driving generators may be arranged to be started by using the dynamo as a motor. Dynamometer tests give the torque required to start a high compression oil engine as 40 to 45 lbs. per sq. in. of piston referred to the whole engine. Lifting the valves to release compression reduces the torque at the first movement, but they must be dropped quickly or work is wasted in pumping losses.

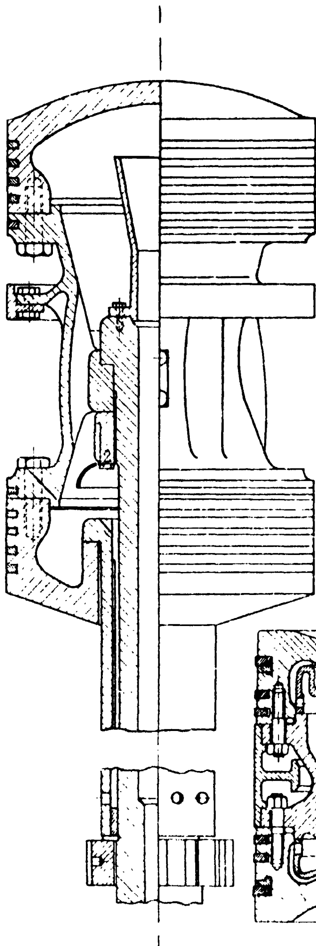

FIG. 1.

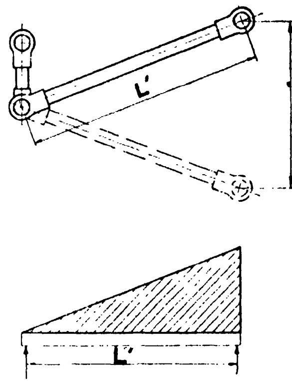

FIG. 2.

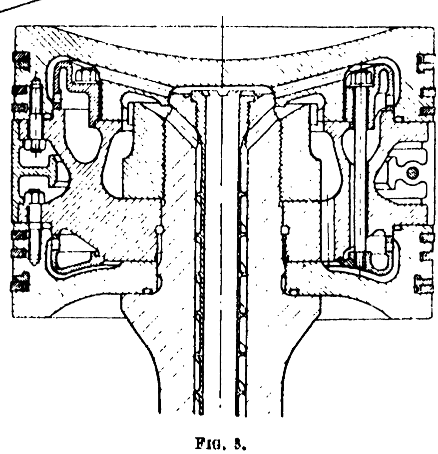

FIG. 3.

Connecting Rods

The bottom ends are almost always of the 'marine' type, *i.e.* two separate halves, of cast or forged steel, white-metalled on the working face, and bolted to the tee end of the shank.

The diameter is dictated by the shaft requirements, but with the higher pressures now ruling, the projected area should be as generous as possible. In large line engines the diameter may be as high as 0·83 of the cylinder bore, with a width of half the diameter, giving a net projected area, exclusive of fillets of about 0·4 of the piston area. This may be reduced to 0·33 of the piston area if necessary. The top end may also be in halves bolted on, but is sometimes integral up to the gudgeon pin centre line, with the top cap or caps (in the case of forked rods) bolted to ears on the shank.

Connecting Rod Bolts.

In single acting two-strokes, these have little or nothing to do, since the pressure is almost always downwards but in double acting engines they must be made to take the full gas load as a recurring stress. The area of the bosses round the bolt should be adequate, so as to limit the stress variation (see note and equations at the beginning of this section, on through bolts and pillars in frame construction).

When running at speed, the gas load is reduced by inertia forces, but at starting and accelerating from low speeds nearly the whole of the gas pressure is realised. In single acting four-stroke engines the periodic pull on the bolts is that due to the inertia of the reciprocating parts at T.D.C. plus the centrifugal load of the upper half of the big end and half the shank.

If r = crank radius, L = connecting rod length, R.P.M. = revs. per min., W = weight in pounds of the piston with all reciprocating parts including the small end and half the connecting rod shank; W_1 = the upper half of the big end plus half the connecting rod shank; then the periodic upward pull on the big end bolts at T.D.C. is :—

$$[W_1 + \left(1 + \frac{r}{L}\right)W] \times 28\cdot2 \times r \times \left(\frac{RPM}{1{,}000}\right)^2 \text{ pounds}$$

and they must take this as a fatigue load.

As much stretching length as possible should be provided, turned well below the bottom of the thread (0·85 the area of the bottom of the threads is a good figure) and the changes of section should be well rounded and absolutely clear of tool marks.

A high tensile steel may be used for the bolts, but a much softer material is better for the nuts, and reduces the stress concentration on the first two threads where fractures usually occur.

Connecting rod bolts must be tightened up initially so that the surfaces of the half caps never separate. A good plan is to make the bolts to a limit gauge length over-all and tighten up until they stretch a specified amount, which is calculated to give the required load, with a margin for errors and settling down of threads and surfaces—say 30 to 40 per cent.

Crankshafts

On account of the size of the forgings required, the shafts of large engines are made in several parts, and a usual practice is to make a pair of webs and one pin in a piece and shrink them on to the journals.

The pins are also sometimes shrunk into separate webs, so that the whole shaft is built up of simple turned sections and slabs bored and faced.

The first method, however, makes a lighter shaft, and what is often much more important, one with smaller unbalanced masses to produce centrifugal loadings on the bearing, and of lower polar inertia.

The thickness of the shrunk band round the shaft may be of the order of 0·35 of the shaft diameter. In better steels than the mild quality usually employed, the yield point is higher and a greater initial interference may be used. The band thickness in one design which has been running for several years was reduced below 0·3 d (d = shaft diameter) without any subsequent trouble.

The interference suggested for mild steel (30 tons ultimate and 18 tons yield) is 0·0015 per 1 in. of shaft diameter, and higher according to the yield of the material. Since the shaft squeezes in a little, the nominal stretch per inch at the bore may be about 10 per cent. more than that given by the yield.

E.g. 18 tons/sq. in. + 10 per cent. = 44,500 lbs./sq. ins.

$$\text{Stretch per inch for this stress} = \frac{44{,}500}{30 \times 10^6} = 0\cdot00148 \text{ in., say } 0\cdot0015 \text{ in.}$$

The grip may be approximated as under for the torque required to *start* the web moving on the journal. Once it has moved from the original position, it goes more easily.

Area of ring section on one side (sq. ins.) × total interference in thous. × 8,000 = lbs. ins. torque.

A generous factor should be allowed for possible torsional oscillations and the peak torque from each cylinder.

A shrink grip is not apparently affected by alternating loads which are below its slipping torque.

If, therefore, the grip torque is such that the corresponding shaft stress would be above the permissible figure for fatigue, the shrink should be safe.

A torque giving between 9 and 10 tons per sq. in. at the journal is suggested.

Dowels were at one time fitted after shrinking, but they cut the most heavily stressed ring of metal, and cause an objectionable concentration of stress above the hole, as well as loosening the grip. They are better left out. Rough surfaces are apparently no advantage. Close limits on journal and hole diameter must be adhered to and careful temperature control is essential.

If a gas furnace is used, a reducing flame helps to prevent oxidation of the surfaces. When heating the web, allowance should be made for sufficient expansion to give clearance for sliding on easily, and preliminary arrangements made to ensure rapid and easy handling, so that no time is lost. With care, 250 to 300° C. is adequate for 1·5 thous. per inch interference, and between 300° C. and 350° C. for 1·75 thous. per inch. Scaling will not occur at these temperatures.

For large shafts carbon steels up to about 37 tons ultimate are preferable to alloy steels, on account of—

(a) Cost.
(b) Ease of production.
(c) Consistent qualities throughout heavy sections.
(d) Ease of heat treatment.
(e) Ductility, which ensures less danger from stress concentrations.
(f) High hysteresis damping.

Size Limit.

The most powerful engine yet built is an 8-cylinder in line double-acting two-stroke 840 mm. × 1,500 mm. giving 22,500 b.h.p. at 115 r.p.m., but larger cylinders are found in blast furnace gas engines, one of which—a horizontal double-acting tandem—had cylinders 51-in. bore.

The limit of power has not been reached by any means. Higher speeds and the increase of the number of cylinders as well as the raising of the overload by boosting will all contribute, and the builders of the engine referred to above, have declared themselves ready to build a unit of 45,000 b.h.p. in 12-cylinders running at 180 r.p.m. It is questionable, however, whether this as a land engine could compete with the modern steam turbine if coal is readily available.

Notes particularly applicable to High-Speed Engines.

The term 'high speed' is somewhat vague and requires qualification.

High rotary figures may be attained by very small engines with stresses no higher than those in large engines running at less spectacular revolutions per minute.

The mean linear speed of the piston is the best figure for comparison, as it can be shown that for similar designs of widely different sizes the same piston speed means the same stress in corresponding parts of the engines.

Theory of Proportionality.—According to this theory, engines of all sizes built to the same design and proportions in all details have the same stress in all corresponding parts when subject to the same gas pressure and run at the same piston speed.

Consider two engines of stroke S and s respectively, everything being in proportion. Then the piston areas will vary as S^2 and s^2. The sections of the connecting rods, the bearing areas, etc., will all vary in the same proportion, *viz.*, S^2 and s^2, so for equal gas pressures the stresses and bearing loadings due to this cause will be equal.

At the same piston speed the rotary speeds of the two engines will be inversely as the linear dimension, *i.e.* 's' r.p.m. for the larger and 'S' r.p.m. for the smaller.

The dynamic forces will vary in proportion to :—

(1) The weight of the parts, *i.e.* S^3 and s^3.
(2) The stroke S and s,
(3) The rotary speed squared s^2 and S^2

the result being $\dfrac{S^3 \times S \times s^2}{s^3 \times s \times S^2} = \dfrac{S^2}{s^2}$

but the cross-sectional areas of all the parts, such as connecting-rod shank, big-end bolts, etc., also vary as $\dfrac{S^2}{s^2}$.

Therefore, the dynamic forces also produce equal stresses at the same piston speed. Similarly, it can be shown that the torsional frequency of the crankshaft, the surging speed of the valve springs, etc., all move in step and are inversely proportional to the linear dimensions S and s. Thermal stresses, however, increase with size and must be considered separately.

It is assumed then that 'high speed' engines are those designed to run at a high piston speed, say 2,000 ft. per minute and upwards.

The highest speed yet attained, as far as the writer knows, was in a racing car engine of 132 mm. stroke, which ran at 5,000 r.p.m., normally aspirated, and reached 6,000 r.p.m., when supercharged—a mean speed of 5,200 ft. per minute on the pistons.

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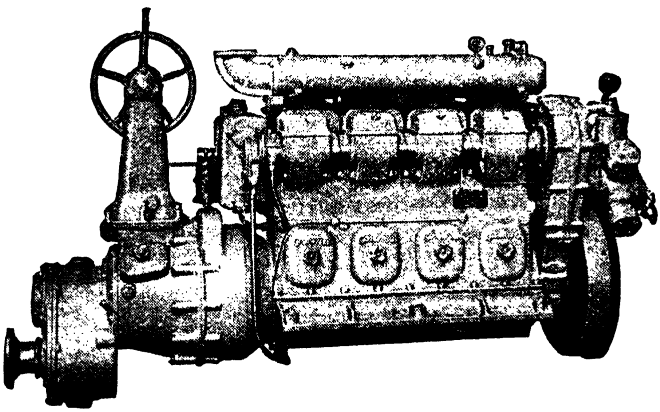

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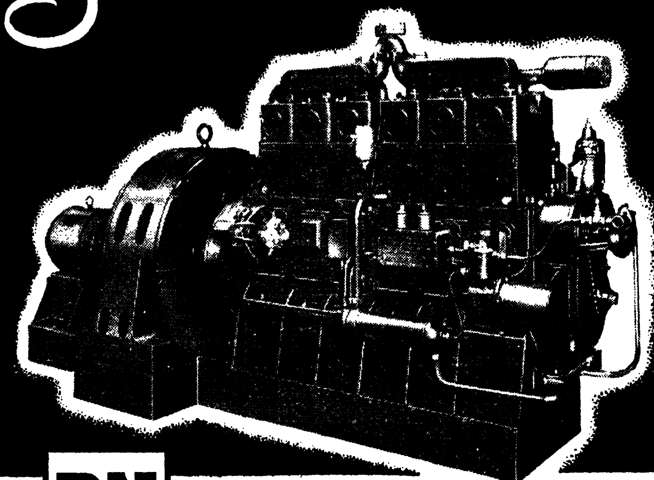

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Racing cycle pistons exceed 4,000 ft. per minute at times, but in car engines 3,000 to 3,300 is the usual limit.

Aero engines have run at 3,600 ft. per minute, but 3,000 is the normal figure for both liquid and air-cooled machines.

It must be remembered, however, that the load is much higher and more continuous and that most aero engines are supercharged.

The problems introduced by high piston speeds are :—

(a) Thermal.

(b) Dynamic.

The piston is subjected to higher heat flow, and in consequence all aero and cycle engines and nearly all car engines have aluminium pistons, because its conductivity is so much higher—being about 2½ times that of cast iron. Greater clearances must be allowed, as aluminium expands more than the iron cylinder in which it works. The expansion ranges from 0·0018 of its length per 100° C. for certain alloys, up to 0·0023 or 0·0024 for others. Unfortunately, those of higher expansion are also stronger, and therefore usually used in high duty pistons.

Compare cast iron 0·0012 per 100° C. The top land is given a clearance of about 0·008 D + 0·010 in., and this is reduced progressively to the skirt where the allowance depends on the duty.

It is not possible to give a general rule for piston skirt clearance as this varies with the alloy used, the heat flow, the piston design and the cylinder cooling.

In general, the higher the horse-power per square inch of piston, the hotter it will be, and short pistons with few rings also attain higher temperatures on the skirt and require greater clearance than those with a more generous length above the gudgeon pin centre.

Skirts may be given a slight taper, say 0·0005 in. to 0·001 in. on the diameter per inch of length, and are frequently made oval, so that distortion of the piston pin bosses may not cause binding, while the thrust faces, in the line of the connecting rod swing, may be made a somewhat closer fit.

The increased dynamic forces due to the high rotary speed cause every effort to be made to keep the piston weight down, and for this purpose aluminium is without a real rival.

Mangesium alloys are being tried, but are not yet really in the field, since the mechanical properties at present are much poorer than those of the modern aluminium alloys, especially at high temperatures, and outweigh the gain in weight (specific gravity aluminium alloys about 2·7 and of magnesium alloys as used 1·8) while the mean thermal expansion is 0·00275 per 100° C. against aluminium 0·00235.

Die cast or chill cast pistons are in great demand for all except the highest class aero engines where all machined forgings of 'Y' or similar alloy are used. These are of course very expensive by comparison. 'Upset' forgings with piston pin bosses and internal head ribbing all complete are being made in America in special die forging machines. The external turning, ring grooves, and boss boring are the only machining operations required. The mechanical properties of the metal are improved by this method of production.

Grinding of the skirts is giving way to diamond turning as a finish, high speeds and fine feeds being used. A beautiful surface is produced which is very true.

As a guide to weight, one well tried modern 5 in. aero piston body weighs about 2·5 lbs. bare or 3·75 lbs. with pin and rings.

The head thickness is about 0·1 D (D = diameter).

Three rings are usually used, two gas and one slotted oil scraper ring with the groove freely vented by drilling at the back.

Oil engines usually have three gas rings and one scraper, all above the pin. Sometimes provision is made for a second scraper ring near the bottom of the skirt. This is fitted when the oil consumption begins to rise, owing to wear. Gas rings made taper (5° to 10°) on the top side, and with sufficient side clearance to prevent binding when bottoming in the groove, will remain 'alive' (i.e. not stuck with carbon) for longer than parallel ones.

Two representative pistons are shown in figs. 4 and 5. The pins vary from 0·25 D for petrol to 0·36 D for oil engines.

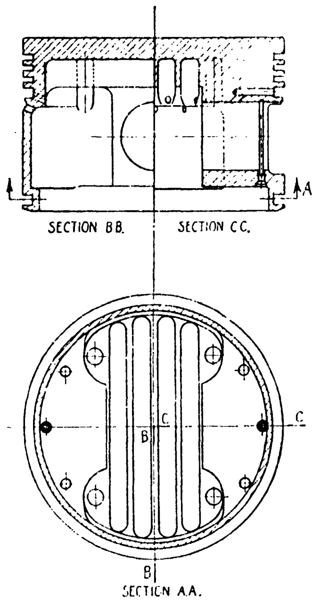
Where very high heat flow is anticipated, as in a supercharged oil engine, extra rings with sufficient metal behind them to conduct the heat, should be used, otherwise the pin bosses will become too soft at the bearing surface and will wear very rapidly. Attention should be paid to the support of the pin bosses, thin ribs overstressed at the inner edge are worse than useless. (See T section ribs in fig. 5.)

Proprietary pistons are being used more and more, and nearly all rings are now made by firms specialising in their production. Narrow rings are now general, the width being 0·025 to 0·03 D (D = piston diameter).

Other parts subjected to severe heat flow are the exhaust valve and its seat.

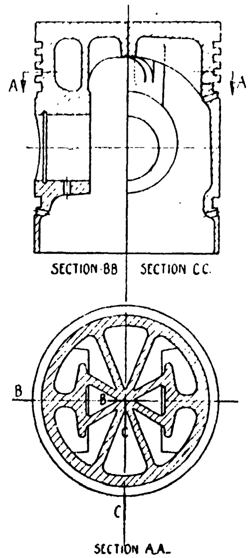
Various heat resisting steels, tungsten alloys and others are used, but for special cases of really high output the most recent development is the hollow valve, in which about half the internal

space is filled with metallic sodium. This melts at about 97·5° C., and being shaken rapidly up and down transmits the heat to the large diameter hollow stem whence it is taken away by the guide (the boiling point of sodium at atmospheric pressure is 825° C., so that it forms an excellent medium). Note that extra care must be taken to see that the guide boss has a good flow of air or water round it, and has ample cooling area to deal with the extra heat in this case. Welded on 'stellite' faces are also employed, usually on the valve and give excellent results, while inserted seats of special iron, corrosion resisting steel, aluminium bronze (in aluminium heads) or monel metal are now always used in aero work and nearly always for the exhaust valve seat of commercial vehicle oil engines, as the seat erosion is aggravated by the free oxygen present in the gases from this latter class of engine. In heads of iron or steel, the seats may be simply pressed in with an



HIGH DUTY AERO PISTON

FIG. 4.



PISTON FOR LIGHT LORRY OIL ENGINE

FIG. 5.

interference of 0·0025 per 1 in. of diameter. It is worth remembering that an increase in the stem diameter and lengthening of the guide have been found sufficient to cure some cases of over-heating in exhaust valves, as the extra area for heat flow kept the temperature below the critical point. The guide may also be continued downwards as a shroud, clear of the stem, to protect the working surface from the gases.

Connecting rods are sometimes stampings of aluminium alloy for car engines, but steel stampings are more general.

Aero rods are machined all over and of material too hard to stamp to finished section.

The cross-section under the small end, *i.e.* the minimum section, should be large enough to take the maximum gas pressure on the piston as a recurring fatigue load because, although the compressive load due to gas is reduced on the working stroke by the upward inertia pull, the latter appears as a tensile force on the next revolution, so that the fluctuation is between—

Downward force (gas − inertia) and upward force, inertia,

i.e. the range is equal to the gas force.

The cross-section will vary with the material from 1/60 to 1/25 of the piston area for high tensile alloy and medium carbon steels respectively. The shape of the shoulders running down to the bolt bosses should be designed to spread the load over the pin.

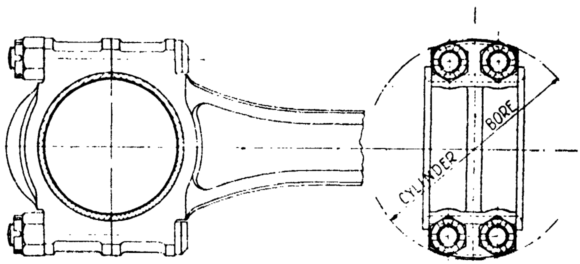
The cap should be stiffened by a deep arch-shaped rib to take the top inertia pull, and there should be a fair balance of area inside and outside the bolts at the joint face.

For connecting rod bolts see note, p. 209.

An upper limit for 60 ton steel bolts is 10 tons per sq. in. referred to the inertia pull at T.D.C. and maximum speed. This is for well-designed bolts without sudden changes of section and surrounded by symmetrical bosses of sufficient area. The rod length centre to centre may be very short in 6-cylinder engines, but in 4-cylinders it is not advisable to use less than four cranks long, as the secondary forces are unbalanced and rise rapidly, being proportionate to $w^2r\left(\frac{r}{l}\right)$; r = crank radius; = connecting rod length, centre to centre; w = angular velocity radians per second.

In many small engines it is necessary to remove the pistons from the top, and to do this the connecting rods must be (*a*) small enough in the big end to pass through the cylinder bore; or (*b*) long enough in the shank to allow the gudgeon pin to be slid out over the cylinder top face when the head is removed and before the big end bolt bosses foul the lower end of the cylinder.

A four-bolt big end turned on the longitudinal axis and using T head bolts, allows of the largest crankpin and may be used to satisfy (*a*).



TYPICAL HIGH SPEED BIG END

FIG. 6.

Fig. 6 illustrates this construction, which is useful when the crankpin must be the maximum possible size for torsional reasons. Steel is preferable to brass for the backing of the white metal bearing, and may be very thin. Better adhesion is obtained with low carbon mild steel. Thorough de-greasing is essential. It is most important that bearings should be a tight fit and take a good bearing in the big end bore, otherwise fretting will occur and local overheating with consequent cracking of the white metal. 2,500 lbs. per sq. in. is about the maximum *peak* loading for white metal, 2,000 lbs. per sq. in. is a safer figure.

Pressed strip steel bearings $\frac{1}{16}$ in. thick are now being produced with about 0·010 in. to 0·015 in. of white metal on the face. The technique of production is difficult, but the saving in diameter and weight is useful and the cost ultimately much less.

Centrifugally cast lead bronze on a steel backing is now used in many aero engines and oil engines for heavily loaded connecting rod big end bearings, but it is more expensive and needs a high carbon or alloy steel shaft, as it wears soft steel quickly.

Small end bearings use a hard bronze bush, which in hard tensile steel rods may be made to float in the bore of the small end. In this case a very small shallow groove should be turned outside the bush at the centre, and several oil holes drilled from it to connect with the inner surface.

A suitable bronze centrifugally cast is :—

Copper	88 per cent.
Tin	10 ,,
Zinc	2 ,,

Floating gudgeon pins are almost universal on high speed engines. They are always hardened in some way, the materials used for the carburising process being ordinary low carbon case-hardening steel, 3 per cent. nickel steel, or for aero engines nickel chrome.

Special steels, usually containing a little aluminium, are used when a nitrided surface is adopted.

Air hardening steel has also been used, but is not common.

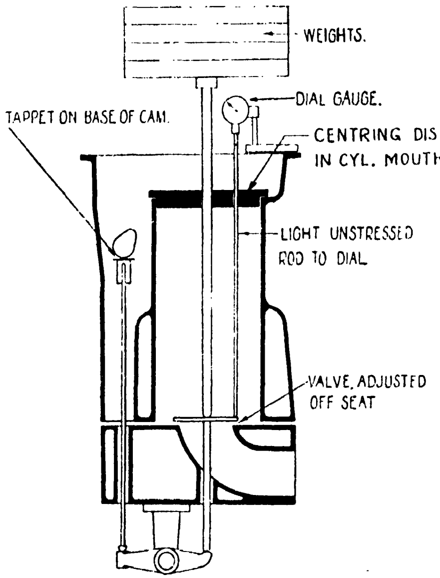
The advantage of the nitrided surface is that it is not affected at quite high temperatures which would begin to soften a carburised case.

Dynamic Troubles.

High speed engines have very low cyclic irregularity and phase variation (see next section) and arrangements of cylinders giving irregular firing angles may be employed if desirable on other grounds, but, on the other hand, they are much more prone to exhibit forced or synchronous vibrations, particularly in the following ways which must always be looked into carefully before the design is allowed to go through :—

(*a*) *Crankshaft*, torsional vibrations due to synchronism.

(*b*) *Frame*, bending vertically due to the combination of free centrifugal and inertia forces, and horizontally due to centrifugal forces only. Vee engines should have each bank treated separately and the resultants obtained. (See 'Crankshaft Balance,' p. 227, for forces involved.)



TEST RIG FOR VALVE GEAR DEFLECTION

FIG. 7.

(*c*) *Valve Gear*.—(1) Bouncing of valves due to faulty cam design or inadequate spring pressure in relation to speed' (see Cam Design). (2) Surging of springs (see 'Valve Springs,' p. 223) (3) Deflection of valve gear. Rockers are the worst offenders, then inadequately or improperly attached fulcrum brackets, then push rods. The deflected parts allow the valve to seat too soon and at too high a velocity, then the stored energy in the parts flips the valve open again, causing it to bounce on the seat. Fracture of the stem and irregular functioning are the results of this weakness. The only way to avoid this trouble and at the same time keep down weight, is to shorten the stress lines, and the overhead camshaft is obviously at an advantage. Rockers should be of I section, and as deep and as short as possible. Fig. 7 shows a simple test for overall deflection, which can be applied to existing parts.

The weight is equal to the positive acceleration force exerted by the flank of the cam at the maximum speed, and can be calculated by the method shown in the section on Cam Design.

SECTION XXVIII

PART III

DESIGN OF I.C. ENGINES — TORSIONAL VIBRATIONS OF CRANKSHAFTS—SCAVENGING AND SUPERCHARGING—MOUNTINGS — FOUNDATIONS — MAINTENANCE, etc.—STANDARD SPECIFICATIONS—B.S.I. AND LLOYD'S RULES

(By R. J. W. Cousins, M.I.Mech.E.)

Design of I.C. Engines

Cylinder Head.

Since the primary purpose of the head is to close the end of the cylinder, it must be strong enough for the maximum explosion pressure, but as it also often forms the support for the valve seats it must be stiff enough to carry the gas load without serious deflection. It should, therefore, be of considerable depth and sufficiently ribbed internally to ensure local stiffness as well as overall strength as a beam. In particular the design should be examined to see that the bottom deck is supported against doming in the centre. When the head is held down by four through bolts, the general outside shape should be rectangular, with straight walls joining the bolt bosses, and when a circular spigot joint is used in conjunction with this bolt arrangement, the corners, where the joint is farthest from the walls, should be reinforced by ribs and generous fillets.

Internal dwarf walls, allowing plenty of space above for coring and water circulation, are also useful in some cases for stiffening the bottom deck.

Where possible a ring of bolts, spaced as regularly as the valve elbows, etc., allow, should be employed, as the smaller span means a more easily made joint with probably a shallower and lighter head.

Aluminium Heads.

When aluminium is used for cylinder heads, valve seats of aluminium-bronze, or a high expansion heat-resisting steel must be fitted. They are usually screwed in (the heads are sometimes heated to about 100° C. for this purpose) and the internal castellations into which the spanner fits are machined away. The guides are made of bronze. The holding-down bolts should be of *high-expansion steel* turned down over the greater part of their length to give plenty of stretch, and the bolt bosses must be of large cross-section, say six times the area of the reduced part of the bolts, otherwise the high expansion of the aluminium will lead to crushing of the bosses or permanent lengthening of the bolts, so that the joint will blow when the engine is cold, and continual tightening only causes further stretching of bolts or crushing of bosses until failure occurs.

In large two-strokes, a water-cooled pad with thin decks is sometimes interposed between the pressure-carrying head and the combustion space so as to reduce the thermal stresses. Cast steel is also used for the same reason, as the greater strength permits the use of thinner sections.

Pistons and Connecting Rods.

See notes and sketches in Part II.

It should be remembered that the piston of the internal combustion engine has usually three functions :—

(*a*) To form a gas-tight plug ;
(*b*) to take away the heat which is unavoidably fed into the piston crown ;
(*c*) to form a crosshead for the small end of the rod (except in certain large engines).

While many designs and proportions are found, there is a growing tendency to use a piston on the following lines, for vehicle and industrial light engines :—

(1) A moderately deep top land—say $0 \cdot 1d + \frac{1}{8}$ in. (d = cyl. diameter).
(2) Three narrow gas rings, $\frac{3}{32}$ in. to $\frac{5}{32}$ in. wide, with $\frac{1}{8}$ in. to $\frac{3}{16}$ in. between.
(3) An oil control ring immediately below the gas rings.
(4) An evenly balanced crosshead portion with the gudgeon pin centrally placed between the oil vent groove and the lower oil control ring at the bottom of the skirt.

The length of this crosshead portion will be cut down to $0 \cdot 35\ d$. for high speed engines and increased to $0 \cdot 7\ d$., for industrial machines where long life is essential, but it should be realised that extending the length on one side only of the pin must lead to asymmetrical loading, and do more harm than good. For this reason the extension shown in fig. 2, Part II, p. 208 though found occasionally, is in the writer's opinion not good practice.

The heat is taken away in most cases by a fairly thick crown (relying on the high conductivity of the almost universally used aluminium) assisted by ribs cast, or pressed, on the underside.

In some cases oil-cooling via the connecting rod is now being introduced, even in small high speed engines, care being taken to get it to the zone behind the rings, since the time required to 'gum up' the latter is largely dependent on the temperature at the grooves. If this is of the order of 230° C. or over, gumming and carbonisation are rapid, but at 200° C. an enormous improvement is effected.

Piston bosses should be well supported from the crown at their inner ends by thick struts, to prevent bending of the walls which would occur with overhanging bosses. Thin ribs over the bosses are no use, they only provide an overstressed part to start cracks.

Gudgeon pins may be from $0 \cdot 25\,d$ for petrol to $0 \cdot 33\,d$ for oil engines, and are almost invariably full floating.

Valves.

Inlet valves used to be made with cast-iron heads and steel stems, screwed and sometimes welded in place, but they are now usually stamped or forged in one piece. Case-hardened stems are not uncommon. Guides are generally of cast iron in iron cylinder heads, and bronze in aluminium—on account of the expansion. Malleable iron guides have also been used and are reported to do well with hard stems. Stems may be made the nearest standard size to $0 \cdot 2\,d + 0 \cdot 125$ in. (d = throat dia.). Stem clearance should not be too tight, particularly in oil engines where slight air leakage does not matter, and sticky stems may cause intermittent action if too close a fit is used. Materials used include 32–37 S.M. steel, silicon-chromium, and 3 per cent. NiO.H.

See notes on Exhaust Valves in Part II, p. 199.

Gas Velocities.

Analysis of a large number of petrol engine results over many years has shown that maximum B.M.E.P. (brake mean effective pressure) occurs at an empirical inlet gas velocity of 130 ft./sec., maximum thermal efficiency at about 150 ft./sec., and maximum power at from 210 to 240 ft./sec., the higher figure applying to larger engines.

Oil engines hold their torque rather better, giving the maximum at about 145 ft./sec., but they have no carburetter restriction and the manifold may be made more generous, as distribution troubles do not arise.

The empirical gas velocity is reckoned as:

$$\frac{\text{piston area} \times \text{piston travel ft./sec.}}{\text{valve throat area}}$$

Valve Lift

Tests show that volumetric efficiency increases with valve lift up to about $0 \cdot 35\,d$ (d = throat diameter), but the gain becomes small towards the end. It is, however, worth going to $0 \cdot 3d$.

Most engines could be improved by a higher lift than the conventional $0 \cdot 25\,d$ and many fall short even of this figure.

Exhaust Valves may be made smaller than the inlets, and when the total space is restricted and the maximum output is aimed at, the relative velocities may be inlet 3, exhaust 4, or even inlet 2, exhaust 3, *i.e.* the exhaust throat diameter may be $0 \cdot 87$ to $0 \cdot 82$ of the inlet throat.

See notes in Part V on Exhaust Valves for high outputs.

Timing.

Inlet Opening.—In oil engines the inlet may open well before top centre. 10° to 15° early is quite common, and with 'blown' (supercharged) engines a greater lead is of advantage in conjunction with a late closing exhaust (say inlet opens 25° before T.D.C., exhaust closes 35° after T.D.C.) as the residual gases are then largely cleared out. This 'overlap,' as it is called, also tends to reduce slightly the pumping losses on the exhaust and inlet strokes as the valve opening is greater all the way.

In normally aspirated multi-cylinder petrol engines too much overlap affects the carburation at low speeds; single-cylinder motor-cycle engines, however, usually have very early opening inlets and late closing exhausts as the total opening periods are thus increased and greater time is given, thereby easing the cam action.

In the case of 'Buchi' exhaust gas driven turbines operating centrifugal blowers, the mean temperature of the exhaust gases is lowered by having an exaggerated overlap of 150° to 160°, so that a large quantity of cool air is passed through the engine and thence through the turbine blades, thus keeping them at a safe temperature.

Inlet Closing.—In hand-started oil engines the maximum quantity of air must be retained, so as to give sufficient temperature rise for starting at hand cranking speed, say 120 r.p.m. For these small commercial engines 25° to 30° after B.D.C. is usual, but in oil engines where other means are provided for starting, or in the spark-ignited engines, the valve may be closed later, so as to obtain the best volumetric efficiency at the running speed. The exact timing depends on the mean inlet gas velocity chiefly, and the following values are suggested:—

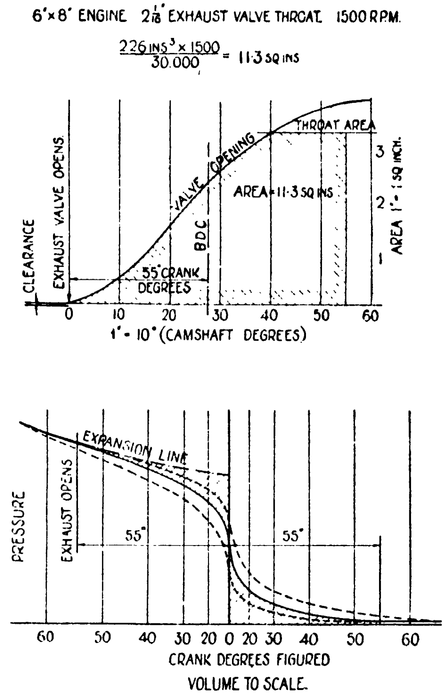
130 to 150 ft./sec.	35° after B.D.C.
170 „ 180 „	40° „ „
200 „ 230 „	45° „ „

Where maximum power is the chief aim the latest timing is the one to use, but if low-speed torque is important 35° to 37° is advisable.

Exhaust Opening.—The exhaust takes place in two stages, first the rapid discharge of the bulk of the weight of gas due to its own pressure, and secondly the mechanical sweeping out of most of the remainder by the rising piston.

The first portion is accomplished while the piston is near the bottom of the stroke and travelling slowly.

If the expanding charge could be retained until the end of the stroke and the pressure then instantly dropped to atmospheric, there would be no work thrown away on the down stroke and no negative work done on the up. This is impossible, but the minimum loss is incurred if the necessary period for the first stage of evacuation is arranged equally each side of B.D.O., because



FIGS. 1 AND 2.

the piston movement is the minimum for the time. The period in crank degrees required depends on (*a*) the swept volume of the cylinder, (*b*) the revolutions per minute, (*c*) the integrated value of the exhaust valve opening area. A simple method which has been applied to a number of engines with success is as follows (see figs. 1 and 2).

Plot the exhaust valve opening area on a base of 1 in. = 10° camshaft movement, and 1 in. vertically = 1 sq. in. valve opening. Then, at some point along the base, the pressure in the cylinder will have fallen to a value near atmospheric. A vertical line at that point cuts off an area of the diagram which in petrol engines is equal to:—

$$\text{A sq. ins.} = \frac{\text{Swept volume of cylinder cubic inches} \times \text{r.p.m. (crankshaft)}}{30{,}000}$$

hence find the position of the vertical line, and half-way between this and the opening point is B.D.O. (See fig. 1.)

In oil engines a devisor of 27,000 may be used to allow for the lower internal energy of the gases due to the impossibility of burning all the oxygen, and the greater expansion ratio.

It is evident that a high lift valve and a cam giving rapid acceleration on the flank, reduce the period required.

Although, as previously stated, the exhaust valve may be made less than the inlet when space is restricted, it should be as large as possible as a few useful pounds mean pressure may be picked up by reducing pumping losses and also by cutting down the period required for the preliminary evacuation. This enables the valve to be opened slightly later, thus increasing the mean pressure near the end of the stroke.

Light spring indicator diagrams should be taken at various speeds and examined critically to find whether any improvement may be effected in either filling up or evacuating the cylinder, by modification in the timing. The cylinder pressure should reach atmospheric when the inlet closes. The exhaust at full load should drop to within a pound or two of atmospheric pressure in a period after bottom centre approximately equal to that between valve opening and bottom centre.

The total lost work represented by the hatched area of the diagram will then have the minimum value. (See fig. 2.)

Two-stroke Ports.

In two-stroke engines the exhaust gases must be evacuated down to a pressure approximating to scavenge pressure by the time the air ports open, and the air and exhaust ports must have sufficient area to allow the scavenge air to drive out the residual exhaust before the piston has risen too far on the up or compression stroke.

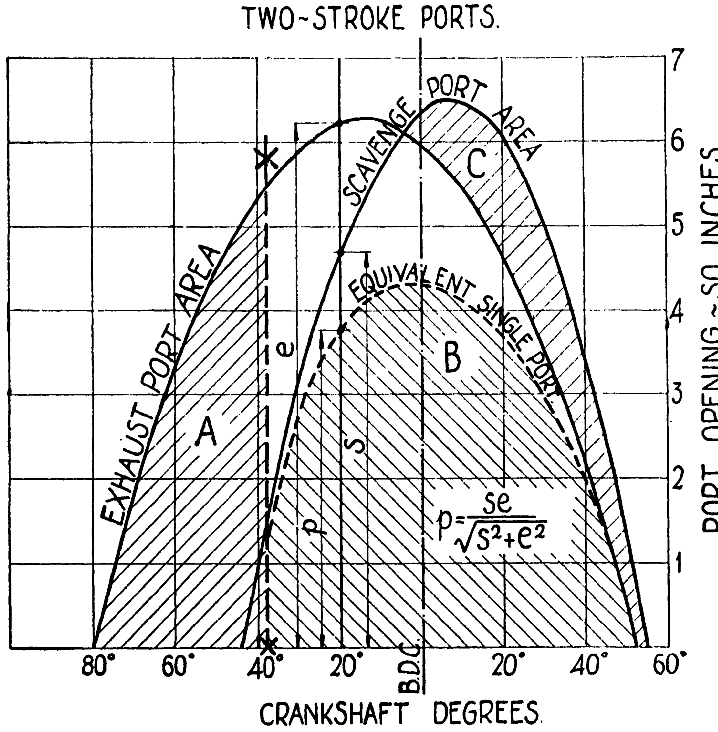


FIG. 3.

The following method, apparently empirical but in reality taking cognisance of all the variables, has been developed and checked from many experimental results.

Draw a diagram of exhaust and air port areas on a basis of 10° = ½ in. horizontally, and 1 in. vertically = 1 sq. in. port area. Then at some point a vertical line X–X will cut off an

area A, such that the integrated value of the port opening with respect to time will be sufficient to discharge the exhaust down to scavenge pressure. Make the area 'A' equal to :—

$$\frac{\text{Swept volume (cub. ins.) of 1 cyl.} \times \text{r.p.m.}}{44{,}000} = \text{A sq. ins.}$$

when using the scales mentioned. The divisor 44,000 is suitable for scavenge pressure 15 ins. Hg, say $7\frac{1}{2}$ lbs. per sq. in.

For low scavenge pressures, 3 to 4 lbs., use 40,000. The time of exhaust port opening and its rate of increase should be such that the line X–X does not encroach more than 5° or so on the scavenge port opening—otherwise blow-back into the air belt will become serious.

The dotted line enclosing area 'B' is the graph of a single port having the same resistance to flow as the combined scavenge and exhaust, and is found as follows :—

At any instant, *i.e.* on any vertical line, let s and e equal the scavenge and exhaust opening areas ; then the equivalent single port is :—

$$p = \frac{se}{\sqrt{s^2 + e^2}}$$

and the enclosed area B (the integrated value of the 'blow through' area) should be such that the available scavenge pressure will be able to pass about 1·1 cylinder volumes per revolution.

Since the scale is $\frac{1}{2}$ in. = 10°, 1 rev. = 18 ins. Then $\frac{B}{18}$ is the mean value of a constantly open port.

$$\frac{\text{Swept vol. (cu. ins.)} \times \text{r.p.m.} \times 1{\cdot}1}{12 \times 60} \times \frac{18}{B} = \text{speed ft. sec. through the equivalent single port, } p.$$

The mean speed is approximately $200\sqrt{P}$ when P = scavenge pressure in lbs./sq. ins., and the air volume is measured at N.T.P. In each case it must be considered whether the pressure or the port areas should be adjusted to suit the desired speed and volume.

Independent control of the timing of exhaust and scavenge ports is practically essential for high-speed two-stroke engines, and some ringing of the changes with the above diagram will enable the designer to cater for each phase of the operations. The last point is that the scavenge area must predominate towards the end, so that the cylinder pressure may be raised to scavenge pressure by the time the ports close. The area O is not critical—20 per cent. of B will suffice.

Cams.

To understand the action of the cam it is necessary to divide its functions into successive phases as under :—

(*a*) Radially outward or positive acceleration of the tappet, etc., up to a maximum velocity which depends on the lift, period and r.p.m.

(*b*) Deceleration, *i.e.* radially inward or negative acceleration of the parts by the valve spring or springs, which must exert sufficient force to keep the tappet in contact with the cam nose, and bring the parts to rest at the full lift.

(*c*) Continuation of the radially inward thrust of the springs until the point of maximum inward velocity is reached where the cam nose joins the flank.

(*d*) Slowing down of the tappet, etc., by the action of the cam flank.

(*e*) Final easy curve which reduces the velocity to a very low figure so as to seat the valve gently.

Taking these in detail :—

(*e*) Where silence is of great importance as in car engines, a gentle connecting curve is arranged between the base circle and the flank of the cam at both beginning and end.

For small car engines which run fast, a slope of 0·0003 in. to 0·0005 in. radial lift per cam degree is satisfactory, and may cover a total lift of 0·010 to 0·015 in. from the base circle. Of this 0·004 in. to 0·008 in. will be clearance.

Larger engines may have 0·001 in. per cam degree, and where silence is not of primary importance this 'approach' curve may be dispensed with. There is now sufficient experimental evidence that the noise due to the contact between tappet and valve at the beginning of lift is negligible, and that the most clatter comes from rapidly seating valve heads. It seems unnecessary therefore to employ an easy approach at the leading side but only at the trailing flank.

Another reason for this is that a smart opening of the exhaust valve helps to prevent the burning and scouring which occurs if the agony be prolonged by allowing the high temperature gases to continue at high velocity through a very small opening.

(*a*) and (*d*) : The flank proper is that part of the cam, joining the base circle to the nose, where the radial velocity continually increases.

The rate of increase or acceleration depends on the period available and the lift required, but it should be greater than the spring force, *i.e.* the time should be divided so that the two sides of the nose have more than the two flanks. This reduces the spring force required.

Constant acceleration cams are sometimes calculated and constructed, but there is no reason why the outward acceleration force should not rise, as in the well-known tangent cam, used with a roller or curved tappet. Rounded flanks *must* be used with the popular flat-ended tappet

otherwise there is violent shock and the gear will be horribly noisy and will not live. Very high maximum accelerations up to about 200g. (6,500 ft. sec. sec.) may be used on cam flanks, but the hollow flank with its violently rising curve is not recommended except for short-lived 'stunt' engines.

(*b*) and (*c*): The shape of the nose determines the deceleration, and the springs must be adequate to keep the tappet on the cam and prevent 'bouncing' at the maximum designed speed.

The spring or springs necessarily exert most pressure when fully compressed, *i.e.* when the valve is full open and the tappet is on the apex of the cam, and less pressure when farther down. It is, therefore, reasonable so to design the cam nose that advantage is taken of this, by having a higher negative acceleration at the apex than at the junction of nose and flank. The spring may then exert a force proportional to that required to follow the cam, over the whole of the nose period.

When R/L is not greater than unity this condition is satisfied (see accompanying graphs, figs. 5 to 8, for cam calculation), *i.e.* the *required* force is greatest at the apex where the *available* spring force is also at a maximum. Graphic differentiation of the valve lift curve (on a time base) gives the velocity curve, and a second differentiation of this gives the acceleration. The points on each successive curve are given by the value of the tangent of the angle of the slope of the preceding curve at each point.

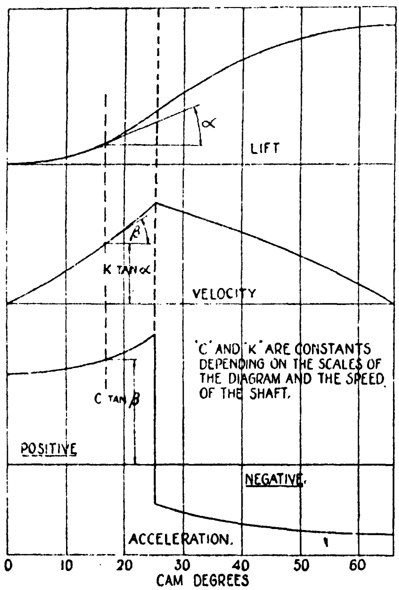


FIG. 4.

This method, however, calls for a very high degree of accuracy in draw ng and is not recommended.

The graphs given (figs. 5 to 8) cover noses and flanks of cams of normal proportions, constructed of circular arcs and having flat tappets, or followers with rounded ends, or rollers.

R is the radius in inches from camshaft centre to centre of curvature of cam profile.

L in every case equals the radius in inches from the centre of curvature of the cam to the centre of the tappet or follower.

In the particular case of tangent flank cams, R equals the distance from the camshaft centre to centre of tappet end when the latter is on the base circle (see small diagrams, fig. 8).

R.p.m. is the crankshaft speed, *i.e.* twice cam speed.

C is a constant from the graphs depending on the $\frac{R}{L}$ ratio and the angle from apex for nose acceleration, or the angle from the base circle for flank acceleration.

Then the acceleration $= \dfrac{R \times C \times \text{r.p.m.}^2}{100{,}000} =$ A ft. per sec. per sec.

If W lbs. is the effective weight of the parts, and the acceleration of the cam apex is A ft. sec. sec., the spring pressure must be not less than—

$$\frac{A \times W}{g} \text{ lbs. } (g = \text{gravity} = 32 \cdot 2).$$

Allow a margin for overspeed, manufacturing errors and friction, say 30 to 50 per cent.

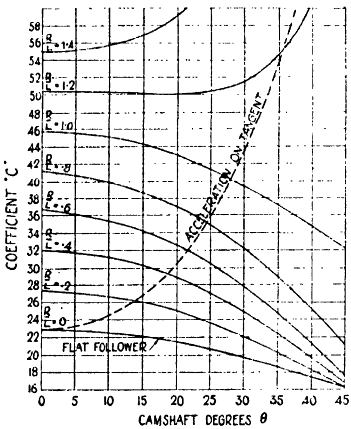


FIG. 5.—Circular Noses and Tangent Flanks.

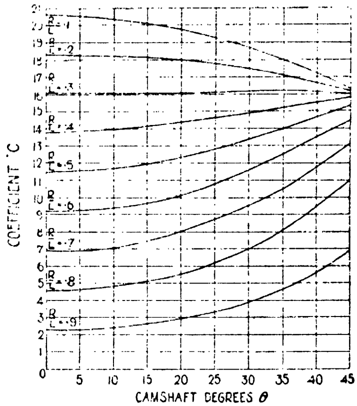


FIG. 6.—Round Flanks.

Acceleration $= \dfrac{C \times R \times N^2}{100{,}000}$.

C = coefficient from table according to angle and proportions of cam.
R = Radius in inches. See fig. 8.
N = Revs. per min. of crankshaft assuming camshaft runs half engine speed. Dotted curve gives acceleration on tangent.
θ = Angle in camshaft degrees.

When $\frac{R}{L} = 0$, **then L = ∞ equivalent to the case when a flat follower is used.**

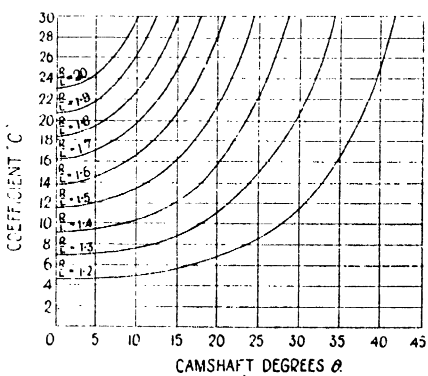


FIG. 7.—Hollow Flanks.

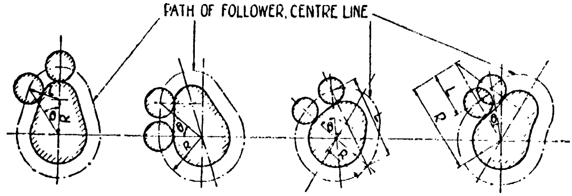


FIG. 8.

The equations for the instantaneous value of the acceleration at any point in the cases covered by the curves are as under:—

Straight line tangent to base circle:—

$$W^2R\left(\frac{1 + 2\tan^2\theta}{\cos\theta}\right) \text{ ft. sec. sec.} \qquad (1)$$

Round nose and round or hollow flank—circular arcs:—

$$W^2R\left(\cos\theta + \frac{n^2\cos 2\theta + \sin^4\theta}{(n^2 - \sin^2\theta)^{\frac{3}{2}}}\right) \text{ ft. sec. sec.} \qquad (2)$$

Simple harmonic cams (circular arcs) with flat followers—

$$W^2R\cos\theta \text{ ft. sec. sec.} \qquad (3)$$

R = radius in feet from shaft centre to centre of curvature for (2) and (3), and shaft centre to roller or tappet centre for (1).
W = angular velocity of the camshaft in radians per second.
θ = Angle moved through by cam from base circle (or origin of the cam) in the case of flanks in (1), (2) or (3), and in the case of the nose (2) or (3) the angle from the apex = $180 - \theta$.
$n = \frac{L}{R}$ when L is the radius of curvature in feet.

To use r.p.m. of the crankshaft instead of radians per second of the cam, and R and L in inches instead of feet, multiply the results by 0·000228. In all tappets, and in flat followers particularly, be careful that the head is large enough to cover the maximum eccentricity of the point of contact, which occurs at the junction of nose and flank.

Effective Weight of Valve Parts.

In directly operated valves when the cam lift equals the valve lift, simply take the weight of valve, cotters, spring carrier, tappet, etc., and add one-third of the spring weight (a rough estimate of the latter must be made and corrected later if far out).

When an unequal rocker of ratio R to 1 is used, the acceleration on the parts at the valve end is in direct proportion to that at the cam (or tappet) end. The force per pound for the valve end parts is therefore increased in the ratio R.

Assume acceleration Ag at the valve, *i.e.* A times gravity. Then the acceleration a the cam is $\frac{Ag}{R}$.

Take the weight to be W_c and W_v for cam and valve ends. Then the force directly applied to the parts at the cam end is $W_c \times \frac{Ag}{R}$ but, since it is usually done through the rocker from the spring at the valve end, it is reduced $\frac{1}{R}$ of this amount by the leverage, *i.e.* $\frac{W_c\,Ag}{R^2}$. This means that to find the *effective weight at the valve*, the weight of the moving parts at the cam end is divided by the *square of the rocker ratio* and added to the parts at the valve end.

This total weight is multiplied by the cam acceleration and by the rocker ratio to find the forces at the valve.

Flat Followers.

Chilled cast iron is sometimes used; reports vary as to its behaviour. If hard enough it is very good, but if not, it 'rings,' *i.e.* develops deep circular grooves, badly and quickly. (Incidentally all tappets and valve gears seem either to wear very well or tear up in the beginning. Good hardening and oil in the places where it will be drawn between the loaded surfaces, seem to be the chief desiderata.)

Probably the quickest way of arriving at a suitable cam is to try it out on the drawing-board. Determine the timing first—giving as long a period as possible considering the engine speed, etc. Then lay off the gentle approach curve if used, remembering that since it opens the valve so little it does not account for much gas flow and the period may therefore be extended a little. Next fix the cam apex, giving the desired lift and period, and then fill in with tangent or cuved flanks, varying the nose radius until a profile giving reasonable spring pressures, etc., is found. A 'hammer headed' tappet with fixed curved end not only works well but gives great latitude in design because, by increasing the radius of the curved end, the nose acceleration may be reduced and the flank acceleration increased or *vice versa* without altering the cam. For any given period and lift smaller camshafts and base circles are also possible with curved followers as compared with flat ones, as with the latter the nose acceleration depends directly on the radius from shaft to nose centre, while with a curved follower it may be twice this figure.

The radius of the nose itself (*i.e.* from nose centre to cam face at the apex) may be very small indeed, and it is often necessary with flat followers to reduce it to 0·075 in. to get the requisite lift without increasing the overall diameter of the cam.

At speed the nose is practically unloaded, while when running slowly the spring pressure is all it has to sustain. The flanks do the positive acceleration and take the gas pressure when opening the exhaust valves.

Valve Springs.

The principal cause of valve spring failure is fatigue, due to the actual stress *range* being too great. (Not the maximum stress, but the difference between maximum and minimum.) Usually this is not due to the stress range accounted for by the normal valve lift, but by the extra relative travel of the coils when surging takes place.

This surging is more likely to occur at high speeds when the natural frequency of the spring tunes in with one of the harmonics of the cam profile, usually starting when the camshaft speed is as high as one-tenth to one-twelfth of the spring frequency and being bad at one-eighth. As the engine speed is usually fixed, the best thing to do is to keep the spring frequency as high as possible.

This frequency is approximately—

$$F = 0{\cdot}213 \times 10^6 \times \frac{d}{R^2N}$$

where,

F = frequency vibrations per minute.
d = diameter of wire.
R = Mean radius of coils.
N = number of active coils.

or—

$$F = 590\sqrt{\frac{S}{W}}$$

where,

S = stiffness in lbs. per 1 in. axial deflection.

W = weight in pounds of the active coils. (Do not count the two end coils as they are dead. Now, since W (the weight) is proportional to the number of coils (N), and S (the stiffness) is inversely proportional to N,

$$F \propto \sqrt{\frac{S}{W}} \propto \sqrt{\frac{1}{N^2}}$$

i.e. frequency varies inversely as the number of coils, other things being equal.

It is clear that for a fixed valve lift the working deflection per coil, and therefore the norma stress range, also varies inversely as the number of coils.

Therefore the *Spring Frequency is proportional to the Stress Range for a fixed working travel.*

The situation then is this:—

(*a*) If a low stress range is used for the normal lift there is a danger of surging of the coils and fatigue failure from this cause, because the spring frequency is too low.

(*b*) If too high a stress range is employed, failure may occur from the normal travel.

The best solution seems to be to use the highest *safe* range possible, thus keeping the frequency high and avoiding surging, without incurring the risk of failure from the second cause.

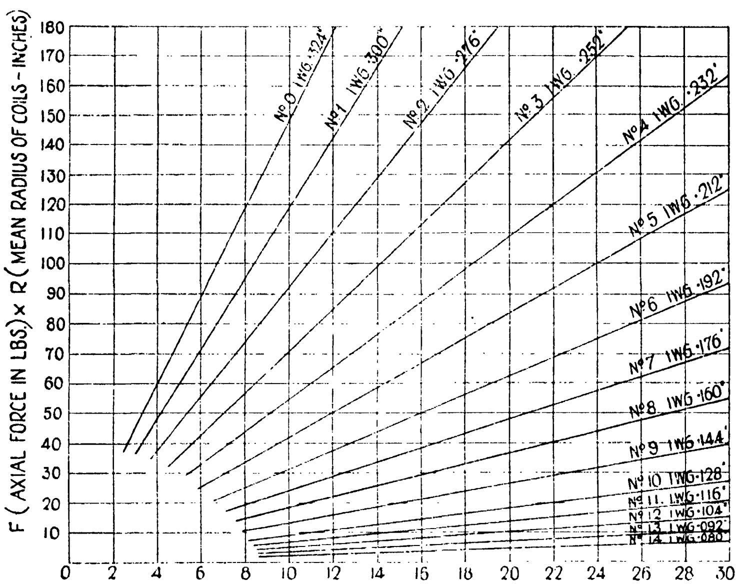
If the *maximum* stress is not too great a larger *range* may be allowed, and from the experience of a number of cases, the following stresses are suggested:—

Maximum . .	24 to 28 tons/sq. in.	
Range . .	14 ,, 12 ,, ,,	(the higher range with the lower maximum).

This is for good quality commercial springs.

The maximum force required (F) having been found from the cam form and valve gear weights, the spring may now be calculated.

Generally the design suggests a convenient outside diameter, and from this a mean radius R to the centre of the coils may be assumed. The stress graphs (fig. 9) give the stresses for



PARALLEL HELICAL SPRINGS ~ ROUND WIRE.

FIG. 9.

different standard gauges of wire for any value of F × R, and from this the correct wire may be chosen at once to come within the maximum permissible stress decided upon.

The deflection per coil is then found from the second graph (fig. 10), multiplying the deflection for 1 ton per sq. in. by the actual maximum stress employed. To find the number of *useful* coils, divide the *total* deflection required by the deflection per coil as found above. Add two for the dead coils at the ends.

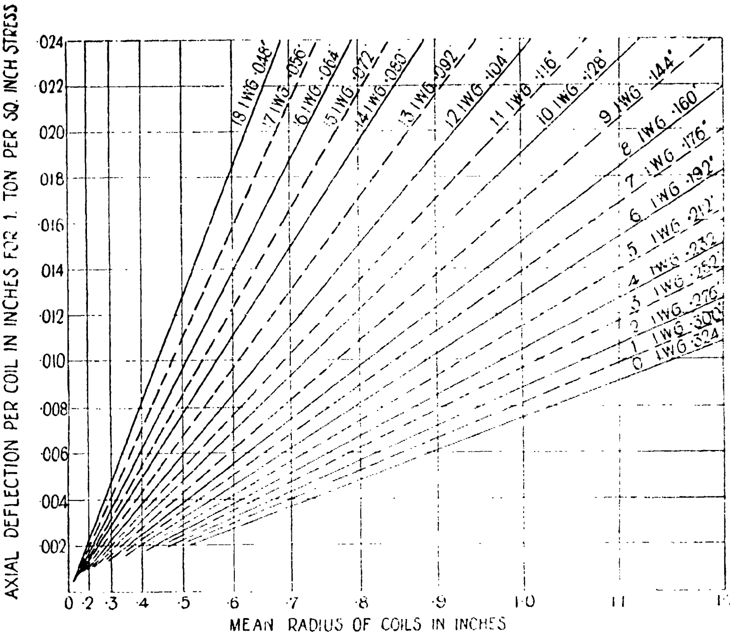
$$\text{The } \textit{total deflection} \text{ from free length to valve open} = \frac{\text{Maximum stress} \times \text{working lift}}{\text{Working stress range}}$$

The minimum working length (valve open should be just a little longer than the solid length. This helps to discourage surging, as it causes the coils to touch, and introduces some interference with the wave motion every time the spring is compressed. The free length is the minimum working length plus the total deflection.

The spring force must exceed the nose acceleration force at all points from the apex to the junction of nose and flank, and as the spring force decreases from the apex, it is important that the acceleration force should also decrease—see 'Cams' (p. 219).

With a curved or roller follower, the cam motion must be measured at the *centre of the follower*, not on the cam surface. Similarly with flat followers the radial motion of a point on the tappet centre line gives the correct reading.

Springs should not be too small in the mean diameter compared with the thickness of the wire, as an additional stress is thereby introduced. The graph (fig. 11) gives the addition, which should be allowed for if appreciable.



DEFLECTION - PARALLEL HELICAL SPRINGS - ROUND WIRE

FIG. 10.

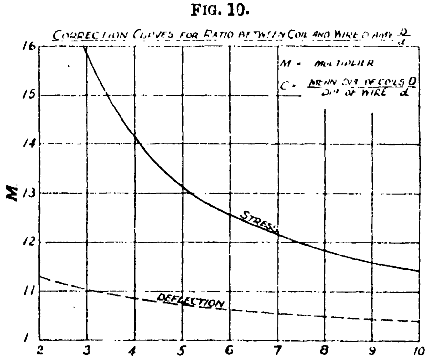


FIG. 11.

The equations for stress and deflection in ordinary parallel helical springs of round wire are:—

$$\text{Stress (lbs./sq. in.)} = \frac{16FR}{\pi d^3}$$

$$\text{Deflection} \quad . = \frac{64\ FR^3 n}{C\ d^4}$$

F = axial load lbs., R = mean radius of coils inches, d = wire diameter inches, n = number of free coils, C = transverse modulus of elasticity = $11 \cdot 8 \times 10^6$ approx. for spring steel.

Since the stress due to surging of helical springs is greatest at the ends (theoretically twice, practically about 1·5 times at the point of reflection of the wave) and breakages almost always occur in the first free coil, it would be an advantage to bring the two ends in to a smaller diameter, provided the ratio $\frac{\text{coil diam.}}{\text{wire diam.}}$ is not already so small that the modification will cause an even greater increase to offset the gain. Reported failure to derive any great advantage from 'barrel' springs may possibly be due to this cause (see graph, fig. 11, showing this effect).

Investigation by the N.P.L. has shown that the great discrepancy between the fatigue range of polished laboratory specimens and that of actual springs is due largely to the decarburising of the outer skin by heating in the presence of free oxygen during manufacture. This leaves in effect an outer layer of mild or low carbon steel, much weaker than the core, at the most highly stressed zone. Minute cracks form in this layer and the stress concentration at the bottom of each small fissure soon destroys the underlying part. Grinding of the wire between draws and tempering in an inert atmosphere have enabled springs to be made for aero engines which will stand a very much increased range, but at present this is hardly economic for commercial engines.

If the crankshaft speed r.p.m. be multiplied by the valve lift in inches, a figure of merit is given which is a measure of the tendency to surge. If the product is over 1,250 (say ½" lift at 2,500 r.p.m.) some surging may be expected, and at 1,500 it is certain, even with a high stress range, while 2,000 is usually destructive. The manufacturers of racing motor cycles (which generally have only two valves per cylinder, and consequently large lifts in proportion to the speed) have been driven to revive the old 'hairpin' spring, in which the wire is stressed in bending, and the coils do not move bodily as in the usual helical spring.

With the popular duplex arrangement the maximum force is divided between four wires.

The bending moment is this force (negative acceleration force divided by 4) multiplied by the radius from point of application on valve to centre of coil.

The bending stress is maximum and uniform in the coils, and varies from this figure to zero at the ends of the arms forming the abutment as well as those engaging with the valve.

Force required F lbs.

Force on each wire, $\frac{F}{4}$.

Bending moment $M = \frac{FR}{4}$ (R ins. = radius from coil centre to end of arm).

Wire diam. d ins. stress $= \frac{32\ M}{\pi d^3}$ lbs./sq. in. (round wire).

$\frac{6M}{d^3}$ for square wire of side d ins.

The maximum stress should not exceed 45 tons/sq. ins., and the range 20 tons for this class of spring.

The total deflection aimed at may be from twice to three times the valve lift—2·5 being a usual figure. The deflection of the spring depends on the length of wire in the coils—the arms accounting for very little as under :—

Length of arm to valve = R ins. deflection $= \frac{MR^2}{3\ EI}$.

Length of arm to abutment = A ins. deflection $= \frac{MA^2}{3\ EI}$.

$E = 30 \times 10^6$ for spring steel. I = moment of inertia of wire in bending.

$$I = \frac{d^4\pi}{64} \text{ for round wire.}$$

$$I = \frac{d^4}{12} \text{ for square wire.}$$

Take these two deflections from the total required and the remainder must be the deflection of the coils. The length of wire necessary (L ins.) is found from :—

$$\text{deflection of coil} = \frac{MRL}{EI} \text{ or } L = \frac{\text{def.} \times EI}{MR}$$

The mean diameter of coils is then chosen to give $n + \frac{1}{2}$ coils so that the arms are in the correct relationship. The diameter of coil has no bearing on the stress or deflection. Springs of this design have been run up to ½ in. lift at 4,000 r.p.m. crankcfaft speed without any sign of surging —a figure quite impossible with helical springs.

Crankcases.

In some engines the upper part of the crankcase also forms or contains the cylinders, while in others separate cylinder blocks or individual cylinders are bolted on.

In the latter case especially it is advisable, for the sake of vertical longitudinal stiffness, to carry the crankcase walls below the shaft centre, thus giving extra depth, while a wide flange at the bottom joint (where the crankcase or column joins the sump or bedplate) will stiffen the structure horizontally. Where the cylinders or jackets are formed integrally with the crankcase the latter may end at the shaft centre line where it bolts to a light sump. The stress lines due to the gas forces should be made as direct as possible, so that the flow of stress from cylinder base to the bearings on each side may be free from sudden alterations in direction which introduce heavy stress concentrations and deflections.

Crankshafts.

One of the most important members and one which presents some of the most interesting problems in design is the crankshaft.

The principal questions arising are:—

(1) Materials.
(2) Balance.
(3) Torsional resonance.

(1) *Materials.*—In very large engines mild steel, or recently rather better S.M. steel of 35 to 37 tons/sq. in. ultimate, is used almost exclusively, as the large sections demand a steel easily manufactured and heat-treated, and of sufficient ductility to accommodate itself without damage to the internal stresses consequent on its thickness.

As smaller sizes are reached, stronger and harder steels, such as tempered high carbon, are used, while in most car engines high tensile alloy steels are used—so as to provide a harder surface for pins and journals.

Aero engines use the highest class alloy steels up to 70 tons tensile, and are bored out for lightness.

In automobile work cast shafts of steel or special iron have been introduced recently, and the latter particularly are very cheap to produce, and not only wear very well, but have very high hysteresis or internal friction, which, combined with a low stress/strain modulus, gives heavy damping and reduces resonance.

These special irons such as 'acicular' and 'nodular' irons, are held to a close specification, particularly for carbon content, and ladle additions are made which become finely divided and well distributed in the metal, each particle forming a nucleus about which the graphitic carbon collects. This prevents or reduces the formation of the characteristic laminae or thin plates of crystalline graphite which weaken the metal by preventing continuity over large areas.

Cast-iron shafts must be of generous proportions, particularly web thicknesses, and while flying webs may be used in petrol engines the use of bearings between all cranks is imperative in the case of compression ignition engines.

Pins and journals should be cored out to give more uniformity of metal section and balance weights of any reasonable shape may be cast integrally.

The torsional frequency is lower than for a similar steel shaft in the ratio of the square roots of the respective moduli of rigidity, *i.e.*

$$\sqrt{\frac{8 \cdot 25 \times 10^6}{11 \cdot 8 \times 10^6}} \text{ say } \underline{0 \cdot 835.}$$

Another development is the hardening of steel crankpins when the shaft is practically complete

Two methods are employed :—

(*a*) An oxy-acetylene flame is directed on to the slowly rotating pin, followed immediately by a jet of water, both flame and water jet being just the width of the pin. A very hard carburised skin is obtained. The job is done in a special machine, and the technique has called for considerable development work.

(*b*) A suitable steel is used which will harden sufficiently by quenching, and the pin is surrounded by heavy coils (made split for ready dismantling) through which very high frequency alternating current is passed. The magnetic hysteresis produces a high surface temperature, and after the necessary time water is forced through slots between the coils, quenching and hardening the surface. The object is to produce a very hard surface to resist wear, particularly when using lead bronze bearings.

Balance.—All the masses forming or attached to the crankshaft may be separated into :—

(*a*) Revolving masses.
(*b*) Reciprocating masses.

The connecting rod may be represented by two masses, one revolving with the crankpin, and the other reciprocating with the piston. These masses when in motion give rise to revolving or reciprocating forces and couples, the latter occurring when one force is balanced by another not in the same plane (*e.g.* in a two-cylinder engine with crank at 180°).

These forces and couples may be primary, *i.e.* at crankshaft speed, or (due to the obliquity of the connecting rod acting on the reciprocating masses) secondary, *i.e.* at twice crankshaft speed.

Forces or couples of a higher order are too small to be troublesome.

If the connecting rod be suspended horizontally by the crankpin and gudgeon pin centres, from suitable balances, the weights recorded are the revolving and reciprocating components respectively, and the latter is added to the piston (complete with pin and rings) to give the total reciprocating mass = W_1 pounds. The big end mass is added to the crankpin and the equivalent unbalanced mass of the webs to give the total unbalanced revolving mass = W_2 pounds. These two figures W_1 and W_2 with R ins. crank radius, L ins. connecting rod length, O distance from centre to centre of cranks longitudinally, and the r.p.m. enable the forces and couples to be calculated :—

$$\text{Reciprocating force at TDO} = 28 \cdot 2 \left(1 + \frac{R}{L}\right) R\, W_1 \left(\frac{RPM}{1{,}000}\right)^2 \text{ lbs.}$$

$$\text{BDO} = 28 \cdot 2 \left(1 - \frac{R}{L}\right) R\, W_1 \left(\frac{RPM}{1{,}000}\right)^2 \text{ lbs.}$$

Revolving force—radially outward at all points—

$$= 28 \cdot 2\, R\, W_2 \left(\frac{RPM}{1{,}000}\right)^2 \text{ lbs.}$$

The vertical component is $28 \cdot 2\, R\, W_2 \left(\frac{RPM}{1{,}000}\right)^2 \cos\theta$ lbs.

θ = angle from T.D.C.

The reciprocating force may be separated into :—

Primary, $28 \cdot 2\, R\, W_1 \left(\frac{RPM}{1{,}000}\right)^2 \cos\theta$

and Secondary, $28 \cdot 2\, R\, W_1 \left(\frac{R}{L}\right) \left(\frac{RPM}{1{,}000}\right)^2 \cos 2\theta$

(This is a close approximation—the error is not of practical importance.
Tabulate the values of θ and $\cos 2\theta$ from T.D.C. to B.O.D.

θ	2θ	$\cos 2\theta$
T.D.C. 0°	0°	+ 1
45°	90°	0
90°	180°	− 1
135°	270°	0
B.D.C. 180°	360°	+ 1

The secondary force is therefore upward at both T.D.C. and B.D.C., and downward at 90°, its value depending on the crank to connecting rod ratio. With an infinitely long rod $\frac{R}{L} = 0$ and there would be no secondary forces. This is the case with the 'Scotch Yoke' used in boiler feed pumps, where the connecting rod is replaced by a block on the crankpin sliding at right angles to the piston rod.

A simple method of finding the forces and couples of a crankshaft is as under :—

Primary Forces and Couples.

Draw the end view diagrammatically, placing the centre crank or the two centre cranks vertically.

Project horizontally and vertically as shown in fig. 12. The horizontal view will normally be symmetrical, showing that no unbalanced couples are present tending to displace the shaft in that plane.

The end view in all except single cylinder engines should be an equiangular star, showing that there are no free translational primary forces shaking the engine bodily, but the vertical view gives the opposing couples which may now be worked out by multiplying the projected length of each crank by its distance from the longitudinal centre line. In this way the best crank arrangement for any number of cylinders may be found by a few simple sketches, and the balance weights necessary to neutralise the larger couple can be determined. Note that only the *revolving* masses can be balanced by revolving balance weights, but *half* the reciprocating masses may be included, in which case *two couples at right angles*, but each of half the value, will replace the single one.

To find the secondary forces and couples, draw the same views with the centre crank or pair of cranks vertically, but double *all the angles* between the cranks as in fig. 13.

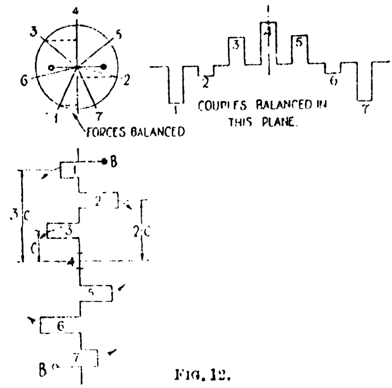


FIG. 12.

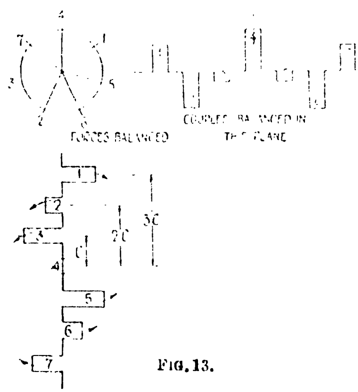


FIG. 13.

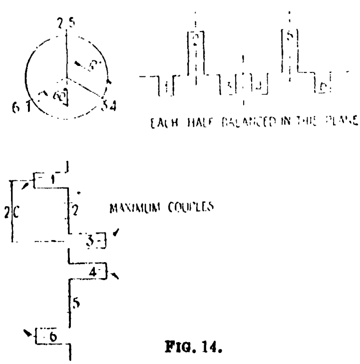


FIG. 14.

The secondary out of balance couple in fig. 13 is not observable in practice, as the polar inertia of the whole engine is so great in proportion to the small unbalanced secondary couple.

The idea is thus :—

(1) Free forces, if any, will show in the end view.

(2) If the horizontal view is drawn so that no *couple* is shown, *i.e.* so that the cranks form a symmetrical figure about the centre of length, then the vertical view at right angles will contain the maximum couple if any exists.

In the four-cylinder engine the primary forces and couples are balanced, but the secondary diagram is bad, as all the virtual cranks come to the top together, showing that all the secondary forces are added. They total about 1 crank in the primary $\left(\frac{R}{L} \times 4 = 1 \text{ approx.}\right)$ and shake the engine bodily in a vertical direction. In cars the secondary forces are now usually circumvented, by allowing the engine to move up and down on its rubber or spring mountings.

Frame Couples.

With four, six, eight cranks (or any even number) one-half usually forms a mirror image of the other so that the couples are balanced against each other. Though no free couples are present, the balancing is done through the bearings and frame, and therefore the latter must be strong and stiff enough to stand them.

Take a six-cylinder engine with symmetrical shaft arrangement consisting of two 3 throw cranks forming right- and left-hand helices (see fig. 14).

Since the two halves do not show any couple in the horizontal view, it will be maximum in the vertical, and is equal to $2\,C \times \sin 60° \times W_F$ lbs. ins. Where W_F = the centrifugal force in lbs. due to one crank weight W_1 at the running speed, *i.e.* $28 \cdot 2\ W_1 R$ ins. $\left(\frac{\text{RPM}}{1{,}000}\right)^2$. The frame must stand this bending moment horizontally, and for the vertical forces add W_2 (the reciprocating weight) to W_1 (the revolving). Treat other numbers of cylinders similarly.

Flywheels.

The function of the flywheel is to form an available store of kinetic energy for the following purposes :—

(1) To assist starting.
(2) To prevent sudden run away due to cutting off the load so that the governor may have time to act.
(3) To reduce speed variation and prevent flicker of lights, etc.
(4) To limit phase variation, *i.e.* advance and retard with reference to uniform angular velocity, thus enabling several generating sets to run in parallel on alternating current.

The energy stored in the flywheel is proportional to the mass of the rim multiplied by the square of its linear velocity, *e.g.*—

Weight W lbs., velocity at mean radius v ft. sec.

Energy in ft. lbs. $= \frac{Wv^2}{2g}$ (g = gravity = $32 \cdot 2$).

This may also be expressed as polar inertia (mass by radius ²) multiplied by angular velocity and divided by 2.

The polar inertia of a flywheel of outer diameter D, inner diameter d and face f (all in feet) is :—

$f(D^4 - d^4)\,\frac{\pi}{32} \times \frac{w}{g}$ when w = weight of 1 cu. ft. of the material and g = gravity.

Multiply by the square of the angular velocity in radians per sec. and divide by 2 to get the energy in ft. lbs.

Cyclic Irregularity and Phase Displacement.

The diagrams show the method of finding these, given the total polar inertia of the revolving mass (including flywheel, crankshaft with balance weights if any, and any rigidly attached mass, such as a generator) together with the torque curve.

The latter may be developed from an indicator card by multiplying the available pressure at any point by the piston area, and by the instantaneous value of the effective crank radius (fig. 15).

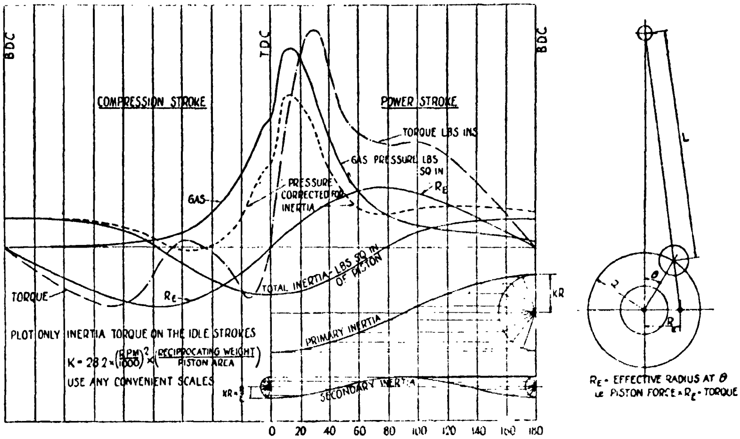

BDC
TDC
BDC
COMPRESSION STROKE
POWER STROKE
TORQUE LBS INS
GAS PRESSURE LBS SQ IN
PRESSURE CORRECTED FOR INERTIA
GAS
TOTAL INERTIA LBS SQ IN OF PISTON
TORQUE
PLOT ONLY INERTIA TORQUE ON THE IDLE STROKES
USE ANY CONVENIENT SCALES
PRIMARY INERTIA
SECONDARY INERTIA
KR
L
R_E - EFFECTIVE RADIUS AT θ
i.e. PISTON FORCE × R_E = TORQUE
0 20 40 60 80 100 120 140 160 180

FIG. 15.

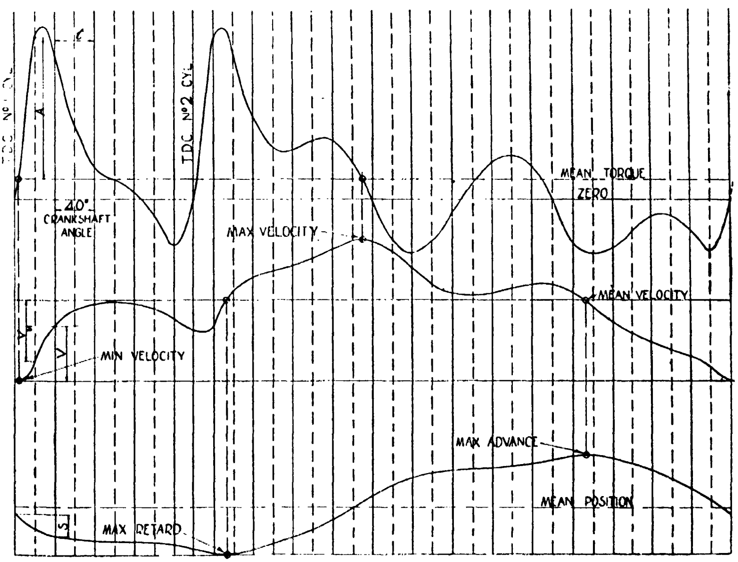

TDC Nº2 CYL
40° CRANKSHAFT ANGLE
MEAN TORQUE
ZERO
MAX VELOCITY
MEAN VELOCITY
MIN VELOCITY
MAX ADVANCE
MEAN POSITION
MAX RETARD

FIG. 16.

The first integration of the torque curve gives the velocity, and the second integration the phase alteration or angular movement fore and aft of a point at uniform velocity (see diagrams, fig. 16).

Method.—Divide the torque diagram (on a time base) into any number of equal divisions by vertical ordinates and dot in the mid ordinates, the time for each division is t secs. The height A at each mid ordinate, above or below the mean torque line, is a measure of the velocity V gained or lost by the *end* of that division, and the velocity curve will rise or fall by that amount in that division (use any convenient scale for each curve). Similarly the height of the mid ordinate V_M in each section of the velocity curve above or below the mean velocity for the whole cycle, gives the advance or retard S by the end of the section on the phase or displacement curve.

The actual value may be computed as under—the letters refer to fig. 16 :—

t = time (seconds) for 1 division.

A = mean angular acceleration force (*i.e.* torque lbs. ft.) in each division above or below mean torque line.

I_p = polar inertia $\frac{\text{lbs. ft.}^2}{g}$ of flywheel, etc.

V = velocity increase or decrease at *end* of time t.

$$= \frac{At}{Ip} \text{ radians per sec.}$$

V_M = difference between mean velocity during time t and mean velocity for the whole cycle.

S = space moved through during time t

$$= V_M t \text{ radians.}$$

The curves have been worked out for a number of engines from one to six cylinders, and the results are given below :—

V = swept volume of one cylinder in cubic inches.

Ip = polar inertia of the whole system (flywheel, crankshaft, generator, etc.) in lbs. mass × ft².

i.e. $\frac{\text{Weight}}{g} \times (\text{radius of gyration in ft.})^2$

g = gravity 32·2.

R.P.M. = revolutions per minute.

N = number of pairs of poles in the alternator (for phase alteration).

O = positive or negative values of constant for percentage cyclic irregularity, giving rise above or fall below the mean speed.

K = positive or negative values of constant for phase displacement in electrical degrees.

Then cyclic speed variation above and below mean speed—

$$= \frac{V \times O}{Ip \times RPM^2} \begin{array}{l}\text{per cent. up (O positive)}\\ \text{or down (O negative)}\end{array}$$

and phase displacement—

$$= \frac{V \times N \times K}{Ip \times RPM^3} \text{ electrical degrees fore or aft.}$$

Add the positive and negative values of O or K in each case for total variation.

Note the advantage of high revolutions in reducing the cyclic irregularity as the *square* of the speed and the phase displacement in effect as the *cube*, since (for the same electrical frequency) N (the number of pole pairs) decreases as the designed speed is raised, thus reducing the displacement in electric degrees.

Thus on the National 'grid'—50 cycles per sec. = 3,000 cycles (min.).

500	r.p.m.	= 6	pairs	of poles	in the	alternator.
600	,,	= 5	,,	,,	,,	,,
750	,,	= 4	,,	,,	,,	,,
1,000	,,	= 3	,,	,,	,,	,,
1,500	,,	= 2	,,	,,	,,	,,

CONSTANTS.

No. of Cyls.	Cyclic Irregularity. Per cent.		Phase Alteration in *Electrical* Degrees.	
	'O' Pos.	'O' Neg.	'K' Pos.	'K' Neg.
1	$+ 4{\cdot}35 \times 10^4$	$- 7{\cdot}85 \times 10^4$	$+ 3{\cdot}42 \times 10^4$	$- 5{\cdot}5 \times 10^4$
2	$+ 6{\cdot}62 \times 10^4$	$- 7{\cdot}36 \times 10^4$	$+ 5{\cdot}5 \times 10^4$	$- 5{\cdot}75 \times 10^4$
3	$+ 3{\cdot}75 \times 10^4$	$- 5{\cdot}7 \times 10^4$	$+ 1{\cdot}65 \times 10^4$	$- 1{\cdot}9 \times 10^4$
4	$+ 1{\cdot}08 \times 10^4$	$- 1{\cdot}43 \times 10^4$	$+ 0{\cdot}23 \times 10^4$	$- 0{\cdot}38 \times 10^4$
5	$+ 2{\cdot}8 \times 10^4$	$- 3{\cdot}83 \times 10^4$	$+ 0{\cdot}66 \times 10^4$	$- 0{\cdot}76 \times 10^4$
6	$+ 1{\cdot}88 \times 10^4$	$- 3{\cdot}25 \times 10^4$	$+ 0{\cdot}35 \times 10^4$	$- 0{\cdot}41 \times 10^4$

The favourable figures for four-cylinders are due to the fact that the reciprocating inertia of *all* the pistons is available to reduce the negative work on *each* compression and absorb positive work on each explosion. For moderate speeds an allowance of $4{\cdot}0 \times 10^4$ *total* cyclic variation should be made.

Water Pump.

The volume delivered on radiator or heat interchanger jobs should not be less than 15 gals. per b.h.p. per hour. This will give a temperature rise of from 12° to 18° F. according to the type and design of the engine. The flow should be definitely directed to the parts of the head—exhaust valve seats, etc.—where heat flow is greatest, and the injector or sparking plug boss should also be well cooled.

From actual measurements on small pumps with cast vanes, the volume delivered against $0{\cdot}8\ h$ head corresponds approximately with v_r (radial velocity at eye) $= v_e \tan \frac{\alpha}{2}$ (see fig. 17) when v_e = tangential velocity of the eye, while the stalled pressure is :—

$$\frac{v_w^2}{2g} = h \text{ feet head (1 ft.} = 0{\cdot}434 \text{ lb./sq. in.)}$$

v_r is the radial velocity and v_t the tangental velocity at the vane tip.

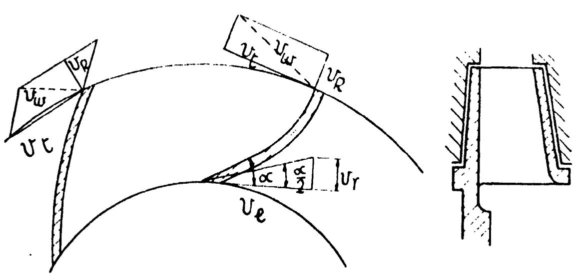


FIG. 17.

The right-hand parallelogram of velocities is drawn for a radial end to the vane, and the left hand for a backward sloping vane. The difference in the maximum water velocity, and therefore the pressure, is apparent. Larger pumps should be dealt with more faithfully by hydraulic engineers.

Oil Pumps.

These may be of many forms, but the most favoured is the gear pump, which when made with the special tooth forms developed for this work can deliver very large quantities from a small machine. As few as 7 teeth may be used, very deep in proportion to the diameter. If ordinary gears are used they should not be fully meshed, but set so that there is a little tangential play,

thus providing a vent at the centre, where otherwise severe 'lock-up' pressures will be generated forcing the gears apart.

The volume delivered per revolution is from 0·55 to 0·6 of the volume of one annulus between bottom and top of the teeth. Obviously the coarser the pitch the greater the volume from the same size pump.

From 2 to 2·5 gals. per hour should be allowed for every b.h.p. for small and medium high-speed engines.

For very large engines less oil is needed per b.h.p. for the bearings, etc., but for piston cooling about 1 gal. per hour per b.h.p. should be available. The specific heat of lubricating oil is 0·5 compared with water, and its specific gravity about 0·95. The temperature rise is therefore about twice that of water under similar conditions.

Sizes of Filters for Engines and Compressors.

To use the following table the filter manufacturer needs to have a table of the C.F.M. which his own range of filters will pass. This formula then tells him the size suitable for the engine or compressor in question.

$$\text{C.F.M.} = \frac{\text{L.A.N.}}{\text{C.}}$$

Where L = Length of stroke in inches.

A = Area of one (L.P.) cylinder in square inches.

N = R.P.M.

C = Constant (from accompanying table)

Values of C.

Impulses per Revolution of Crankshaft.	Type of Engine or Compressor.	Constant.
½	1 Cylinder engine four-stroke cycle	864
[illegible]	2 " " " " "	864
[illegible]	3 " " " " "	864
	4 " " " " "	864
	5 " " " " "	576
	6 " " " " "	576
	7 " " " " "	432
	8 " " " " "	432
*	12 " " " " "	288
†	16 " " " " "	216
‡§	1 Cylinder single acting single stage compressor . .	864
	2 " " " " " " . . .	864
	3 " " " " " " . . .	576
	4 " " " " " " . . .	432
	6 " " " " " " . . .	288
‡ 2	1 Cylinder double acting single stage compressor . .	864
2	2 " " " " " " . .	432
2	3 " " " " " " . .	288
[illegible]	2 Cylinder single acting two-stage compressor . .	864
½	4 " " " " " . .	864
[illegible]	6 " " " " " . .	576
	2 Cylinder double acting two-stage compressor . .	864
	4 " " " " " . .	432
	6 " " " " " . .	288

(*Vokes Ltd.*)

* The equivalent of 2 banks of 6 cylinders at 60°.
† The equivalent of 2 banks of 8 cylinders at 45°.
‡ or { Two-stage (tandem) compressor.
§ { Two-stroke cycle engine.

Torsional Vibrations of Crankshafts.

This is a very wide subject and it will not be possible to do more than touch on the major points.

An elastic structure is one which will return to its original shape if a force which has distorted it is removed. If the force be too great it may cause permanent distortion or rupture at once, but we are here concerned with forces of a smaller magnitude, easily withstood by the body under consideration if applied steadily. When the disturbing force is removed the restoring force or 'spring' of the material under stress brings the parts back to normal, but with this important difference, that they are now moving at a considerable velocity because the potential energy of the stressed material has been converted into kinetic energy of the moving masses.

These continue to move in the opposite direction until they are gradually brought to rest by the stress produced in the reverse sense as the body is distorted the other way, and the parts oscillate backwards and forwards at a speed depending on the inertia and stiffness of the structure. This speed is known as the 'Natural Frequency' and is independent of amplitude, *i.e.* it is the same for large or small swings of the one structure. This process would continue indefinitely were it not for the fact that some energy is dissipated at each oscillation by the parts of the structure moving in a viscous medium, air, water or oil, and also by the internal friction of the particles of the metal over each other (this latter effect is known as 'Hysteresis').

A harmonic force is one which varies as the projection on the diameter of a point moving round a circle at uniform angular velocity.

Its graph on a time base is a sine curve forming waves, equally above and below the zero line, and *the force is proportional to its distance from that line.* This is the essential property of a harmonic force and is very important, since the force required to deflect an elastic structure also varies with the deflection (up to the elastic limit of the material), and this agreement accounts for synchronous vibrations.

When an elastic structure is subjected to a recurring alternating force of this character *at the same frequency* as the natural oscillating frequency of the structure, the disturbing force is always in the same direction as the motion and causes the swings to become larger and larger every time until rupture occurs or the loss of energy at each cycle is equal to the energy put on at each cycle by the applied harmonic force. Take the sample case of a steel shaft fixed to an infinite mass at one end, and to a flywheel at the other.

Let the stiffness of the shaft be such that it requires S ft. lbs. to twist it through 1 radian, while the mass of the flywheel is M gravitational lbs. $\left(M = \frac{\text{Weight lbs.}}{32 \cdot 2}\right)$, and its radius of gyration R ft.

Then the polar inertia of the flywheel is MR^2, and the time for 1 complete oscillation is:—

$$\text{secs.} = 2\pi\sqrt{\frac{MR^2}{S}}$$

i.e. the frequency (vibrations per sec.) $= F = \frac{1}{2\pi}\sqrt{\frac{S}{MR^2}}$

In order to form a mental picture of what happens when a crankshaft is subjected to varying recurring torques, consider first the moving coil of a loud speaker. This can only be in one place at one instant, *i.e.* its movement on a time base is a single line, yet a full orchestra, with all the widely varying frequencies of vibration, is reproduced, and the ear can pick out the different instruments.

If several different harmonic curves are drawn and then combined, a peculiarly shaped wave is produced which repeats itself at intervals—the least common multiple of all the vibration times (see fig. 18).

The reverse process, though much more difficult, is also possible and, as pointed out by Fourier, any recurring wave form, however irregular, may be resolved into a fixed quantity (its mean height) and a number of simple harmonic waves, the largest of which has one pitch in the time required for the irregular wave to complete its cycle. Any simple multiples may be present in different amplitudes and phases. The ear analyses the exceedingly complicated movements of the loud speaker by means of numerous filaments, each one of which only responds to its own frequency.

In the same way a shaft of a certain natural frequency only responds to vibrations of the same frequency as its own, but these may be contained in a complex torque curve in such a way as not to be recognised easily. Since the torque curve repeats every cycle for each crank, the shaft is subjected *at each crank* to a number of concealed harmonics of every multiple of the cycle from one upwards, *i.e.* every multiple and half multiple of the revolutions in four-cycle engines.

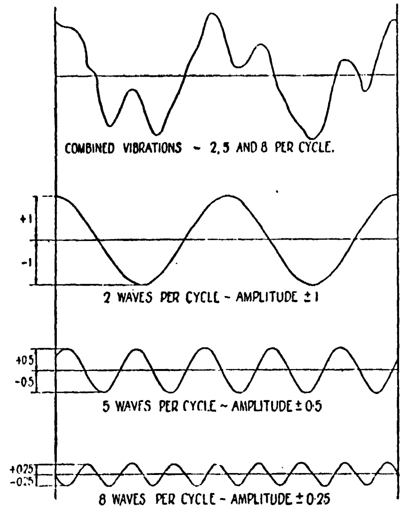


FIG. 18.

The approximate values of the amplitudes of all these harmonics from ½ to 12 are given in the accompanying table. Full torque is assumed, and the values are in pounds per square inch on the piston, so that to find the actual torque in lbs. inches, multiply the piston area in square inches by the crank radius in inches by the suitable figure from the tables. The number of vibrations per revolution is called the 'order.' When the engine speed multiplied by any integer or half integer equals the natural frequency of the crankshaft system, vibration will be present in some degree depending on the extent to which the impulses from the different cranks are in phase with each other. If two equal impulses are a half-pitch apart, they will cancel, and no perceptible vibration will result. If, however, they are not exactly opposed, some fraction of the impulses remains operative, and the simplest way to find the value of the unopposed forcing torque is as follows:—

If several sine waves of the same frequency but with their crests at different places (*i.e.* not in the same phase) are drawn, the resultant wave will be of the same frequency, and may be found by adding a number of ordinates, but this is a slow and cumbersome method, and a much better way is to draw a 'vector diagram.' In this method each harmonic forcing torque is represented by a vector or radial line from a common centre, the *length* of the line being the half amplitude, and the *angle* the phase of each individual sine wave.

The resultant, found either geometrically or by simple trigonometrical calculation, as for a number of forces, will again be a radial line and will represent by its length and angle the amplitude and phase of the resulting vibration. One circle represents the time for a complete vibration, and therefore the angle on the vector diagram between two harmonic forces associated with two cranks on the engine will be the angle between the firing points of those cranks multiplied by the 'order,' *i.e.* the number of vibrations per revolution of the crankshaft.

Now imagine a shaft (with six cranks and one flywheel) of such stiffness that when vibrating about one node (stationary point) the different cranks move through the amplitudes shown on the curve (fig. 19). This curve is called the 'elastic' curve, and the method of plotting it will be described.) Then the vibratory energy input at each crank will be in proportion to the ordinate of the elastic curve at that crank.

Now draw a 'vector diagram' for each order, in which one revolution equals the time for one *vibration* and each crank is shown in its proper firing sequence by a vector or radial line at the *angle* corresponding to the crank angle multiplied by the 'order' of vibration under investigation, and of a *length* equal to its ordinate on the elastic curve. It will be found that several orders have the same diagram, *e.g.* 3, 6, 9, — 1½, 4½, 7½, etc., for a six-cylinder engine and then the only difference is in the value of the applied harmonic torque for each order (see table).

It will be seen that in some orders (in this case 1½, 4½, 7½ and 2½, 3½, 5½, 6½) an alteration in the firing sequence makes a considerable difference in the result, and sometimes this alteration alone is a complete solution to the problem.

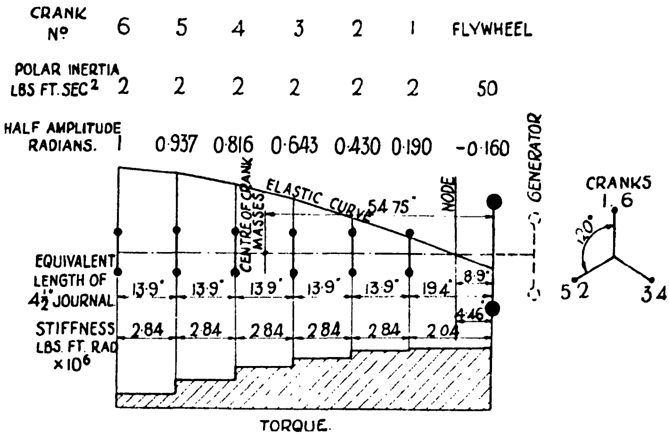


FIG. 19.

In eight-cylinder line engines several firing sequences are possible with the normal crankshaft and all should be investigated before deciding. A quick approximation may be made in simple

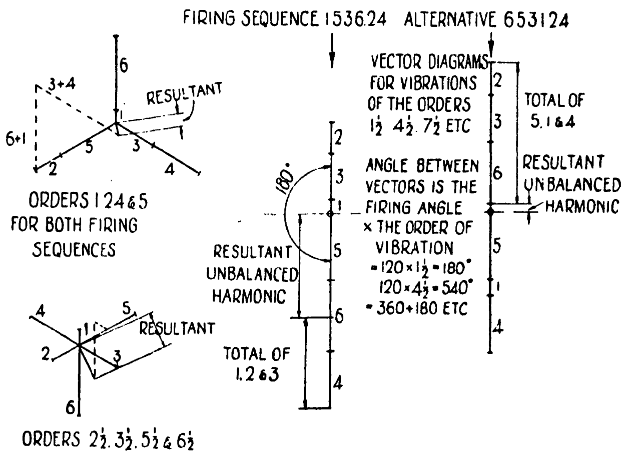


FIG. 20.

cases by numbering the cylinders from the flywheel end and reckoning the cylinder number as equal to the amplitude.

Thus for an eight cylinder running between the $5\frac{1}{2}$ and $6\frac{1}{2}$ orders the question of firing 1, 6, 2, 5, 8, 3, 7, 4, or 1, 3, 2, 5, 8, 6, 7, 4, may be settled by a rudimentary sketch.

Firing angle, 90°. Vector angles $90° \times 5\frac{1}{2}$ and $6\frac{1}{2} = 360° + 135°$ and $360° + 225°$
The diagrams for the two vibrations will therefore be the same, but in reverse directions.

1.6.2.5.8.3.7.4. FIRING SEQUENCE. 1.3.2.5.8.6.7.4.

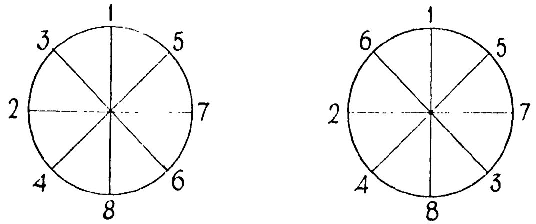


FIG. 21.

The second firing sequence is obviously much better in this case. The 6th order is approximately zero for both.

EFFECT OF RECIPROCATING MASSES.

Quite apart from the *mean* effect of the piston, small end and other reciprocating masses (which can be allowed for by adding half their value to that of the revolving mass with which they are associated) these parts introduce inertia forces which in turn resolve themselves into recurring torques having a number of harmonic components which affect the value of the corresponding gas torque components. The inertia torque components of sufficient value to be considered, are the 1st, 2nd, 3rd and 4th. They are all symmetrically disposed about the top dead centre, *i.e.* zero phase angle, while the gas forces tend to group further forward. Therefore it is impossible to add or subtract them directly, a vector sum is the only solution. Moreover, the gas forces alter slightly in amplitude and phase with I.M.E.P., but the inertia forces are proportional to the square of the angular velocity (or R.P.M.).

The approximate values of these orders are therefore given as sine and cosine values for the gas components and the sine values only for the inertia (the cosine values are zero).

To find the vector sum for a particular order, proceed as follows:—

Work out the weight in pounds of the reciprocating parts and divide by the piston area. This gives the weight per sq. in. Call this W. t_n is the torque coefficient (sine value) for the n^{th} order taken from the table below; g is the gravitational unit 32·2; R is the crank radius in inches and the angular velocity ω is in radians per second. Then the inertia sine component is S_I and equals

$$\frac{t_n \times W \times R^2 \times \omega^2}{12g}$$

Note that t_n may be of *negative* value, thus cancelling part of the gas sine component. Call the gas sine and cosine components S_G and C_G. Then the resultant value T_n of the harmonic component of the n^{th} order for this particular engine is :—

$$T_n = \sqrt{(S_G + S_I)^2 + C_G^2} \text{ lbs. per sq. in. of piston and 1 in. crank radius.}$$

n^{th} HARMONIC TORQUE DUE TO INERTIA OF RECIPROCATING PART.

Con. Rod / Crank Ratio		3	3·5	4	4·2	·4	4·6	4·8
Order No.	Sign.	Harmonic Component t_n (Sine only).						
1	+	0·086	0·073	0·064	0·060	0·058	0·055	0·053
2	—	0·500	0·500	0·500	0·500	0·500	0·500	0·500
3	—	0·261	0·220	0·192	0·182	0·174	0·166	0·159
4	—	0·030	0·022	0·016	0·015	0·013	0·012	0·011

SINE AND COSINE VALUES OF 1ST, 2ND, 3RD AND 4TH ORDER GAS TORQUES, LBS./SQ. IN.

Order No.	Sin. or Cos.	I.M.E.P. of Petrol Engines. (Spark Ignition.)				I.M.E.P. of Diesel Engines (C.I.)	
		30	70	110	150	30	130
1	Sin.	13·65	24·9	37·8	50·6	23·0	42·0
1	Cos	4·25	8·2	13·1	17·65	4·2	16·6
2	Sin	10·70	16·95	25·40	34·0	26·0	38·0
2	Cos.	−1·5	−4·0	−6·45	−8·6	0	−1·25
3	Sin	6·3	8·8	12·9	16·7	20·0	27·7
3	−Cos	−2·25	−4·55	−6·95	−9·05	−1·0	−3·9
4	Sin.	3·1	4·0	5·8	8·0	12·3	16·2
4	Cos.	−2·05	−4·7	−7·55	−10·6	−1·3	−5·1

TABLE OF HARMONIC COMPONENTS OF TORQUE CURVE (T_n).

Pounds per sq. in. of piston (one cylinder) 1 in. radius crank, full load, single-acting engine, all values ±.

Note.—(1) There are no half orders in two-stroke cycle engines.

(2) In double-acting engines the reversed connecting rod effect introduces modifications in the values, but they may be treated as two-cylinder firing at 180° on the same crank, *i.e.* odd orders cancel out, even orders double and half orders are $\sqrt{2}$ times the single-acting engine.

(3) See separate table for sine and cosine values of orders 1, 2, 3 and 4, and the effect of reciprocating masses on these orders.

Order.	Petrol 4-Stroke.	Oil 4-Stroke.	Oil 2-Stroke
½	53	45	
1	53	45	90
1½	47·5	42	
2	35	38	76
2½	26	33	
3	19	28	56
3½	16	23	
4	13·3	19	38
4½	10·7	15	
5	9·3	11	22
5½	8·7	9	
6	5	7·5	15
6½	4	6	
7	3·2	4·8	9·6
7½	2·6	3·9	
8	2·2	3·2	6·4
8½	1·7	2·6	
9	1·4	2·2	4·4
9½	1·2	1·9	
10	1·0	1·6	3·2
10½	0·9	1·4	
11	0·8	1·2	2·4
11½	0·7	1·1	
12	0·6	1·0	2·0

The Elastic Curve.

The first step is to find the stiffness of each section of the system, conveniently expressed as a length of journal of equal stiffness.

To do this from first principles is almost impossible with a complex structure such as a crankshaft, but several empirical formulæ have been suggested.

The simple one worked out by Major Carter gives good agreement with torsion tests on very widely differing proportion of shafts, and is as under:—

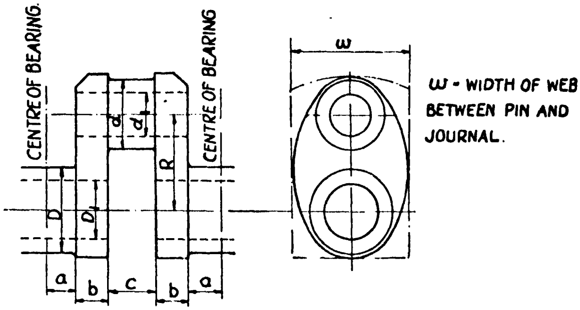


FIG. 22.

$$2a + 0{\cdot}8b + \frac{0{\cdot}75c \times (D^4 - D_1^4)}{d^4 - d_1^4} + \frac{1{\cdot}5R \times (D^4 - D_1^4)}{b \times W^3} = l''$$

i.e. the length of a shaft of diameter D, and bore D_1 (*i.e.* the journal size) equal in stiffness to a crank unit from centre to centre of bearings. Add the necessary amounts at either end from the drawing of the crankshaft, and fill in on the diagram (fig. 19).

The next thing is to fill in the polar inertia of each crank (including the big end and half the reciprocating mass) the flywheel and the auxiliaries, if the latter are driven from the free end. The calculations for one crank mass may be taken in the order given below, the letters as for the crank stiffness:—

Each moving part must be multiplied by the square of its radius of gyration, *i.e.* the radius at which the mass would have to be concentrated to have the same inertia. As the dimensions are almost always given in inches and the weight in pounds, these units will be used for the separate parts, and the total corrected to pounds per feet per second² by dividing by 144*g*., *i.e.* 4630.

Part.	Radius of Gyration.
Piston and Small End (*i.e.* total reciprocating weight).	Allow half the total weight × crank radius² = R^2 = polar inertia lbs. ins.
Big End.	Total weight × crank radius² = R^2 = ,,
Crankpin.	Total weight × $\left(R^2 + \frac{d^2 + d_1^2}{8}\right)$ = ,,
Rectangular Webs.—Total weight, length L, width *w*, centre of gravity C″ from shaft centre.	Weight of 1 pair × $\left(C^2 + \frac{L^2 + w^2}{12}\right)$ = ,,
Eccentric Disc Webs.—d''_w dia., C″ eccentric.	Weight of 1 pair × $\left(C^2 + \frac{d_w^2}{8}\right)$ = ,,
Concentric Disc Webs.—d''_w dia.	Weight of 1 pair × $\frac{d_w^2}{8}$ = ,,
Elliptical Webs.—m'' major axis, m''_1 minor axis, C″ eccentric from shaft centre.	Weight of 1 pair × $\left(C^2 + \frac{m^2 + m_1^2}{16}\right)$ = ,,
Journal	Total weight × $\frac{D^2 + D_1^2}{8}$. = ,,
Balance Weights.—These can generally be reduced to an equivalent sector of a disc or annulus of radius *r* outer, r_1 inner.	Weight × $\frac{r^2 + r_1^2}{2}$ = polar inertia lbs. ins.²

Divide the total pounds weight × ins.² for each crank by 4,630 to give the result in pounds ft. sec.², and insert in shaft diagram (fig. 19).

Flywheel rim outer dia., D_r inner D_d

$$\text{Weight of rim} \times \frac{D_r^2 + D_d^2}{8} = \text{polar inertia lbs. ins.}^2$$

Flywheel disc, dia. D_d

$$\text{Weight of disc} \times \frac{D_d^2}{8} = \quad ,,$$

Now using the inertia values as weights, and treating the equivalent lengths as giving their positions on a beam, find the point of balance—multiplying each inertia figure by its equivalent distance from the flywheel, and dividing the total by the total inertia including the flywheel.

Moment of crank inertias about flywheel (see fig. 19) $= 6 \times 2 \times 54 \cdot 75 = 657$.

Total inertia, 62 lbs. ft. sec.^{2}.

$$\text{Point of balance} = \frac{657}{\text{total inertia}} = 10 \cdot 6 \text{ ins.} = \text{ins. from flywheel.}$$

Then find the frequency of the flywheel oscillating about the point of balance as a stationary point or node, thus :—

$$\text{Shaft stiffness from node to flywheel: } \left\{ \begin{array}{l} \text{dia. of shaft } D \\ \text{bore of shaft } D_1 \\ \text{length from node } L \end{array} \right\}$$

$$= S = \frac{(D^4 - D_1^4)\pi \times 11 \cdot 8 \times 10^6}{32 \times L \times 12} \text{ lbs. ft. per radian}$$

$$= \frac{(D^4 - D_1^4) \times 10^6}{10 \cdot 35 \times L} \text{ lbs. ft. per radian.}$$

Time of oscillation

$$= t \text{ secs.} = 2\pi \sqrt{\frac{I_p}{S}}$$

where I_p = polar inertia of the flywheel $= MR^2 \dfrac{\text{lbs. ft.}^2}{g}$

Frequency $= F = \dfrac{1}{t}$ vibrations per second.

This figure will be too low and should be corrected by dividing by a factor varying with the number of cranks as under :—

1	2	3	4	5	6	7	8	number of cranks.
1·0	0·95	0·94	0·93	0·92	0·92	0·91	0·91	divisor.

Stiffness of 10·6 ins. of $4\frac{1}{2}$ shafts $= 3 \cdot 73 \times 10^6$.

$$\text{Frequency} = 2\pi \sqrt{\frac{3 \cdot 73 \times 10^6}{50}} = 43 \cdot 8. \quad \text{Corrected} = \frac{43 \cdot 8}{92} = 47 \cdot 5 \text{ per sec.}$$

In many cases this is sufficiently accurate and the frequency may be found to be too high to be dangerous (say more than six times the r.p.m. in a six cylinder), but for an accurate determination of the frequency and the value of the ordinates of the elastic curve, proceed as follows :—

Let F vibrations per second be the approximate frequency as found by the method just explained. Then the harmonic torque in pounds feet, required to oscillate a mass of polar inertia 1 lb. ft. sec.^{2} through an amplitude of $\pm$ 1 radian is $4\pi^2 F^2 = 39 \cdot 4 \ F^2$. Starting from the end opposite the flywheel where the movement will obviously be greatest, assume a half amplitude of 1 radian at the end crank and make a table as in the worked example below.

Assumed frequency 47·5 per sec. Acceleration torque per lbs. ft. sec.^{2}

$$39 \cdot 4 \times 47 \cdot 5^2 = 89 \times 10^3 \text{ lbs. ft.}$$

1.	2.	3.	4.	5.	6.	7.	8.
Mass.	Inertia Lbs. Ft. Sec.^{2}	Half Amplitude $\pm \theta$ Radians.	Acceleratn. Torque T_A Lbs. Ft. $\times 10^3$.	Total Torque T_T Lbs. Ft. $\times 10^3$.	Stiffness of Section Lbs. Ft. Rad. $\times 10^6$.	Deflection of Section ϕ Radian.	Remaining Angle $\theta - \phi$.
Crank No. 6	2	1	178	178	2·84	0·063	0·937
,, 5	2	0·937	167	345	2·84	0·121	0·816
,, 4	2	0·816	145	490	2·84	0·173	0·643
,, 3	2	0·643	115	605	2·84	0·213	0·430
,, 2	2	0·430	76	681	2·84	0·240	0·190
,, 1	2	0·190	34	715	2·04	0·350	0·160
Flywheel	50	0·160	715	715	4·46 Between flywheel and node = 8·9″ of $4\frac{1}{2}$ shaft.	0 160	—

Column 4 is the product of the unit acceleration torque (89×10^3 for the assumed frequency of 47·5 per sec.) by the inertia (column 2) by the half amplitude (column 3).

Column 5 is the addition of the acceleration torque in column 4 and the previous entry in column 5.

Column 6 is from the shaft diagram.

Column 7 is column 5 divided by column 6.

Column 8 is column 3 minus column 7.

The *stiffness* to the node is found by dividing the torque in that section by the half amplitude of the flywheel movement:—

$$\frac{715 \times 10^3}{0 \cdot 16} = 4 \cdot 46 \times 10^6.$$

The *distance* from flywheel to node is given by dividing the stiffness per inch of the shaft by the actual stiffness between flywheel and node:—

$$\frac{39 \cdot 5 \times 10^6}{4 \cdot 46 \times 10^6} = 8 \cdot 9 \text{ ins.}$$

If the last section does not balance, *i.e.* if the flywheel acceleration torque at the calculated amplitude is greater or less than the total torque of the cranks, then the assumed frequency is also greater or less than the true figure, and a second approximation must be made. When practical agreement is reached, the figures in column 3 are the ordinates of the elastic curves for the mode of vibration under consideration, *i.e.* one node in the crankshaft, and column 5 gives the total periodic torque in each section, from which the stress is readily calculated.

Note.—The mean useful torque should be added to the periodic torque when calculating the maximum stress.

The Resultant from the Vector Diagram for any order of vibration n is called the Vectorial Sum, and is written $\Sigma\theta$. When multiplied by the value from the table of Harmonic Components T_n (for the order under consideration) and by the piston area square inches and crank radius inches, the product is the harmonic torque in lbs. ins. for that order.

$$\Sigma\theta \times T_n \times A \times R = T_h.$$

and the energy input for $\pm$ 1 radian movement at the free end of the shaft is:—

$$\pi T_h \text{ lbs. ins. per cycle.}$$

Damping.

This is very indeterminate, and no hard and fast rules can apply. Internal energy loss or hysteresis of the material, piston friction, bearing friction, air resistance, deflections of the engine frame, propeller effect, all tend to damp out synchronous vibrations. However, it is safe to assume that the amplitude of movement for hysteresis damping alone will be at least twice the actual deflection.

The hysteresis energy in each section of shaft depends on the stress, f lbs. sq. inch, radius r ins. and length l ins. of the section, and is approximately as under—

$$f^{2 \cdot 3} \times 2r^3 \times l \times 10^{-10} \text{ lbs. ins.}$$

For hollow shafts use $2 \times \dfrac{r^{4 \cdot 3} - r_1^{\,4 \cdot 3}}{r^{1 \cdot 3}}$ instead of $2r^3$.

Work out the value for each section and find the total H lbs. ins. per cycle for the shaft with $\pm$ 1 radian movement at the free end.

The amplitude for hysteresis damping only is then—

$$\pm\, a = \sqrt[1 \cdot 3]{\frac{\pi T_h}{H}} \text{ radians}$$

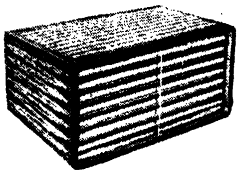
and from experimental evidence this may be halved to give the actual amplitude to be expected.

The stress in the various sections will be that worked out for 1 radian (fig. 19) multiplied by $\frac{a}{2}$.

Generator Sets.

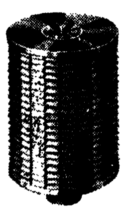
When a generator is directly coupled to the flywheel with no appreciable length of shaft between, the generator inertia may be added to that of the flywheel and the two considered as one body.

Usually, however, the shaft has sufficient length to make a considerable difference in the natural frequency of the system and its equivalent length and stiffness should then be calculated and (with its polar inertia) added as an extension of fig. 19, the new point of balance being reckoned in the same way by moments about the generator in this case.

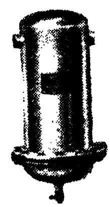


KOMPAK

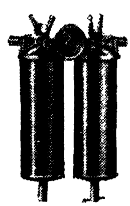
LUBRICATING OIL



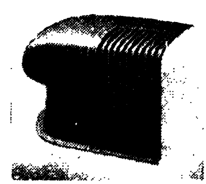
MICRO-VEE



PIPE LINE



TWIN FUEL



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When a very long flexible shaft connects the engine to a mass, such as a propeller, two frequencies are to be dealth with.

The two-node frequency will be practically the same as that for the bare engine and flywheel, with one node in the crankshaft near the flywheel, and the other near the propeller.

The periodic torque in this condition of vibration is high in the crankshaft and low in the propeller shaft, which need not therefore be considered on this count. There will, however, be a much lower frequency due to the whole engine system, cranks and flywheel oscillating against the mass of the propeller and its entrained water.

For long propeller shafts regard the crank masses as added to the flywheel.

Then if the engine polar inertia is I_e and the propeller with the entrained water is I_p while the length of the shaft reduced to a uniform diameter is l ins., then the node will be l_e from the engine and l_p from the propeller, when

$$l_e + l_p = l$$

and

$$I_e l_e = I_p l_p$$

hence

$$\frac{I_e l}{I_e - I_p} = p.$$

From this find the frequency, using either the propeller inertia or the total engine inertia, and the stiffness of the shaft between the mass and the node.

The elastic curve (neglecting the inertia of the shaft which is small) is a straight line through the node, and the amplitudes are inversely as the inertias.

For this form of vibration (one node in the whole system) the amplitude of movement of all the engine cranks is substantially equal. All the minor orders therefore cancel and only the major orders are troublesome.

Major orders are those in which all the vectors are in phase and additive.

If there are n cylinders, then all orders which are a multiple of $\frac{n}{2}$ are major orders in 4-stroke engines, and multiples of n in two strokes.

Vee Engines.

Find the resultant or vectorial sum $\Sigma\theta_A$ for one bank and then the resultant for the whole engine, by combining two of these at the angle, separating the *firing positions* of the two banks. Note that this is not usually the vee angle, but 360° more on the engine, *e.g.* in a 60° vee, 16-cylinder, B.1 does not fire 60° after cylinder A.1, but 360° + 60° = 420°. This does not make any difference on integral orders, but on the half orders it is very important.

Take the $5\frac{1}{2}$ order with firing sequence on each bank, 1, 6, 2, 5, 8, 3, 7, 4. The vectorial sum for A bank will be approximately 1·45 for 1 radian at the free end. The vector for B bank will be at 420° × $5\frac{1}{2}$ = 2310° to A, *i.e.* (6 × 360°) + 150°, *i.e.* nearly opposite, so that the final resultant is quite small, 0·74 approximately. Whereas, if 60° (the vee angle) is taken, the result is 60° × $5\frac{1}{2}$ = 330°, which means that the vectors for the two banks are at 30° to each other, and the resultant is almost the arithmetic sum of the two—actually 2·8.

The most serious cases of torsional vibration are usually those involving one node in the crankshaft itself.

In general, orders of higher number than the 6th are not destructive, but may be very objectionable and intolerable with gearing. Modern large diameter crankshafts will stand running through the 6th and $4\frac{1}{2}$ orders in 6- and 12-cylinder engines, but must not be held on them.

Flexible Couplings.

Great care must be taken in choosing and specifying these. The coupling must be as strong in torsion as the shaft and must have the specified stiffness. This must be such that when the coupling is represented in the diagram of the system as a shaft of equivalent stiffness, the frequency of the whole plant is favourably placed with regard to the running range of speed.

The construction of the flexible coupling must be such that its torque/torsion diagram departs considerably from a straight line, so that with increasing amplitude the natural frequency rises and puts the system out of tune with the exciting harmonic.

Dampers.

These are, broadly, of two kinds:—

(*a*) Those which absorb work when vibration occurs, and convert it into heat by solid or viscous friction.

(*b*) Those which put the system out of tune by altering the frequency at short intervals or when vibration occurs.

The first type consist of a fairly heavy rim frictionally attached to a light driving plate, so that when alternating torque occurs it will slip, and so absorb energy and limit the building up of vibration. The driving plate must be very rigidly attached to the shaft to stand alternating loads without developing play, and the friction torque should be enough to hold the rim without slip during the ordinary cyclical variations of the engine, otherwise the device will wear rapidly.

It must not be used at a steady speed at which vibration occurs. Some modification in mass or stiffness should be made to alter the vibratory speed in that case. A modification popular in car work is to attach the rim to the driving plate by a thick pad of rubber vulcanised to both. The very high hysteresis of the rubber then absorbs the vibratory energy, and the device, when not overloaded, runs without lubrication or attention.

A further variation uses oil trapped in pockets between the vanes of the inner and outer members, small channels allowing relative motion and the viscous friction dissipates the vibratory energy.

The second class does not really damp out the vibration, but alters the frequency, so that the system no longer responds to the exciting force. This is fundamentally a much sounder scheme.

One form consists of inner and outer members, the latter of considerable inertia, and driven by oil between fitting radial vanes. By means of a stationary valve about which the device rotates, the oil is alternately locked in the spaces or allowed to flow from one to another, so that in effect the crankshaft has a large mass attached to its free end at one instant, and only the light inner mass the next. This occurs several times in each revolution. The frequency is, therefore, constantly varying and no building up can occur.

Another form is the adaptation of a well-known flexible coupling, the light member driving the free heavy mass by means of 'gridiron' springs, arranged in curved slots, so shaped that the frequency rises rapidly with increasing amplitude. This does not produce heat and appears to last without attention for long periods.

The friction types are only palliatives and cannot be recommended for large plants, where steps must be taken in the design stage to arrange that the various frequencies (one, two or even three node) shall not produce heavy stresses in the running range.

Measurement of Actual Stresses.

Torsiographs of several makes are available which consist essentially of two parts, first, a very light pulley driven from the engine shaft under examination and following all its speed fluctuations; secondly, a heavy flywheel driven from the light pulley by a very soft spring, so that it maintains a steady smooth speed sensibly constant over short periods and the mean of the engine speed.

The differences in angular position of these parts show the fluctuations of the shaft. A light rigid pen mechanism draws a graph of the variation on a moving strip of paper in the Geiger machine, while a stylus draws a similar diagram on a celluloid band in the Cambridge instrument.

The drive must be from an antinode where movement is greatest, and for vibration over 3,000, a steel belt must be used.

Where possible a direct drive through a short tubular cardan shaft, flexible in bending but rigid in torsion, is recommended, especially for vibrations of 4,000 per minute and upwards. The graph drawn by the instrument, multiplied by suitable constants, represents the twist of the shaft at the driving point, taking the node as stationary. A second pen or stylus marks time intervals and a third marks the engine cycles on the same strip. The frequency and order of the vibrations may therefore be checked.

It is to be noted that the record from the instrument is in itself no use in calculating stresses. The necessary shaft diagram of stiffness, etc., and the table of torques in the several sections, must be done first in order to interpret the graph.

High-Speed Electrical Torsiographs.

As rotational speeds increase and crankshaft stiffnesses are put up, it is found that in spite of every refinement—small overall size, light pen mechanisms, restricted travel (afterwards photographically magnified), etc., the various forms of mechanical torsiograph fail to give correct amplitude readings and something fundamental must be done to record the real movements of the shaft.

One device consists of a simple electro-magnetic machine of very small size in which the armature is firmly attached to the shaft, while the field magnets revolve at a sensibly uniform speed. Multiple brushes (to ensure continuous contact) running on slip rings, pick up the varying voltage due to the constantly changing relative velocity of the field and armature. This is fed to a valve amplifier and, after integrating by a suitable electrical circuit to give displacement, is shown on a cathode ray tube in which the vertical component is the shaft movement and the horizontal component is controlled by a 'time sweep' which by suitable means may be divided into crankshaft degrees, and either read directly or photographed for record.

To assist in interpreting the somewhat complicated diagram 'harmonic analysers' have been devised, by means of which each component may be separated from the rest, using a resonant circuit which may be tuned over a very wide range of frequencies.

On account of the very small potentials generated, possible brush losses, etc., other methods are now being investigated, for example—photo electric pick-ups which do away with sliding

contacts and give a much stronger 'signal' needing less amplification and reading directly in displacement without the intervention of an 'integrator,' which, particularly at low frequencies, is a possible source of error and better eliminated where possible.

Rapid strides are being made and a choice of really good pick-ups should soon be available. Reliable frequency analysers are already obtainable, and reduce enormously the labour of interpreting the diagrams. It should be emphasised that the greatest *apparent* amplitude measured from the unfiltered diagram is seldom the amplitude of the fundamental vibration. This only occurs when a slightly damped system is running right on a violent synchronous vibratory speed, showing almost perfect sine waves without any interference. In all other cases, at least a rudimentary harmonic analysis must be made, and this tedious process is done quickly and thoroughly by the electrical analyser.

For important and complicated cases the reader is advised to look up the more extended works devoted solely to this subject, or to obtain expert advice.

Scavenging and Supercharging.

With the great increase in the number of large two-stroke engines for marine work and the practically universal adoption of super-charging or 'blowing' for aero engines, these subjects have become more important.

Timing and port area for two-strokes have been touched on in Part III, and it will be evident that as speeds rise, it becomes important to hold a nice balance between the power lost by a somewhat exaggerated timing (exhaust opening very early so that it may get away and allow time for the scavenge operation) and the power expended on the fan or blower, since the pressure required varies inversely as the square of the time for equal ports. In general, the port area will also be smaller if the period is smaller, so that a careful preliminary survey is essen-

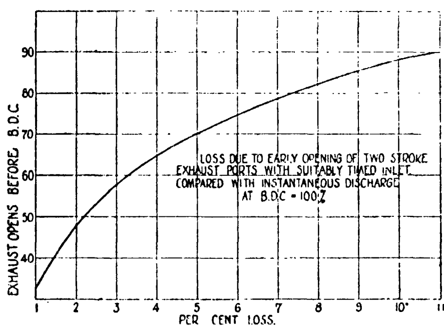


FIG. 23.

tial. The graph, fig. 23, gives the approximate value of the loss due to exhaust release, and will help in deciding the timing. Fig. 24 gives the approximate power for the blower.

Blowers for marine work are often of the rotary displacement pattern. These, though noisy, are efficient when working at the designed pressure, and are particularly suitable for the steady speed at which these engines usually run.

In some forms the volume enclosed between the vanes reduces before the delivery port opens, thus compressing the air to delivery pressure and avoiding back surge through the ports. This makes for a very high efficiency at the designed pressure.

The simple 'Rootes' type is common when driven from its own engine, for direct reversing marine plants, the air inlet and outlet being interchanged by valves connected to the reversing control, so that the air goes into the cylinder whichever way the blower runs. This avoids reversing gear.

The gear-driven centrifugal blower has an inherent advantage for two-stroke variable speed engines, in that, while the volume delivered per minute varies in direct proportion to the speed, the pressure varies as the square of the speed, which is just what is required for forcing the air through the ports. The power expended is therefore in proportion to the cube of the speed, but the output follows the engine requirements in volume and pressure over a wide speed range. The adiabatic efficiency, however, is not so good anywhere as that of some displacement blowers at their designed pressure.

Exhaust turbine-driven blowers are, of course, centrifugal fans, and run up to a very high speed. This form is found on four-cycle engines rather than on two-strokes, since back pressure on the exhaust of the latter is not popular, but there does not seem to be any fundamental reason for the discrimination. The percentage of surplus air, above that needed for complete chemical balance, should be greater if an exhaust turbine is used, so that the mean exhaust temperature will be lowered by distributing the heat over a greater weight of air, thus making conditions easier for the turbine blades, which otherwise may fail. This entails a greater 'port area × time' integral and, in the four-stroke, a very large overlap between inlet opening and exhaust closing—as far as 160° total has been employed.

Supercharging obtains to a slight extent on two-strokes, but usually as a result of the scavenge pressure required to displace the residual gases in the available time, so that, when the cylinder ports close, the charge is well above atmospheric pressure. By that time the piston has, however, reached a point in the stroke which reduces the enclosed volume so much that there is no appreciable gain in the weight of air charge compared with a full cylinder volume at atmospheric pressure. Supercharging proper may therefore be thought of as applying to the four-cycle engine.

HORSE-POWER, ETC., FOR SUPERCHARGING FOR 100 CU. FT. PER MIN. AT 30 IN. MERCURY AND 15° C. INLET.

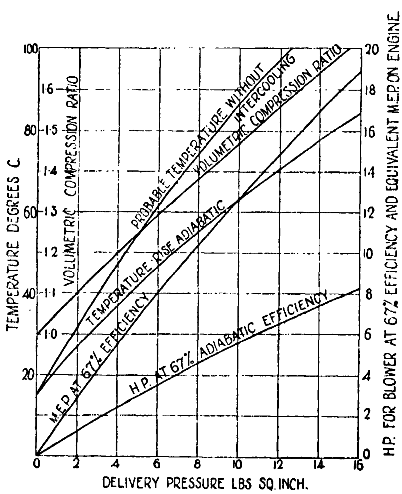


FIG. 24.

The practical limit to the boost pressure is not yet reached, as experimental work has been done up to approximately 4-atmospheres absolute with brake mean pressures of 530 lbs. per sq. in. referred to the whole stroke. This figure does not allow for the work of compressing the air, which was supplied from an outside source, but illustrates what can be got out of an engine. This work was done in a sleeve valve engine, as the absence of the hot exhaust valve gave it a great advantage, and the only trouble experienced was with plugs in the early stages. This was overcome. Incidentally, the exhaust pipes burnt out until made of 'Staybrite' steel.

In aero work 3 to 4½ lbs. above atmosphere at ground level is common, while for 'stunts' or high altitude work, blowers have been used giving over twice the external pressure.

Attention is now being paid to two-stage blowers with some inter-cooling, and to two-speed centrifugals. Either system is intended to give a greater altitude range by maintaining the ground level power at a lower external pressure than is possible with a single-stage or single-speed fan. The measured mean temperature rise for moderate pressure superchargers is 4° C. per 1 in. of mercury (Hg), *i.e.* approximately 8° C. per 1 lb. sq. in. gauge pressure with blower inlet at 30 in. Hg (atmospheric pressure). The more efficient the machine, the less the temperature rise. A curve is given below showing the effect of inlet temperature on power (fig. 25).

In a four-cycle engine the supercharge or 'boost' pressure helps the piston on the downward induction stroke, although some portion of the pressure is used up over the greater part of this

stroke in pushing the greater weight of charge through the valve port. The remainder, however, does useful work on the shaft, and in effect reduces the blower **loss. As a rough estimate, 0·6 of** the pressure is useful. **In comparing** multi-cylinder engines **with single cylinder experimental** sets, it has been observed that the latter with separately driven blowers at 4 to 5 lbs. gauge pressure give results almost identical with the multi-cylinder driving its own blower—the difference in the mechanical efficiency being sufficient to supply the power for the fan.

The permissible compression ratio (R) is reduced as the boost is raised—because the higher induction temperature leads to a still higher compression temperature in the ratio of $R^{0\cdot 35}$ approximately; *e.g.* at 7 to 1 ratio every 1° rise of the charge at B.D.O. becomes 2° when

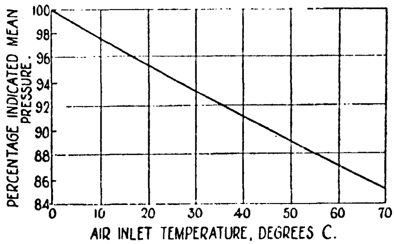


FIG. 25.

compressed. The actual ratios depend to some extent on the engine speed, but mostly on the fuel used, which recently has altered and is still altering rapidly in detonation quality.

Intercooling of the boosted air, when possible, is of value therefore, as it permits the compression ratio to be raised with gain in power and efficiency.

Altitude.

The pressure, temperature and density at any given altitude vary considerably in different places and at different seasons, but mean values have been agreed for what is known as the 'International Standard Atmosphere.' These are given in the graph printed below (fig. 26).

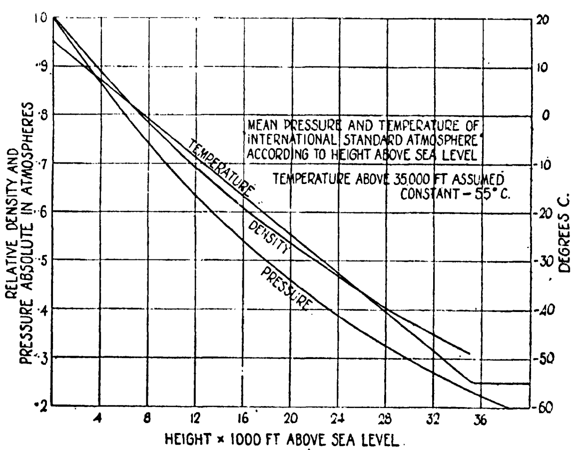


FIG. 26.

Mountings—Foundations—Maintenance, etc.—Standard Specifications—B.S.I. and Lloyd's Rules.

It is customary to put in large masses of concrete to which the engine is bolted by heavy holding-down bolts.

While the addition of this inertia to the engine reduces the amplitude of movement due to torque reaction, unbalanced forces and couples, or engine bed deflections, the rigidity of the arrangement causes vibration waves to be transmitted to the soil, and they may be felt at remarkable distances and have often been the cause of litigation and considerable expense. For some time now there has been a tendency to apply the lessons of the car and aero engines (which run on the flimsiest supports) to heavier work, and foundations now often take the form of a reinforced concrete raft or slab sufficiently deep to form a strong stiff bed for the engine, and also for the generator if close coupled, the whole unit being suspended elastically on a series of springs, or on a generous thickness of some resilient material, so that the bed may move through its vibratory amplitude without any appreciable alteration in the force exerted on the surrounding soil or floor.

Springs should be roughly in two lines parallel to the crankshaft, and as far as possible from it so as to give good resistance to torque reaction (see later paragraph) while being soft (*i.e.* of low rate lbs. per inch) vertically.

If the weight of the whole plant deflects the springs d inches, then the vertical oscillatory frequency of the system will be :—

$$\text{F/min.} = \frac{188}{\sqrt{d \text{ ins.}}} \text{ vibrations per min.}$$

This should be well below the lowest running speed in r.p.m., so as to be clear of vibrations set up by the primary forces on the pistons, etc.

Special felts and rubber in various forms are also used, but in these cases it is most important that an *adequate thickness* should be used—a mere layer is practically useless. The material must have sufficient depth in the direction of vibration to absorb all the probable engine movement and return to its normal dimension continuously, so that the pressure on the ground is almost constant.

In car work the secondary vibrations of four-cylinder engines are now catered for by spring or, more usually, rubber mountings.

Torque reaction is also absorbed to some extent, but as this increases with *decrease* in speed, the mounting is likely to tune in at some low speed near the desired idling speed. It is therefore advisable to make an approximate estimate of the polar inertia of the engine body (cylinders, crankcase and all fixed parts) and see that the stiffness of the rubber or spring mounting is sufficiently low to respond only to impulses below those at the desired idling speed.

If the engine weight exclusive of crankshaft and flywheel is W lbs., and the radius of gyration is r ft., then the polar inertia is :—

$$I_p = \frac{Wr^2}{g}$$

If the supporting springs have a total rate of R lbs. per in. deflection at a distance of r_1 ft. from the centre of the engine, the stiffness of the mounting to resist torque is :—

$$S = 12Rr_1^2 \text{ lbs. ft. per radian,}$$

and the frequency of the whole system oscillating round the shaft is :—

$$\text{F/sec.} = \frac{1}{2\pi}\sqrt{\frac{S}{I_p}} \text{ vibrations per sec.}$$

This frequency should be less than the number of firing impulses per sec. at the idling or lowest running speed, *i.e.* for n cylinders in a four-stroke engine—

$$\frac{n}{2} \times \frac{\text{R.P.M.}}{60}$$

In designing the springs, first choose a suitable diameter of wire and radius of coil to give a reasonable stress in the wire (say under 30-tons/sq. in.). Then make the number of coils sufficient for the deflection (see calculation for Valve Springs, page 223.).

Pads of felt may be arranged just clear of the supporting raft in its normal running position, to check the swing which may occur when stopping and starting.

Attention must be paid to all pipe connections, etc., and flexible sections should be introduced where necessary to allow of the slight engine movements.

Rubber has a fairly even stress strain modulus up to a certain point and then the resistance rises rapidly. This makes it an excellent material for vibration absorbing mountings, but the

pressure per square inch should be kept low, and the thickness as generous as circumstances permit in order to get satisfactory damping and long life.

The oil engine in vehicles calls for very careful mounting. The severe torque reaction due to the high compression ratio and the full air charge, together with the low idling speed which is demanded, make the angular swings of the engine on a flexible mounting very large indeed, and it becomes necessary to provide really flexible pipe connections which can move through big amplitudes continuously without distress.

Controls, etc., must be so arranged that their adjustment is not altered by the engine movements.

A separate gear-box with short cardan shaft connection to the engine is a luxury which is also indicated if competition permits. Steel springs of tapered helical form (in which the large end coils sit down on the support under load and stiffen the rate) have been used with success for this mounting, as greater amplitude can be provided than would be possible with rubber of reasonable bulk.

Maintenance.

The principal points which should be looked to constantly are:—

(1) The supply of clean filtered oil and fuel. Any attention paid to the former will be repaid in bearing life. A constant bleed from the pressure system through a fine filter, returning the oil to the suction side of the pump, will continually remove undesirable matter which may pass the other filters, and the whole of the oil in the system passes through in a short time.

(2) Exhaust thermometers should be provided for each cylinder, as they show very quickly the beginning of any deterioration of the engine, and indicate the line in which it is located.

Routine observation of the temperatures will give warning of the need for periodic attention to such parts as piston rings, and exhaust valves in poppet engines, and head rings and sleeves in sleeve valve engines.

It is absolutely essential that pistons should not be run long with their rings stuck and packed with carbon as the next stage is overheating and possible seizure.

Bearings.

A well-tested design should not give trouble, but wear will occur and should be taken up as opportunity offers.

As crankpins wear most on the inside, they become oval, and big ends should be tried all round to make sure they are not too tight on the large diameter. The shells should bed well in the rod and be hard together at the joint. Tighten the bolts to the makers' instructions, so that the tension is greater than the maximum top inertia, otherwise the joint will open at each cycle, with almost certain breakdown.

Main bearings must be let up very carefully. Although torsional stresses are the most frequent cause of crankshaft failure, many cases have occurred in which alternating bending loads on the shaft, caused by uneven wear-down of bearings or unwise scraping and letting up, have been to blame either by themselves or as forming an aggravation of the previous conditions.

Every effort must be directed to maintaining a straight line through the bearing centres. In well made and properly erected engines the bores of the bearing housings will be aligned, and uneven thickness of the bottom shells is an indication of uneven wear-down.

Where 'hard' water is supplied to an engine, hard scale forms at the hot points, and when possible such places as the insides of the head near the exhaust valve and injector, and all restricted water passages and vent holes should be cleaned.

Injectors should be inspected frequently to see that the nozzles are clear of carbon craters and are not eroded (this occurs when acid products or fine grit are present in the fuel), that the needles are free and the springs correctly adjusted.

Standard Specifications.

The British Standards Institution (B.S.I.) issue booklets containing standard specifications or internal combustion engines of various kinds, and these should be obtained from the offices of

the Institution at 28 Victoria Street, London, S.W. 1, as they will be of assistance in arranging contracts. The chief items of technical interest are as follows :—

(1) *Cyclic Irregularity.*—The angular speed variation shall not exceed 1 in 75 for engines of 1 or 2-cylinders, and 1 in 150 for 3-cylinders or more. This irregularity shall be calculated as $\frac{\text{maximum speed}-\text{minimum speed}}{\text{mean speed}}$ during 1-engine cycle, and is independent of the number of poles in the generator. The engine maker may assume that the armature has 10 per cent. of the flywheel inertia.

(2) *Angular Deviation* shall not exceed $2\frac{1}{2}°$ electrical degrees fore or aft of a point revolving at uniform angular velocity.

Calculations for these two items have been given in Part III.

(3) *Governing.*—5 per cent. for A.C. parallel generators and $3\frac{1}{2}$ per cent. for D.C. or A.C. singly, is specified as the total permanent speed variation.

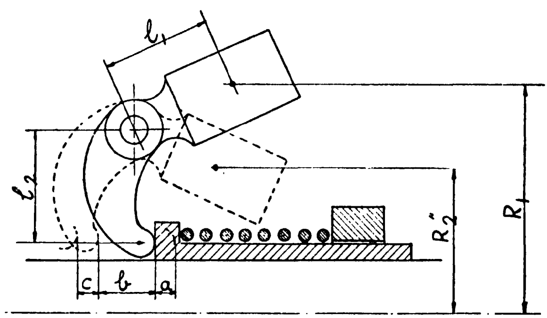
In designing the governor ascertain how much movement of the control rod on the fuel pump is required for :—

(*a*) Idling to shut off.

(*b*) Idling to full load.

(*c*) Full load to maximum overload or extra fuel starting position.

They lay out the weights and connecting gear to cover a total travel = $(a + b + c)$ and calculate the spring stiffness by finding the force required to balance the centrifugal pull of the weights, first at the idling position with a speed of S − $3\frac{1}{2}$ per cent., and secondly at the full load position with a speed S (S = rated speed). These two points cover the distance b and the spring rate is fixed by the difference in load divided by the travel b. Calculate the actual spring as for a valve spring (see page 223) allowing the full travel $(a + b + c)$ with a maximum stress not over 30-tons/sq. in., and a range not exceeding 15 tons/sq. in. Note that if the spring does not move the same distance as the centre of gravity of the weights, the necessary correction must be made for the leverage. See fig. 27.



WEIGHTS SHEWN AT ENDS OF NORMAL GOVERNED RANGE "b"

a = DISTANCE FOR SHUT OFF

C - DISTANCE FOR OVERLOAD.

FIG. 27.

Total of all weights, W lbs.
Rated speed S revs. per min.
Force on Spring at R_1 in.

$$= R_1 \times W \times \left(\frac{S - 3\frac{1}{2}\,\%}{1{,}000}\right)^2 \times 28\cdot2 \times \text{lbs.}$$

and at R

$$= R_2 \times W \times \left(\frac{S}{1{,}000}\right)^2 \times 28\cdot2 \times \text{lbs.}$$

The engine should then settle down with a variation of $3\frac{1}{2}$ per cent. from idling to full load.

All connection should be as free and direct as possible. Any friction or sluggishness in the governor gear will cause hunting, because the delay will be followed by over-correction. Use ball races for the fulcrum pins, and make the governor sufficiently powerful to overcome the pump resistance easily. In big plants a small governor may work a relay, such as a piston valve controlling an oil pressure cylinder which in turn adjusts the speed.

Allowances in engine output are laid down for differences in temperature and pressure conditions as under:—

Standard pressure 30 in. mercury = 760 mm.
,, temperature 62° F. (16·7° C.).
Allowances 4 per cent. per 1 in. of mercury = 1·6 per cent. per cm.
,, 3 ,, ,, 10° F. (5·6° C.).

The lower the pressure and/or the higher the temperature the less the engine output. Humidity is not specified and should be agreed between the manufacturer and purchaser.

Lloyd's Register of British and Foreign Shipping has drawn up rules for the survey of marine internal combustion engines and the auxiliary plant connected with them. While many of the rules concern the installation in the ship and the provision of safety devices, rather than the engine itself, particular attention has been given to the crankshaft and detailed dimensional limitations laid down, in accordance with the cylinder diameter, stroke, bearing span and maximum gas pressure,

It is also required that the plant shall not run at a dangerous synchronous speed.

These rules cover petrol and oil engines of low and high compression and may be obtained from the offices of the Registry, 71 Fenchurch Street, London, E.C. 3.

Particular cases and special designs will always be considered by Lloyd's, and other underwriting bodies, and allowance made for the use of superior materials.

See also Descriptive Section XXVIII.

Peter Brotherhood Ltd.
Gleniffer Engines Ltd.
Hardy Spicer Ltd.
Heenan & Froude Ltd.
Wellman Bibby Co. Ltd.

SECTION XXVIII

PART IV

GAS TURBINES.

(Contributed by B. Wood, M.A. (Cantab.), A.M.I.Mech.E.)

Types and Principles of Operation.

A gas turbine is one in which the working substance is a gas rather than a condensible vapour as in the steam turbine or a liquid as in water turbines. A turbine can utilise gas from any source in hot or cold state where available under sufficient pressure and in suitable quantity to make the installation worthwhile. Cold air turbines have been used in torpedoes and in other applications. Hot waste gas from chemical processes has occasionally been utilised. A turbine was also employed in the German V2 rocket projectile to drive the fuel pumps. These are gas turbines in the sense above defined though they rely on stored energy. However, major technical interest lies in self-contained gas turbine sets designed to produce power by burning fuel as in oil engines. Such a self-contained plant must produce its own compressed gas. The gas can be atmospheric air, compressed and heated by burning fuel in it, then passed through the turbine and finally exhausted to atmosphere (the open cycle): alternatively, it may be air or any other gas in a closed circuit, in which case a further operation of cooling back to the initial temperature must be added to complete the process. The surplus of power generated in the turbine above that required to drive the compressor, representing only a fraction of the whole, is the useful or coupling output.

It will be understood that expansion and compression can also be carried out in reciprocating machines but these are usually heavier and more costly though able to withstand higher temperatures intermittently.

Three different ways of carrying out the process have been tried :—

1) Compression in a rotary compressor followed by constant pressure heating and subsequent expansion in a turbine: examples, Brown Boveri, Escher Wyss, Whittle (in the application to jet propulsion of aircraft).

(2) Compression in a piston compressor followed by constant volume or constant pressure burning with partial expansion in the cylinder and final complete expansion in a turbine: examples, Büchi and Pescara.

(3) Partial compression in a rotary compressor followed by explosive burning at constant volume in a chamber and subsequent expansion in a turbine: example, Holzwarth.

The first method, sometimes referred to as the 'combustion' cycle to distinguish it from the explosion cycles, is the oldest in conception and is that used in most plants to-day. It was tried out experimentally in the early years of the century, but with the low efficiency of the centrifugal compressors then employed and the moderate temperatures that turbine blade materials were capable of withstanding, the sets were unsuccessful, producing little or no surplus power. Consequently interest shifted for some time to the other two methods which seemed to offer a way out of both of the difficulties of high temperature and poor compressor efficiency. Holzwarth worked on the problem over a long period of years and succeeded in installing sets of 1,000 H.P. and 2,000 and 5,000 kW in Germany, running on blast furnace gas, but it is not clear to what extent they were really a success or what efficiencies were attained. Büchi and Sulzer Brothers, experimenting with pressure charging of a Diesel engine, carried the charging pressure up to several atmospheres. The proposal was made to drive the compressor from the engine and to arrange that it should absorb all the engine output, leaving the external work to be generated in a separate turbine taking the hot exhaust gas from the engine. Götaverken and Malone have proposed to use this scheme for the propulsion of ships. Pescara proposed to employ a 'free piston' engine, in which the force of the explosion in the engine cylinder is transmitted direct to the compressor piston, without the need for a crank shaft, in the manner of the Junkers compressor. A plant of this nature was run experimentally in France just prior to the war, and appeared to offer promise of commercial success. It is now being further developed in France in the form of the equipressure boiler.

The Combustion Gas Turbine.

Since this is the type which has recently come into industrial, military and naval use, it merits discussion in detail. The cycle employed in the simplest case is the constant pressure cycle referred to in the U.S.A. as the Brayton cycle, and in England as Joule's air engine cycle. This cycle without refinements such as intercooling, reheating or heat recuperation, permits only a moderate efficiency to be attained when utilising practical temperatures but serves as a basis for preliminary

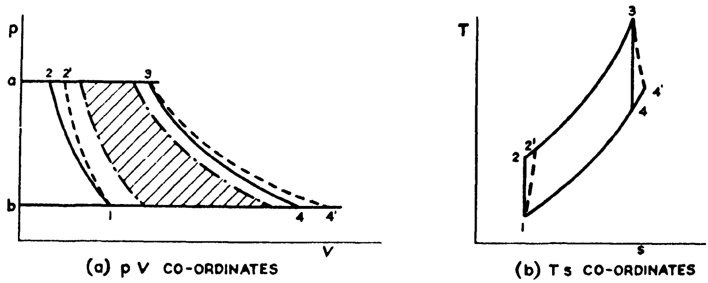


FIG. 1.—Diagram of Simple Cycle.

consideration. The diagram (fig. 1) is the same whether the circuit is open or closed. Fig. 1 (*a*) refers to pressure—volume co-ordinates, and fig. 1 (*b*) to entropy—temperature. In the ideal process shown in full line, the curve 1–2 represents isentropic compression (*i.e.* compression without losses internal or external). The line 2–3 represents heating at constant pressure either by burning fuel in the air or by passing the gas over a surface heater. Curve 3–4 represents isentropic expansion in the turbine. The line 4–1 represents rejection of the exhaust heat to the atmosphere in the open cycle or cooling at constant pressure in the closed cycle. The area a21b shows the work of drawing in, compressing and delivering the air and the area a34b the work done in the turbine or engine. The area of the 'indicator diagram' 1 2 3 4 is the net work output in the ideal case. In practice, compression follows another curve 1–2′ (shown dotted). This is due to stage inefficiency which causes the gas to become hotter and progressively of greater volume than in isentropic compression. The work applied is also greater by the amount of the stage inefficiency, so that a line shown in chain dot can represent the actual compressor work. Similarly, stage inefficiency in the turbine results in only a part of the available work being extracted at each stage so that the volume is increased and expansion follows the dotted line 3–4′. The fraction of the available work extracted is shown by the chain dotted line. The area between the two chain dotted lines shown hatched is the practical work output neglecting any pressure drops in resistances.

Performance.

The approximate performance of gas turbine cycles can be calculated in different ways. The compressor and turbine overall efficiencies e_c and e_t are often assumed, by which means the turbine and compressor work can be derived from the adiabatic work. This may be calculated from the well-known adiabatic expansion law

$$pV^k = \text{constant, where } k = \frac{C_p}{C_v} = \frac{\text{specific heat at constant pressure}}{\text{specific heat at constant volume}}$$

Denoting the absolute temperature by T, etc., and writing

$$\frac{p_2}{p_1} = \frac{p_3}{p_4} = r$$

Then $T_2 = T_1 r^{\frac{k-1}{k}}$ and $T_4 = T_3\left(\frac{1}{r}\right)^{\frac{k-1}{k}}$

The ideal compressor work

$$= \int_{T_1}^{T_2} V dp = C_p T_1\left(r^{\frac{k-1}{k}} - 1\right) \quad \text{or } C_p(T_2 - T_1)$$

The ideal turbine work

$$= \int_{T_4}^{T_3} V dp = C_p T_3 \left(1 - \left(\frac{1}{r}\right)^{\frac{k-1}{k}}\right) \quad \text{or } C_p(T_3 - T_4)$$

The corresponding values of compressor work and turbine work in practical turbines can be obtained by dividing and multiplying the ideal values by the efficiency ratio e_c or e_t respectively and by using appropriate values of C_p according to the mean temperature, making allowance for the extra weight of fuel w added per pound of air.

The cycle efficiency can be readily calculated as

$$\frac{\text{net work}}{\text{heat input}} = \frac{C_{p34}(1 + w)e_t(T_3 - T_4) - C_{p12}\frac{1}{e_c}(T_2 - T_1)}{C_{p23}(1 + w)(T_3 - T_2)}$$

An objection to this method is that e_c and e_t are not actually invariants but depend on the pressure ratio, the stage efficiency and the number of stages.

It is more rational to assume a constant stage efficiency and to take into account the influence of pressure ratio. This can be done readily mathematically if the stage efficiency e_s is taken to be that of an infinitely small stage called the 'polytropic efficiency.' Expansion then follows the polytropic law pV^n = Constant

$$\text{where } \frac{n-1}{n} = e_s\left(\frac{k-1}{k}\right) \text{ in expansion and } \frac{1}{e_s}\left(\frac{k-1}{k}\right) \text{ in compression.}$$

Temperatures are related as above writing n instead of k.

The temperature rise of the compressor then becomes

$$T_2 - T_1 = T_1\left(r^{e_s\left(\frac{k-1}{k}\right)} - 1\right)$$

Similarly the drop in the turbine

$$T_3 - T_4 = T_3\left(1 - \left(\frac{1}{r}\right)^{e_s\left(\frac{k-1}{k}\right)}\right)$$

The cycle efficiency can be worked out as above allowing for variations in k if desired but for many purposes this variation can be ignored. The calculation can also be made readily on a 'polytropic chart.'

The efficiency obtainable with this plain cycle depends in a marked degree on the temperature adopted before the turbine and that before the compressor. Fig. 2 shows in full line how the

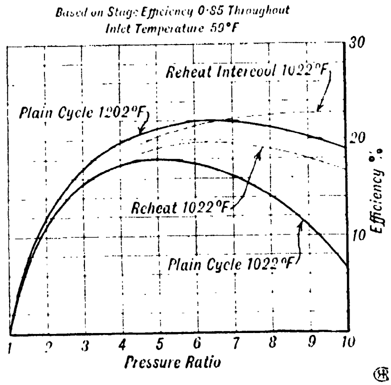


FIG. 2.—Efficiencies of Various Cycles without Heat Exchangers.

calculated efficiency varies with pressure ratio for initial conditions of 1,022° F. (550° C.) before the turbine and alternatively 1,202° F. (650° C.), the air temperature being taken as 59° F. and the stage efficiency in both turbine and compressor as 85 per cent. The lower temperature was considered to be about the limit for continuous operation when a 4,000 kW emergency generating unit was installed at Neuchâtel in Switzerland in 1939. This gave a test efficiency of 17·4 per cent. at the terminals with an air temperature of 78° F. The higher initial temperature might now be accepted as permissible with the same type of turbine. It is clear that the moderate efficiency attainable restricts this cycle to duties where fuel cost is of lesser account than low capital cost, simplicity and independence of water supply as, for instance, in the case of peak load

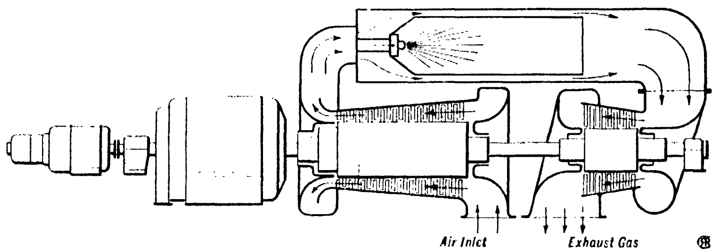


FIG. 3.—Arrangement of 4,000 kW Unit for Neuchâtel.

or emergency plant and also in aircraft. Fig. 3 shows the arrangement of such a plant for land service, and fig. 4 the application to jet propelled aircraft where the shaft furnishes no power.

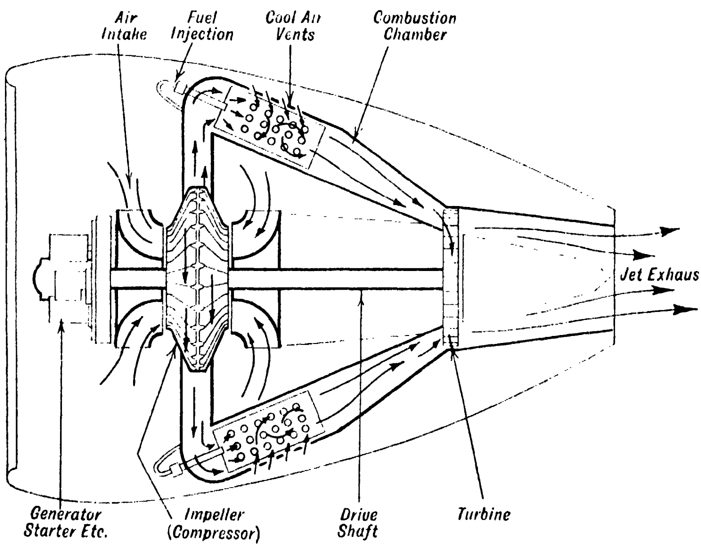


FIG. 4.—Simple Diagram of Jet Propulsion Unit.

Elaborations on the Plain Cycle.

Further developments and elaborations on the simple cycle are shown in fig. 5. The following fundamental means can be employed singly or in combination for obtaining improved efficiency and/or increased output per pound of air :—

(*a*) Reheating in the turbine (multi-stage burning).
(*b*) Intercooling in the compressor (or precooling before the compressor).
(*c*) Exhaust heat recuperation by a heat exchanger (also known as regeneration).

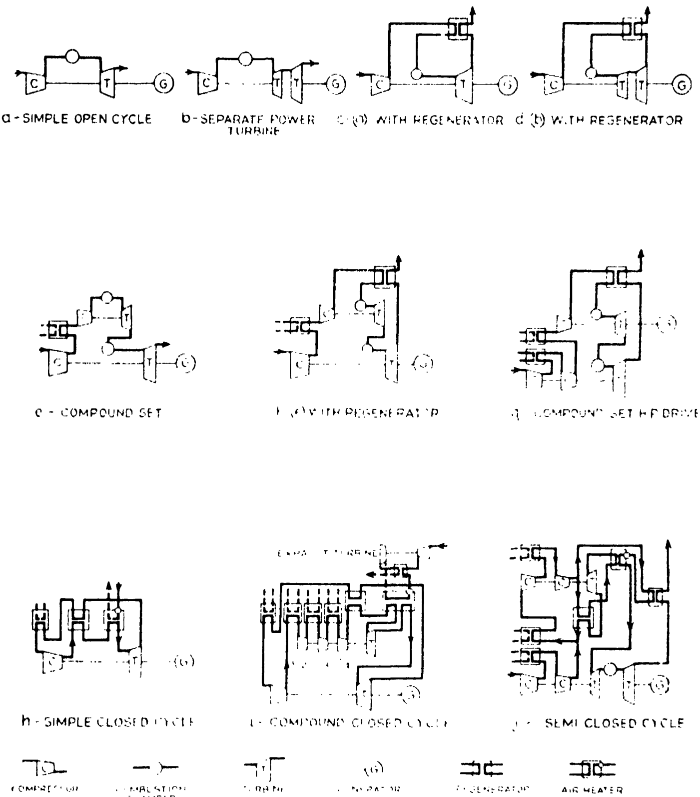


FIG. 5.—Cycles for Gas Turbines.

Various arrangements involving more than one shaft are adopted to avoid the risk of surging in the compressor, to overcome limitations on the quantity of fluid which can be handled on a single entry or exhaust annulus, to obtain part load characteristics suited to the drive and, in some cases, to avoid infringing patents.

It will be understood that the number of possible combinations of multi-shaft machines is large, and only the more important variants are shown. Turbines may be arranged in parallel as well as in series. The cycle may also be closed (fig. 5 (h) and (i)) or in an intermediate form semi-closed (fig. 5 (j)).

Reheat.

Reheat is readily applied in open circuit cycles by burning a second charge of fuel in a second combustion chamber after partial expansion in the turbine. This leads to a higher mean temperature without exceeding the maximum mentioned above and the turbine output is increased without increasing the amount of air or the heat rejected to the exhaust. The specific output and efficiency are thus increased and higher pressure ratios become advantageous.

Cooling the Compressor.

Similarly the mean temperature in the compressor is a guide to the work of compression. As the compression ratio is increased the saving by recooling the air back to its initial temperature after partial compression becomes sufficient to justify the cost of installing a cooler. The useful net output is thus increased resulting in improved efficiency and reduced capital cost per kilowatt of the turbine and compressor. Intercooling becomes important at the higher pressure ratios required when reheating is adopted in the turbine and the division into separate cylinders facilitates it.

Heat Recuperation.

The exhaust temperature in the simple cycle may be of the order of 600° F. while the temperature after compression is of the order of 350° F., therefore there is scope for transfer of heat from the exhaust to the air after compression. Regeneration in this way directly reduces the amount of fuel to be burnt. This has a marked effect on efficiency but the gains theoretically obtainable are reduced by pressure drops in the heat exchanger which must be overcome by the compressor, and by the temperature drop across the heater surface necessary to transfer the heat from the gas side to the air side. It will be clear that the scope for the use of heat exchanger is increased when the compressor is cooled, because the temperature at exit from the compressor is then lower. The most effective combination is reheating and intercooling with heat exchanger.

The thermal performance of a heat exchanger can be specified by its thermal ratio or 'efficiency' viz. :—

$$e = \frac{\text{mean temperature change } R_m}{\text{maximum temperature difference available } H.}$$

In the ideal case of a contra-flow exchanger of area A with equal quantities of fluid Q on each side and equal specific heats C_p we have

$$R + D = H. \quad D = \text{temperature difference.}$$

$$K = (\text{heat transfer coefficient}) \propto \frac{RC_pQ}{AD}$$

$$\therefore\ e = \frac{R}{H} = 1 - \frac{D}{H} = \frac{A}{A + \frac{C_pQ}{K}} = \frac{A}{A + B} \left(\text{writing } B = \frac{C_pQ}{K}\right).$$

The following table shows how A varies with e. The inordinate increase of A for each increment of e at higher values will be noted.

e	0·5	0·66	0·75	0·8	0·875	0·9	0·95	0·99
A/B	1	2	3	4	7	9	19	99

Status of Development.

There are upwards of 50 gas turbines built or at present in construction by some eighteen firms in Switzerland, the U.S.A. and Great Britain. This does not take account of the large number of smaller machines built from 1934 as auxiliaries for Velox boilers (about 90) and for Houdry oil cracking plants (about 33). The list on page 259 includes 25 units totalling 200,000 kilowatts ordered for electric power stations, of which particulars have been published. There are a few other units known to be in construction but not yet made public. In addition there are some 11 machines built or in construction for locomotives, and at least 5 for ships (possibly more). The remainder are mainly being built for internal experimentation and development. There are at least ten firms in Great Britain building such sets, apart from aircraft units.

The largest machine constructed to date is of 27,000 kW, and the best efficiency offered is 35 per cent. Nearly all builders are using the open cycle, the closed cycle being represented only by Escher Wyss and their licensees, and the semi-closed only by Sulzer Brothers, though patent literature shows interest by other concerns.

LIST OF GAS TURBINES ORDERED FOR ELECTRIC POWER STATIONS.

Date.	Location.	No.	Rating.	Maker.	Cycle (see Fig. 5).	Temp. °F.	Efficiency per Cent.	Fuel.
In service 1940	Neuchatel, Switzerland	1	4,000 kW	Brown Boveri	a	1,022	17·4 test	Oil
Tested 1946	Bucharest, Rumania	1	10,000 kW	Brown Boveri	e	1,100	23·3 test	Natural gas
Tested 1946	Chimbote, Peru	1	4,000 kW	Brown Boveri	a	1,022	19·5 test	Oil or gas
Delivered 1948	Lima, Peru	1	10,000 kW	Brown Boveri	g	1,100	28·3 guar.	Oil
Ordered 1945	Cucuta, Colombia	1	1,650 kW	Brown Boveri	c	1,100	25 guar.	Oil
Ordered 1945	Caracas, Venezuela	2	1,650 kW	Brown Boveri	c	1,100	21 guar.	Oil
Ordered 1945	Alexandria, Egypt	1	1,200 kW	Brown Boveri	c	1,100	22·9 guar.	Oil
Ordered 1945	Persia	2	4,000 kW	Brown Boveri	a	1,100	18 (at 95° F.)	Natural gas
In service December 1947	Beznau, Switzerland	1	13,000 kW	Brown Boveri	g	1,100	30·6 (at 41° F.)	Oil
Promised January 1949	Beznau, Switzerland	1	27,000 kW	Brown Boveri	g	1,100	31 (at 41° F.)	Oil
Promised August 1949	St. Denis, Paris	1	12,500 kW	Escher Wyss	i	1,250	34 guar.	Oil
Promised December 1950	Weinfelden, Switzerland	1	20,000 kW	Sulzer	j	1,200	35 approx. (at 41° F.)	Oil
Promised 1951	Stretford, Manchester	1	15,000 kW	Metro.-Vick.	f	1,200	30 approx.	Oil
Promised 1951	Dunston, Newcastle	1	15,000 kW	C. A. Parsons	—	1,200	27 approx.	Oil
Promised December 1950	Dundee, Scotland	1	12,500 kW	John Brown	i	1,250	34 guar.	Oil
Ordered 1948	Oklahoma, U.S.A.	1	3,500 kW	G.E.Co., U.S.A.	a	1,400	17 at 80° F.	Natural gas
Ordered 1948	Oklahoma, U.S.A.	2	5,000 kW	G.E.Co., U.S.A.	g	1,500	26·4	Natural gas
Ordered 1948	Central Maine, U.S.A.	1	3,500 kW	G.E.Co., U.S.A.	a	1,400	17 at 80° F.	Oil
Ordered 1948	Bangor, U.S.A.	1	5,000 kW	G.E.Co., U.S.A.	g	1,500	28	Oil
Ordered 1949	Metropolitan Water Board	1	2,500 kW	Brush	b	—	16·2	Oil
Ordered 1949	Metropolitan Water Board	1	2,500 kW	Metro.-Vick.	a	—	16·8	Oil
Ordered 1948	Metropolitan Water Board	1	1,875 kW	E.E. Co.	b	—	16	Oil

Efficiencies quoted on lower calorific value and on about 60° F. air or water temperature except where mentioned.

It appears that the limit of output for open cycle machines on a single exhaust is about 10,000 kW, unless the air temperature is particularly low. It also appears unlikely that a set larger than the present limit of 27,000 kW will be built in an open circuit gas turbine.

The following brief notes refer to most outstanding machines on point of size, efficiency or other aspect.

(a) *Stationary Plants.*

A 10,000 kW machine built by Brown Boveri for the Filaret Station, Bucharest, Rumania was demonstrated on test in 1946. It has two shafts using a pressure ratio of 11 : 1 with reheating but without regenerator according to the arrangement shown in fig. 5 (e). The set is intended to run on natural gas, hence high efficiency is not called for. The efficiency obtained was 23·3 per cent. with an inlet temperature of 1,063° F. at 12,000 kW. The time for starting up to putting on full load was seven minutes.

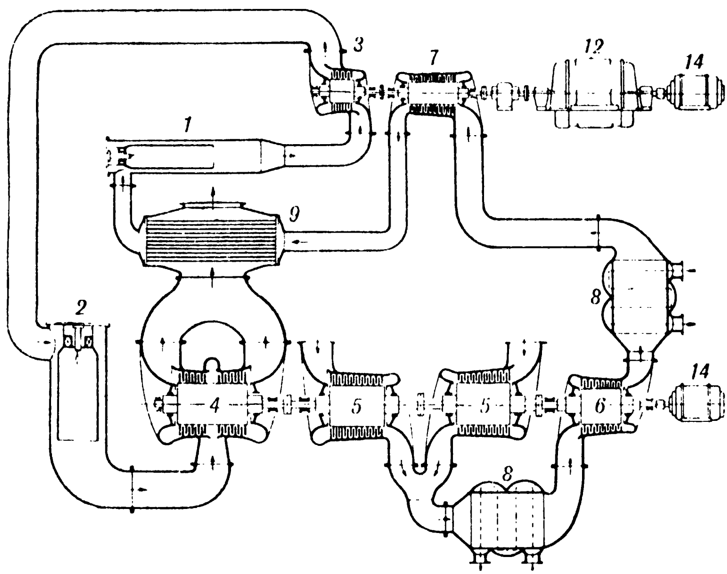


FIG. 6.—Schematic arrangement of the 27,000 kW unit for Beznau. (Numbers refer to Fig. 7.)

A somewhat similar Brown Boveri machine also of 10,000 kW was completed a few months later for Lima, Peru. In this case a higher efficiency was called for with oil firing. The cycle is that shown in fig. 5 (g), a regenerator being used. The alternator is driven from the h.p. shaft. The l.p. shaft thus serves as a supercharger and runs at variable speed according to load. The part load efficiency is thereby improved. The efficiency guaranteed was 28·3 per cent. with 86° F. air inlet and 1,100° F. maximum temperature. Pressure ratio is 9 : 1. The price was understood to be about £23 per kW.

A similar machine using the same frame size was installed in December 1947 at the Beznau Station in Switzerland. Rating on 41° F. air and water temperature and taking account of the overload capacity permitted of raising the output to 13,000 kW and the efficiency to 30·6 per cent. It was not tested in the works prior to despatch.

A second set of 27,000 kW is being erected in the same station. The cycle is the same, as are some of the components, duplicate l.p. machines being connected in parallel as shown in fuller detail in fig. 6. The efficiency guaranteed is 34 per cent. The Beznau Station, containing 40,000 kW, is the largest gas turbine station in the world. The generating plant cost approximately £17·5 per kW, and the whole station £29 per kW. A plan and elevation are shown in fig. 7.

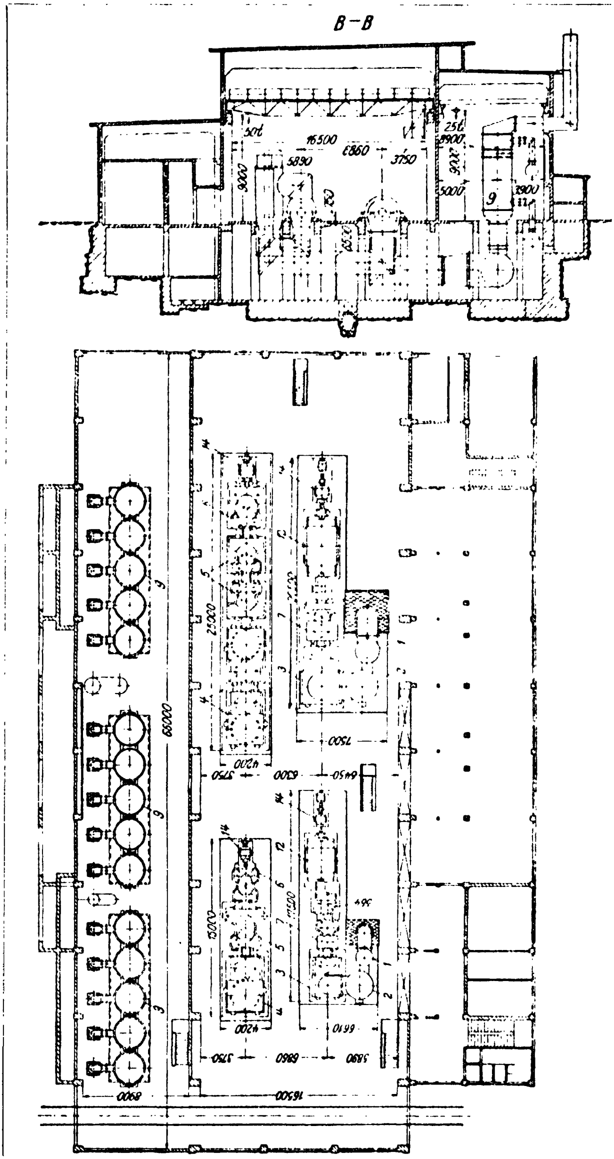


FIG. 7.—Beznau Power Station—Plan and Elevation (dimensions in millimetres).
13,000 kW Two-shaft Set on left. 27,000 kW Two-shaft Set on right.

(1) H.P. combustion chamber, (2) L.P. combustion chamber, (3) H.P. turbine, (4) L.P. turbine, (5) L.P. compressor, (6) I.P. compressor, (7) H.P. compressor, (8) Intercooler, (9) Regenerator, (12) Generator, (14) Starting motor.

Escher Wyss expect to have a 12,500 kW closed-circuit set in service at St. Denis Station, Paris, by about August 1949. The cycle, that shown in fig. 5 (i) and fig. 9 resembles very closely a steam cycle. Air is continuously recirculated, being first heated in an air heater equivalent to a boiler, expanded through the turbines with reheating, passed through a regenerator to a pre-cooler and then compressed in multi-stage compressors with intercooling and returned through the regenerator to the air heater. All heat rejected is carried away by cooling water, the pre-cooler and intercoolers being the analogue of a steam condenser. The main advantage over the steam cycle is in the use of a large regenerator and a small expansion ratio of about 10 : 1. There are two shafts, the h.p. turbine with the higher pressure compressors being free. The l.p. turbine drives the l.p. compressor and the alternator. The guaranteed efficiency at 1,250° F. with reheat and with 60° F. circulating water, is 34 per cent. The air heater is pressure-fired at about 3 atmospheres absolute by a combustion air blower driven by an exhaust gas turbine, which also drives a recirculating fan to attemperate the initial flame temperature. A feature of this cycle is the

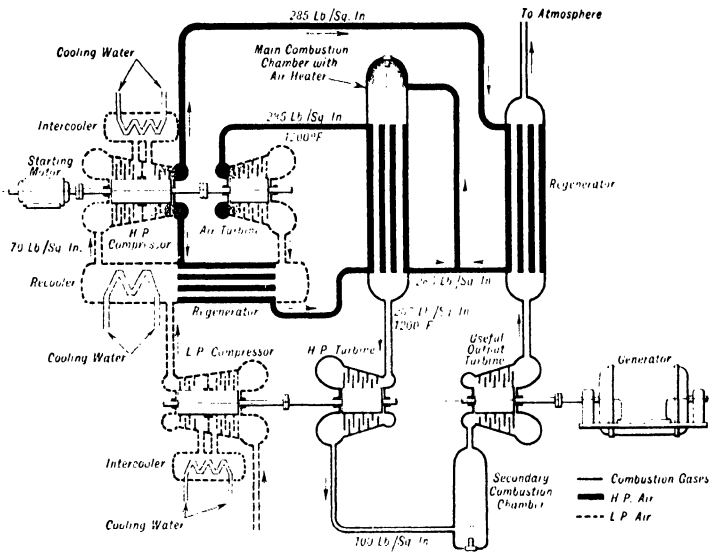


FIG. 8.—Circuit of 20,000 kW Sulzer Semi-closed Cycle Gas Turbine for Weinfelden.

variation of load by changing of the density of the medium. This results in a very flat efficiency characteristic. To raise load, air is let into the closed-circuit in conjunction with increase of fuel rate so as to maintain approximately constant inlet temperature. To reduce load air is expelled at the same time as the fuel is reduced and in addition the turbines may be momentarily by-passed.

Sulzer Brothers are building a 20,000 kW unit for Weinfelden Station, Switzerland, which is on the same company's network as Beznau. This is expected to be completed by the end of 1950. The circuit employed is a semi-closed cycle shown in fig. 5 (j). The efficiency expected is of the order of 35 per cent. based on an initial temperature of 1,200° F. and on cooling water and air at 41° F. The somewhat intricate circuit will be understood from fig. 8. There are three shafts which may be designated ; ' the air turbine,' ' the charging set,' and ' the work turbine.' The fluid fed in through the charging compressor passes out through the work turbine, but roughly an equal quantity of air is recirculated in the closed-circuit through compressor, regenerator, air heater and turbine back through regenerator and cooler. The pressure in the air heater is about 20 atmospheres on both sides of the tubes. One-third to a half of the heat in the fuel is transmitted through the tubes of the air heater into the air cycle, the remainder is employed in the charging turbine with supplementary reheating in combustion chamber before the work turbine. The recuperator permits of recovering a large fraction of the heat rejected by this turbine.

Two sets each of 15,000 kW have been ordered by the British Electricity Authority, one for Trafford Power Station, Manchester, from Metropolitan-Vickers Electrical Company, and the

other for Dunston Power Station, Newcastle, from C. A. Parsons & Company. These are expected to be completed in 1951. The cycle employed by Metropolitan-Vickers will be that shown in fig. 5 (f), employing an initial temperature of 1,200° F. with which it is expected to obtain an efficiency of about 30 per cent. The details of the Parsons machine have not yet been made public. Parsons have also in hand a 10,000 kW machine for the National Gas Turbine Establishment, Pyestock.

A 12,500 kW set is being built by John Brown of Clydebank under Escher Wyss licence for the Dundee Station of the North of Scotland Hydro-Electric Board. This is scheduled for completion late in 1950. It employs a closed cycle and is generally similar to the Paris unit described above. Fig. 9 shows the cycle in more detail. The efficiency quoted as 31·9 per cent. on the higher calorific value is equivalent to 34 per cent. on the lower, which is the basis on which most other gas turbines are rated.

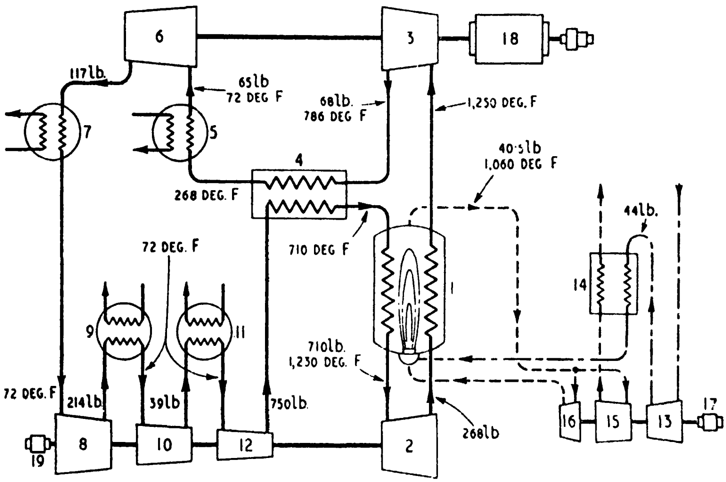


FIG. 9.—Circuit Diagram of Hot-Air Turbine (pressures shown in lb. per sq. in. absolute.)

(1) Air heater; (2) H.P. turbine; (3) L.P. turbine; (4) Heat exchanger; (5) Pre-cooler; (6) L.P. compressor; (7) First intercooler; (8) First I.P. compressor; (9) Second intercooler; (10) Second I.P. compressor; (11) Third intercooler; (12) H.P. compressor; (13) Combustion air compressor; (14) Combustion air pre-heater; (15) Exhaust gas turbine; (16) Recirculating gas fan; (17) Motor; (18) Main generator; (19) Starting motor or turbine.

Among smaller sets mention may be made of a 2,000 kW closed-circuit machine built by Escher Wyss, Zurich, in 1939 and tested in 1945. It is a two shaft unit without reheat and with an atmospherically aspirated air heater. The tested efficiency was 30·5 per cent. at full load with the high initial temperature of 1,300° F. and cold circulating water. It has now run about 3,500 hours.

The Oerlikon Company, Zurich, built a 1,000 kW set which has been running since December 1946, and is of interest as having a centrifugal compressor. The circuit is nearly as in fig. 5 (d). The exhaust gas is used to fire the works boilers. It has run some 4,000 hours and generated some 2,000,000 kWh, but no statement of the efficiency has been published.

Sulzer Brothers completed in 1948 a 7,000 h.p. unit employing their semi-closed cycle somewhat as above described which is to be tested in their works on a waterbrake.

In this country three industrial machines have been completed to date. One of these, a 500 h.p. unit, was begun by C. A. Parsons in 1939 and completed in 1945. It has been run for some 1,500 hours on a water-brake to provide experience with different blade materials and with different fuels. The efficiency is about 15 per cent. (as at generator terminals) based on 1,050° F. inlet. The circuit is as in fig. 5 (c). A somewhat similar machine built by John Brown to Pametrada designs has been run intermittently for about a year. Metro.-Vick. have recently completed a 2,500 kW machine for their own works supply. This is derived from aircraft practice, being somewhat similar to the unit in the gunboat mentioned below, but with the addition of a regenerator fig. 5 (d). The efficiency has not been disclosed.

Orders for three units of 1,875 and 2,500 kW for emergency standby purposes have just been placed with British firms by the Metropolitan Water Board at prices varying from £18 to £30 per kW.

In the U.S.A., four firms, the G.E. Company, Westinghouse, Allis Chalmers and the Elliott Company, have built and tested machines either as Government sponsored experiments or as speculations. A G.E. machine, which is derived from aircraft technique, was recently tested and at 3,500 kW gave an efficiency of 17 per cent. with an initial temperature of 1,470° F. The G.E. have several machines in hand including one of 5,000 h.p. for New Hampshire. In a 3,500 kW unit for the Huey Station in Oklahoma the exhaust gas at about 1,000° F. will be used for feed heating. An Elliott machine, employing Lysholm compressors, tested in 1946 gave an efficiency of 27 per cent. at 1,230° F. A Westinghouse 2,000 h.p. set completed in 1946 has run some 1,000 hours at 1,350° F. with an efficiency of 16·7 per cent.

(b) *Locomotives.*

Of the eleven locomotive-type gas turbines already built or being built eight are ordered for installation in actual locomotives and four are built or being built as speculations. All will use D.C. electric drive and all are for oil-firing, except two mentioned later destined to run on coal.

A 2,000 h.p. Brown Boveri gas turbine locomotive commissioned by the Swiss Railways in 1940 showed an efficiency at generator terminals of 17·7 per cent. based on a maximum temperature of 1,100° F. It has now run some hundred thousand kilometres on Swiss and French lines and has shown no serious difficulty with fouling of the compressor or in burning heavy fuel oil. When tested on the French Railways against a diesel locomotive hauling the same 600-ton train, the consumption worked out as three times that of the diesel.

A similar locomotive rated at 2,500 h.p. with an expected efficiency of about 19 per cent. is being supplied by Brown Boveri to the British Railways. Both these Brown Boveri locomotives employ a single shaft unit coupled through gears to the generator, the circuit being as fig. 5 (c), with a cross flow regenerator of less than 50 per cent. efficiency mainly to improve part load performance. A machine of the same rating being built by Metro.-Vick. for the former G.W.R. will employ a single shaft set without regenerator fig. 5 (a).

In the U.S.A. a G.E. Co. 4,500 h.p. unit installed in an ALCO locomotive recently began track tests on the Union Pacific line. The circuit is as fig. 5 (a).

The Elliott Company has two gas turbines in hand for locomotives, one an oil-fired unit of 3,000 h.p. for the Santa Fé railroad, and the other a coal-fired unit of 3,750 h.p. to be fitted in a Baldwin locomotive, as part of a coal-burning experiment sponsored by the Locomotive Development Committee of Bituminous Coal Research Inc. Both will use centrifugal compressors and an initial temperature of about 1,275° F. to obtain an expected efficiency of 24 per cent.

Allis Chalmers have completed a similar coal-fired unit of 3,750 h.p. at 1,300° F. to be fitted in an ALCO locomotive. The coal-firing equipment, combustion chamber and grit arrester are to be provided by the purchaser. Further information on this project is given under coal-firing.

The Northrop-Hendy Company also has in hand a 2,000 h.p. oil-fired unit for the Union Pacific Railway.

Among machines built for experiment should be mentioned the Brown Boveri 'comprex' set demonstrated in 1946. The addition of a 'comprex' raised the output of a 2,500 h.p. machine to 4,000 h.p. and the efficiency to 22·5 per cent. with small additional weight and space. The 'comprex' is a combined compressor and expander device consisting of a rotating wheel in the cells of which expansion and contraction take place by making use of compression waves. The maximum temperature has been raised to 1,400° F. since it is confined to the cell wheel and is only encountered intermittently.

(c) *Marine Units.*

No vessel has yet been built engined solely by a gas turbine. The first vessel to be propelled by a gas turbine was MGB 2009 which was run in trials in 1947, using a 2,500 h.p. gas turbine in place of one of its three 1,200 h.p. petrol engines. The design was derived from the Metro.-Vick. jet propulsion Beryl engine with the addition of a multi-stage low pressure turbine coupled through gears to the centre propeller fig. 5 (b). The efficiency obtained was 13·0 per cent. at full load of 2,550 h.p. and about 9 per cent. at half to two-thirds speed. The weight including gears, etc., was 3·5 tons.

The Elliott Company completed in 1940 an experimental unit for the U.S. Navy designed for installation in a Liberty ship ; this is mentioned above. So far as is known the machine has not yet been put into use.

The B.T.H. Co. have in hand a 1,200 h.p. two shaft machine for installation in the Anglo-Saxon Petroleum Company's tanker *Auris*. This will take the place of one of four Diesel electric sets. The h.p. turbine drives the compressor and rests on the l.p. turbine which drives the alternator. The maximum temperature is to be 1,200° F. and the efficiency expected over 25 per cent. A regenerator will be used, the circuit being as fig. 5 (d).

Other development machines include a 3,500 h.p. set by Pametrada for experiment, and a number of sets by different makers for naval purposes. A difficulty in marine applications of gas turbines is the necessity for astern going. This was avoided in the MGB by relying on the two flank petrol engines, and in both the Elliott machine and the *Auris* by the use of electric drive. The use of a reversible pitch propeller is generally looked to as the way out in case of direct drive.

(d) *Aircraft.*

Britain leads the world in aircraft gas turbines. Practically all first line high-speed fighters now employ jet propulsion engines which are allied to the gas turbines used in land practice. The jet engine produces no net power on the shaft, the whole surplus energy going into accelerating the exhaust gases. In aircraft service an immense premium is attached to saving in weight and cross sectional area, while a short life is acceptable. Efficiency is of importance primarily for reduction of fuel weight. Hence elaborations are at present not generally acceptable since their own weight exceeds that of the fuel saved in short flights. Aircraft machines contain the same items, viz. compressor, combustion chamber and turbine. Compressors may be centrifugal as in the Rolls Royce ' Derwent ' and ' Nene,' or the De Havilland ' Goblin ' or ' Ghost,' or axial as in the Metro.-Vick. ' Beryl ' and several foreign types. The axial machine has a better efficiency and a smaller diameter but is at present heavier. Combustion chambers are of metal and invariably multiple as opposed to the single large unit of land types. The turbine has usually a single row of blades, occasionally a double row. Temperatures are commonly in the region of 1,500 to 1,600° F., and life between 200 and 500 hours. The stressing of components is much higher than in land practice. The weight to power ratio is of the order of 0·31–0·46 lb. per lb. of thrust (1 lb. thrust at 375 m.p.h. = 1 h.p.). While the efficiency of such gas turbines is low, 13 per cent. or less, the efficiency of the jet as a means of propulsion is better than that of a propeller at high speed and improves as the flight speed approaches that of the jet. The maximum thrust obtainable from a single jet unit at 5,000 lb. is higher than in propeller types.

In civil and long range military aircraft there is a case for the gas turbine driving a propeller and providing some thrust from the exhaust. Several such units are available, *e.g.* the Armstrong Siddeley ' Python,' the Napier ' Naiad,' the Rolls Royce ' Dart,' and the Bristol ' Theseus.' Fuel consumption is better than in the jet types but still not so good as in reciprocating types, because intercooling and regeneration have not yet been successfully applied.

Commercial Running Experience.

In spite of the large number of land gas turbines actually completed, there is rather scanty running experience amounting, at most, to a few thousand hours in each of several, mainly small, plants. The number of hours run on test in the manufacturers' works amounts to a considerable total, but this is to be differentiated from normal commercial running. Brown Boveri as the most experienced builders can quote some 112,000 miles service running of the locomotive, and a total running into millions of hours with Houdry sets and Velox boiler auxiliaries together, but these operate mostly at lower temperatures than they are rated for. Similarly the locomotive runs much of its time at lower than the full temperature. The only higher temperature machines which have been run on site, are the Neuchatel set and Beznau No. 1, both of which have a total of 400 running hours and about 2,000 respectively.

Fuels.

All gas turbines to date consume oil or cleaned gas (see, however, coal-burning section). The use of gas if not already under pressure entails a separate compressor commonly centrifugal for the sake of regulability. Open cycle machines to date have generally been run on rather light oil, usually a blend of gas oil and heavier residual oil. Gas oil is a distillate and contains no ash. Heavy oils may contain ash to the extent of say 0·25 per cent. The ash may contain certain slag-forming substances which stick to the turbine blades in some designs, while with other designs no trouble has been experienced when using the same oil. The controlling variables have not yet been isolated ; it is possible that the design of the combustion chamber or of the blades may be significant, but pending further investigation several makers are prepared to use only distillate oil.

The closed cycle and the semi-closed expect to be able to use heavy oil, as the conditions are not substantially different from those in marine boilers. The slag is expected to be intercepted within the air heater so that it will not reach the following turbine. The surfaces within the closed circuit of course remain clean.

The price of oil in the United Kingdom varies somewhat with grade and with locality and quantity rebates. Average price are about £7 7*s.* 6*d.* per ton for heavy fuel and £9 10*s.* for gas oil. The price for ships bunkering at United Kingdom ports is £6 4*s.* 6*d.* per ton for heavy oil and £7 15*s.* for diesel oil. The higher calorific value may be taken as about 18,500 B.Th.U./lb., hence the inland price of oil is roughly 60 per cent. above that of coal on a heating basis. A similar ratio applies in parts of the U.S.A. (Price as at March 1949.)

Coal-burning Gas Turbines.

The burning of coal in a gas turbine can be tackled in at least four ways :—

(1) directly in an open circuit turbine as pulverised coal ;

(2) indirectly by gasifying in a producer with subsequent detarring and cooling of the gas ;

(3) indirectly by gasifying in a pressurised producer inserted between the compressor an combustion chamber ;

(4) in a closed-circuit gas turbine by any of the above methods.

Methods 1 and 3 were tried out in a small way by Brown Boveri some years ago and serious difficulties were encountered. Method 1 has been the subject of a very determined and costly full scale research over the last four years sponsored by the Locomotive Development Committee of Bituminous Coal Research Inc., but according to the latest reports no satisfactory means has yet been evolved of removing the ash particles before they enter the turbine. The notion that the ash would pass through the turbine without harm if sufficiently finely ground appears to be unfounded. Method 2 presents no obvious technical difficulties but is wasteful thermally because of the heat lost from the producer and as sensible heat in the gas. It also involves rather costly equipment. Method 3 was to be tried out in Germany but work on the project ceased with the end of the war, though one high pressure producer was built. It appears promising if a tar-free fuel can be used. The whole of the sensible heat would be retained in the cycle and the capital cost reduced by pressurising. Method 4 has not yet been tried but seems to offer fewer unfamiliar problems than some of the other alternatives.

Possibilities of Higher Temperatures.

Both the plain cycle and its variants show a marked improvement in efficiency and in specific output when the temperature at entry to the turbine is raised. Hence designers strive to employ the highest temperature allowable. The permissible temperature is limited mainly by the strength of the material of the first row of moving blades as determined by acceptable creep rate, erosion, or embrittlement, all of which are functions of operating time.

The maximum temperature for continuous operation has been put at 1,150° to 1,300° F. (both well in the red region), depending on the material and the opinion of the designer. For short life operation as in jet propelled aircraft, where the period of service can be restricted to say 300 hours, and more expensive material may be employed, very much higher temperatures, *e.g.* 1500° F. or above, may be tolerable.

The initial temperature permissible depends to some extent on the type of turbine. Most of the commercial sets so far installed are of the reaction type in which the gas enters the moving blades at practically its full initial temperature. In impulse turbines the temperature and pressure are first dropped in a set of stationary nozzles which, not being highly stressed, can withstand a higher temperature. The proposal has been made to employ 1,500° F. and to drop the temperature to a much lower value before entering the first wheel. However, the efficiency of a wheel utilising such a large drop tends to be low and it is not clear whether there will be any net advantage in the high temperature when used in this way.

Methods of Regulation.

Nearly all sets employ axial compressors and since these do not lend themselves to efficient regulation of the quantity of air delivered, except by varying the speed, all constant speed sets in which the compressor and turbine are coupled direct to the drive suffer from the disability that the quantity of air aspirated remains constant and the compressor work therefore changes very little with load. In the simple cycle the only practical means of varying the power delivered at constant speed is to vary the rate of fuel supply and thus change the initial temperature. The maximum efficiency is accordingly reached at full load when the temperature is the highest, while the efficiency falls off rapidly at lower loads. This does not apply to variable speed applications where low speed is associated with low load, *e.g.* in ship propulsion or to the drive of D.C. machines, nor does it apply in such marked degree to the more complicated cycles. If the compressor is dissociated from the constant speed drive, *e.g.* by coupling to a separate turbine on a free shaft, the load can be regulated mainly by varying the air quantity which results in a much better part load efficiency. In the latest Brown Boveri sets as shown in fig. 5 (g), the generator is coupled to the h.p. shaft. This allows the load to be varied by altering the speed of the l.p. shaft. In the Escher Wyss sets load is apparently varied by an altogether different method, but fundamentally the fuel rate remains the means of control. For rapidly throwing off load in the more complicated cycles the gases are by-passed round the turbines.

Starting.

Gas turbines are not self-starting; it is necessary to drive the compressor by external power in order to deliver sufficient air for lighting up the burners, after which the turbine begins to produce sufficient torque to bring the set up to full speed. The starting time in certain open circuit designs is between 5 and 10 minutes, but a much longer time may be required in other machines.

Gas Turbines as Auxiliary Machines.

The gas turbine and compressor find application in connection with certain chemical or combustion processes carried out under pressure. The compressor provides the combustion air required for the process and the turbine recuperates the energy in the products of combustion, furnishing the power to drive the compressor and sometimes additional external power. A step on the way to the emergence of the gas turbine as a prime mover was its successful use since 1936

in connection with the Houdry oil-cracking process. Some 33 sets have been built for this purpose by Brown Boveri and their licensees in the U.S.A. These are started on oil and were tested on oil in the works using temperatures of the order of 1,050° F., but in service surplus power is not the main consideration, and they are operated at lower temperatures, the fuel being carbon which is to be burnt off the catalyst.

A further application of gas turbines is in connection with the provision of wind for blast furnaces. In this case, instead of external power being generated, the excess energy is absorbed in providing compressed air for the furnaces, the compressor being tapped at a suitable point. Blast furnace gas is used as fuel. Two sets, each equivalent to 3,000 kW nett output, were supplied to the Hermann Goering Steelworks just before the war. One of them has now been brought to England. Two further sets for metallurgical works, one for Baracaldo in Spain, and the other for Luxembourg, are being built by Brown Boveri.

Future Prospects.

The difficulty in generalising on gas turbines is the wide variety of types available, each with its own characteristics and associated merits and defects. For instance, the efficiency may vary between 13 per cent. and 35 per cent. while the specific weight may range over a wider ratio. The present wide interest taken in the gas turbine is to a large extent the result of the publication of details of the intensive work carried out during and since the war on the development of aircraft types. The prospects of future development for commercial purposes and for the generation of electrical power should, however, be viewed with caution since it is likely to be found that good efficiencies are obtainable only in high quality, expensive and bulky plant and entail a very accurate knowledge of individual component characteristics. This combined with a knowledge of costs of production will tend to restrict future applications to those duties for which the gas turbine is well suited. These include aircraft use, metallurgical, peak load generation and possibly base load where oil is the only fuel and water is scarce (though this restricts the efficiency obtainable). There are also special cases where the high temperature by-product heat in the exhaust or the coolers makes the gas turbine attractive.

Whether the gas turbine can outclass the diesel in locomotives or operate successfully without electric drive remains undecided. Similarly, marine applications are waiting on land experience though navies are experimenting with sets up to 7,500 h.p. The prospect of the gas turbine competing with steam for base load power stations in industrial countries seems remote in view of the high price of oil, while even if the gas turbine could succeed in burning coal it is not certain that it would offer decisive all-round advantages. The present position must accordingly be regarded as still largely experimental.

SECTION XXIX

NAVAL ARCHITECTURE AND MARINE ENGINEERING

PART I

NAVAL ARCHITECTURE

(Revised by Norman L. Gemmell)

(pp. 271–286)

PART II

MARINE ENGINES (STEAM)

(Revised by A. W. Davis, B.Sc.)

(pp. 287–303)

PART III

MARINE ENGINES (OIL)

(Contributed by A. P. Chalkley, B.Sc. Editor, 'The Motor Ship')

(pp. 305–317)

SECTION XXIX

PART I

NAVAL ARCHITECTURE

(Revised by Norman L. Gemmell)

DIMENSIONS OF SHIPS.

Moulded Dimensions.

LENGTH L = 'Length between Perpendiculars' = Length on load water-line measured from the fore side of the stem ('F.P.,' or 'Fore Perpendicular') to the aft side of the stern post ('A.P.,' or 'After Perpendicular'), or fore side of rudder stock if there is no stern post.

BREADTH B = 'Moulded Breadth' = Breadth measured over the widest part of the frame at the middle of the length L.

DEPTH D = 'Moulded Depth' = Depth measured from top of keel to top of deck beam at side at the middle of the length L. To avoid error the name of the deck to which the depth is measured should be stated.

Registered Dimensions.

LENGTH = Length from fore part of stem head to aft side of stern post, or fore side of rudder stock if there is no stern post.

BREADTH = Breadth measured over the outside of the shell plating at the widest part.

DEPTH = 'Depth of Hold' = Depth measured from top of ceiling to top of deck beam at the centre line of the vessel at the middle of the length. This depth is measured to the 'tonnage deck,' which is the 'second deck from below.'

Overall Dimensions.

LENGTH = 'Length Overall' = Length measured from the foremost point of the stem or figure-head (but excluding bowsprit if any) to the aft side of the taffrail, or cruiser stern.

BREADTH = 'Breadth Extreme' measured over the outside of the shall plating or belting or paddle sponsons, if any.

DEPTH = Depth from underside of keel to top of deck amidships.

Drafts of a Ship.

The DRAFT d of a ship is the distance of the lowest point of the keel (or of the line of the keel produced) below the water line.

The MEAN DRAFT is half the sum of the drafts at the forward and after perpendiculars or at the stem and the stern post.

The MOULDED DRAFT δ is the distance of the top of the keel (or of the line of the top of keel produced) below the water line.

The LIGHT DRAFTS of a vessel are the drafts forward and aft when the vessel is floating complete, with water in boilers, condensers and/or water in engine cooling systems, but without crew, fuel, cargo, stores, water, or other loads.

The LOAD DRAFT of a vessel is the mean draft of the vessel when floating complete and ready for sea and with all fuel, cargo, passengers, crew, baggage, stores, spares, water, and other loads. The maximum load draft is determined by the freeboard tables of the Board of Trade; for passenger vessels the actual draft is limited to that determined by the spacing of the water-tight bulkheads, in accordance with the recommendations of the Convention for the Safety of Life at Sea. This draft is called 'The Bulkhead Draft.'

The FREEBOARD of a cargo vessel is the height of the top of the steel deck at side at the middle of the length above the water line. In a passenger vessel the freeboard is measured from a point three inches below the upper surface of the deck at side, at the middle of the water-line length, to the water line. The minimum freeboard allowable for any vessel entering or leaving a British port is fixed by law. It is assigned for each vessel by the Board of Trade or Classification Society, and must be marked upon each side of the vessel amidships by the 'Freeboard Disc' or 'Plimsoll Mark.'

The 'TRIM' or 'LEAD' of a vessel is the difference between the drafts forward and aft. It is said to be 'by the head' or 'by the stern' according as the draft forward or the draft aft is the greater. When the drafts forward and aft are equal, the vessel is said to be 'on even keel.' 'Change of Trim' is the arithmetical sum of the changes of draft forward and aft produced by the shifting of any weight.

Displacement.

The DISPLACEMENT of a vessel is the amount of water displaced by the underwater portions of her hull. It may be stated either by volume or by weight. The weight of the water displaced by a floating vessel is equal to the total weight of the vessel and her contents. If the vessel be floating in salt water, her displacement in tons Δ is equal to $\frac{1}{35}$ of her volume of displacement, V, in cubic feet. If in fresh water, $\Delta = \frac{V}{35 \cdot 9}$.

The TONS PER INCH of a vessel at any draft is the number of tons weight which must be placed on board in order to increase the mean draft one inch.

$$\text{Tons per inch, in salt water} = \frac{\text{area of water plane}}{420} = \frac{A}{420},$$

$$\text{Tons per inch, in fresh water} = \frac{A}{430 \cdot 8}.$$

where A is the area of the water plane in square feet.

Coefficients of Displacement.

The BLOCK COEFFICIENT, or *Coefficient of Fineness*, C_b, is the ratio of the immersed volume of displacement to the volume of the circumscribing parallelepipedon.

$$\text{Block coefficient} = C_b = \frac{V}{L\,B\,\delta} = \frac{35\,\Delta}{L\,B\,\delta} \text{ in salt water} = \frac{35 \cdot 9\,\Delta}{L\,B\,\delta} \text{ in fresh water,}$$

where V = Immersed volume of displacement in cubic feet,
Δ = Displacement in tons,
L = Length on load water-line, in feet,
B = Moulded breadth in feet,
δ = Moulded draft in feet.

Builders of cargo vessels usually estimate block coefficient with reference to length B.P., breadth over shell plating, and draft including flat plate keel.

The PRISMATIC COEFFICIENT, C_p, is the ratio of the immersed volume of displacement to the volume of the circumscribing cylinder having the same form of cross-section as the vessel below the water-line amidships.

$$\text{Prismatic coefficient} = C_p = \frac{V}{La} = \frac{35\,\Delta \text{ or } 35 \cdot 9\,\Delta}{La},$$

where a is the immersed area of the midship section to outside of frame in square feet.

The MIDSHIP SECTION COEFFICIENT, C_m, is the ratio of the area of the midship section up to the water-line to the area of the circumscribing rectangle.

$$\left.\begin{array}{r}\text{Midship section coefficient}\\ \text{or midship area coefficient}\end{array}\right\} = C_m = \frac{a}{B\,\delta}.$$

From the above values for the three coefficients it is easily seen that

$$C_p \times C_m = C_b.$$

Mr. W. H. Riddlesworth, M.Sc., gives the following approximate formulas connecting draft, displacement, and block coefficient:—

Let α **= water plane area coefficient =** $\frac{A}{LB}$

β **= block coefficient.**
δ **= moulded draft.**
Δ **= displacement in tons.**

Then, approximately,

(1) The water plane area coefficient is equal to one-third plus two-thirds of the block coefficient,
i.e. $$\alpha = \tfrac{1}{3} + \tfrac{2}{3}\beta$$

(2) The change (c) in block coefficient per foot change in draft is equal to the complement of the block coefficient divided by three times the draft,

i.e. $$c = \frac{1-\beta}{3\delta}.$$

(3) The complement of the block coefficient varies inversely as the cube root of the draft,
i.e. $$1-\beta = \text{constant} \div \sqrt[3]{\delta},$$

the constant being determined for each ship from any one known draft δ_0 and the corresponding block coefficient β_0; constant = $\sqrt[3]{\delta_0}\,(1-\beta_0)$.

The above three relations have been found to be both useful and reliable in practical work.

Curves of Displacement, etc.

As the draft of a vessel changes, the displacement, area of water plane, coefficients, etc., also vary, and it is customary to record these changes by means of curves whose vertical abscissæ represent draft, while the horizontal ordinates represent areas, displacements, coefficients, etc.

The ordinate of the curve of displacement at any draft is proportional to the area of the curve of water plane areas up to that draft.

Centre of Buoyancy.

The CENTRE OF BUOYANCY, B, at any draft, is the centre of gravity of the volume of displacement at that draft. The vertical position of the centre of buoyancy is the same as that of the centre of gravity of the curve of water plane areas, and its longitudinal position is the same as that of the centre of gravity of the prismatic curve. (See next page.)

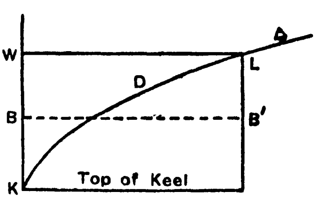


FIG. 1.

To find the height, K B, of the centre of buoyancy above the top of keel, draw the line BB′, across the displacement curve so that the area of the rectangle WB′ is equal to the area W L D K above the displacement curve.

MORRISH'S RULE.

Very approximately $KB = \frac{5}{6}\delta - \frac{V}{3A}$;

where K B = height of centre of buoyancy above top of keel, in feet
δ = moulded draft, in feet;
V = displacement, in cubic feet;
A = area of the water plane, in square feet.

$\frac{V}{A}$ is called 'Rankine's Mean Draft.'

$\frac{KB}{\delta}$ **varies from 0·53 in full form ships to 0·57 in fine forms.**

Wetted Surface.

The WETTED SURFACE or IMMERSED SURFACE of a vessel is the total area of the surface of the shell plating, keel, bilge keels and rudder in contact with the water in which the vessel is afloat.

Approximately,

$$\left.\begin{array}{ll} S = L(1{\cdot}7\,\delta + C_b B) & \text{(Mumford)} \\ S = L(1{\cdot}5\,\delta + 1{\cdot}075\,C_b B) & \text{(Gemmell)} \\ S = 3{\cdot}4\,V^{\frac{2}{3}} + 0{\cdot}5\,LV^{\frac{1}{3}} & \text{(Froude)} \\ S = 15{\cdot}4\sqrt{L\Delta} & \text{(Taylor)} \end{array}\right\} + \text{surfaces of keel, bilge keels, and rudder.}$$

where L = length between perpendiculars, in feet;
δ = moulded draft, in feet;
C_b = block coefficient;
B = moulded breadth, in feet;
Δ = displacement, in tons; V = displacement in cubic feet
S = wetted surface, in square feet.

The second of the foregoing formulæ should be used for vessels of large $\frac{B}{\delta}$ values.

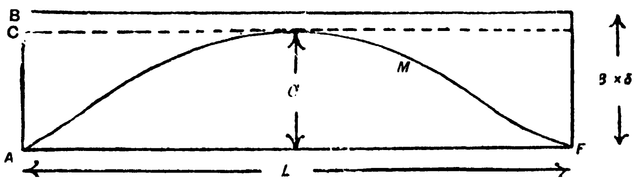


FIG. 2.

The PRISMATIC CURVE or CURVE OF AREAS for any draft is a curve whose horizontal base represents to scale the length of the vessel on load water-line, and whose ordinate at any point is proportional to the immersed area of the corresponding transverse section of the vessel. This curve shows the longitudinal distribution of the displacement, which has an important bearing upon the resistance.

Let A M F (fig. 2) be a prismatic curve, inscribed within a rectangle C F of height representing the area of the midship section, and surrounded by another rectangle B F of height representing the area of the rectangle circumscribing the midship section. Then

Area of prismatic curve is proportional to displacement.

$$\frac{\text{Area of prismatic curve}}{\text{Area of rectangle C F}} = \frac{V}{L \times a} = \text{Prismatic coefficient.}$$

$$\frac{\text{Area of prismatic curve}}{\text{Area of rectangle B F}} = \frac{V}{L \times B \times \delta} = \text{Block coefficient.}$$

$$\frac{\text{Height of rectangle C F}}{\text{Height of rectangle B F}} = \frac{a}{B \times \delta} = \text{Midship section coefficient.}$$

Centre of Flotation.

The CENTRE OF FLOTATION is the centre of gravity of the water plane. The two waterlines defining any small change of trim or heel intersect at the centre of flotation.

Transverse Metacentre.

If a vessel be slightly inclined to one side the centre of buoyancy, B, will move slightly from the middle line towards that side, say to B′, and the vertical line through B′ representing the upward pressure of the water will cut the inclined middle line at the point M called the 'Transverse metacentre.'

$$BM = \frac{I}{V}$$

where,

BM is the height of the transverse metacentre above the centre of buoyancy,
I is the transverse moment of inertia of the water-plane about its middle line,
and V is the volume of displacement.

BM is approximately equal to ·075 to ·085 × breadth^2 ÷ draft, at load draft.

The METACENTRIC HEIGHT, GM or m, in any condition of loading is the vertical distance between the transverse metacentre, M, and the centre of gravity, G, of the whole vessel and her contents. If, in the upright condition, G lies above M the vessel is unstable and will not remain

upright; if G lies below M, the vessel is stable and will return to the upright position if slightly inclined and set free. GM is a measure of the vessel's 'Initial Stability.'

The value of GM can be obtained either by separate calculation of the heights of G and M or by an inclining experiment.

Inclining Experiment.

In carrying out an 'Inclining Experiment' on a vessel, to determine the height, KG, of the centre of gravity, the following procedure should be observed :—

Have the vessel free from wind, tide, current, transverse mooring ropes, gangways, and particularly from loose water or other weights liable to move when the vessel is inclined. Have no men on board except those engaged on the experiment. Read the drafts carefully at each end from a small boat. Move any known weight w (enough to produce an inclination of about 2°) transversely across the deck through a total distance, a, and observe the corresponding deflection d, of a simple pendulum of length l, suspended anywhere on board. Several shifts of the weights are usually made to each side of the ship and the mean value of d taken. From the draft readings and the curves of displacement and transverse metacentres we can obtain the value of Δ, and of KM.

Then GM as inclined $= \frac{w\,a\,l}{\Delta\,d}$, w and Δ being in tons, GM and a in feet, and l and d in inches

and KG as inclined = KM − GM.

These results apply only to the vessel in the inclining condition. To obtain the metacentric height for any other condition it is necessary to make suitable corrections for the different amounts and positions of the various items of loading and for the new values of Δ and KM.

It is usual to estimate the value of GM for the following conditions :—

Light Condition.—Ship and machinery complete, boilers and condensers and/or engine cooling systems full, but no fuel, cargo, passengers, crew, baggage, stores, spares, fresh or salt water, or reserve feed-water on board.

Docking Condition.

Load Conditions.—Fully loaded condition; ship and machinery complete, boilers and condensers and/or engine cooling systems full, and with all fuel, passengers, crew, baggage, stores, spares, fresh and salt water, reserve feed-water, and holds filled with homogeneous cargo of such density as to bring the vessel to her load draft.

Spent Condition.—Same as full load condition, but with all fuel, water, and stores consumed.

Ballasted Condition.—Same as spent condition, but with water-ballast tanks filled.

Any other special conditions likely to arise in practice.

Rolling Period of a Vessel.

In Smooth Water.

K = Radius of gyration of the vessel and all weights on board, about a longitudinal axis passing through her centre of gravity.

m = Metacentric height;

2 T = Time of double roll, say from port to starboard and back again, in seconds;

g = Force of gravity = 32·2.

then

$$T = \pi\sqrt{\frac{K^2}{g\,m}} = \cdot554\,\frac{K}{\sqrt{m}},$$ neglecting water and air resistances.

The period 2T of a double roll varies from about 3 to 30 seconds; in large Atlantic liners it is about 20 seconds.

Stability.

The INITIAL STABILITY of a vessel is the measure of the tendency of the vessel to return to the upright if she is forcibly inclined to a small angle, θ, of not more than about 10°.

Initial Stability or Righting Moment at the inclination $\theta = RM_\theta = \Delta\, m\, \theta$ foot-tons;

where,

Δ = displacement in tons;

m = metacentric height in feet;

θ = angle of heel in circular measure
= inclination in degrees ÷ 57·3.

CURVE OF STABILITY.—When a vessel is inclined to any angle θ (fig. 3), the centre of buoyancy moves to one side (B to B′). The upward force of buoyancy, Δ, now acts through B′, while the weight acts downwards through G, the centre of gravity. These two forces form the righting couple, whose arm is GZ (the righting arm) and whose moment is W.GZ

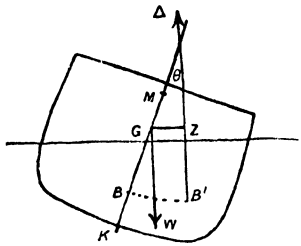


FIG. 3.

(the righting moment). When θ is small, B′Z passes through M, as already stated, and $GZ = m\,\theta$; at larger angles GZ must be separately calculated, and it is customary to obtain its value at each 10° of inclination and to connect the series of righting arms by a curve which is called the curve of stability (fig. 4).

OR is called the 'Range of Stability,' H the 'Maximum Righting Arm' occurring at some

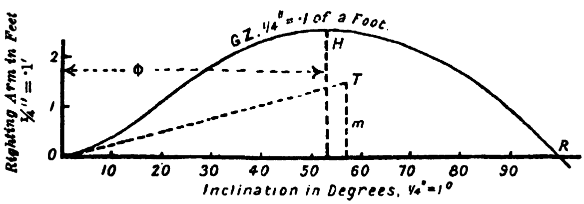


FIG. 4.

angle ϕ, while the initial slope of the curve is measured by the metacentric height, m, set up at 57·3° ($\theta = 1$), as shown in the figure, where OT is tangent to the curve at the origin O. (The scales marked are those to which the curves are usually drawn, but of course do not apply to the reduced figure.) The area of the curve of stability up to any angle is proportional to the work which must be done (by wind or other external forces) in order to incline the vessel to that angle. The work so done is called 'Dynamical Stability.'

Longitudinal Metacentre.

If a vessel be slightly inclined about a transverse horizontal axis, the centre of buoyancy will move longitudinally, and the point in which the vertical line through the new centre of buoyancy cuts that through the centre of buoyancy corresponding to even keel is called the 'longitudinal metacentre.'

The height of the longitudinal metacentre above the centre of buoyancy = LBM = $\frac{I}{V}$, where I is the moment of inertia of the waterplane about a transverse axis through its centre of gravity, and V is the volume of displacement.

The LONGITUDINAL METACENTRIC HEIGHT, LGM, is the height of the longitudinal metacentre above the centre of gravity = LBM − BG = M.

The longitudinal GM is used in calculating changes of trim. If any weight, w, be moved longitudinally through a distance, a, the 'moment to change trim' is wa.

The 'moment to change trim one inch' or 'inch trim moment' = ″M = $\frac{\Delta M}{12L} \doteqdot^{*} \frac{I}{420L}$

* $\doteqdot$ is the symbol for 'approximately equal.'

Approximately, $''M = \frac{30\,t^2}{B}$

where, $''M$ = inch trim moment, in foot tons.
Δ = displacement, in tons.
L = length between draft marks, in feet.
t = tons per inch.
B = moulded breadth, in feet.
I = longitudinal moment of inertia of water plane.

The *change of trim* in inches due to any moment $w\,a$ is $\frac{w\,a}{''M}$. Of this total change of trim half is usually added to the draft at the end towards which the weight w was moved, and half deducted from the draft at the other end.

If instead of merely moving a weight already on board, we desire to find the alteration of drafts due to placing a new weight on board, first imagine the weight to be placed at the centre of flotation F. This will increase the drafts both forward and aft equally by an amount $\frac{w}{t}$. Then imagine the weight moved longitudinally to its assigned position and calculate the change of trim. Then combine these changes of drafts with the additional immersions $\frac{w}{t}$ and so find the total changes of draft forward and aft.

Curves connecting the mean draft with the position of the centre of flotation, the tons per inch, and the inch trim moment should be prepared to facilitate such calculations.

For large additions of weight the above method is not accurate, and it is necessary to prepare a complete list of all the items making up the displacement, together with their moments forward or aft of midships. Then

$$\frac{\text{Total moment}}{\text{Total weight}} = G\otimes \text{ or } \otimes G = \text{distance of centre of gravity aft or forward of midships.}$$

From a curve connecting draft with the distance of the centre of buoyancy for even keel from midships, we can find the value of $B\otimes$ or $\otimes B$, and hence the value of GB or BG, that is the longitudinal distance of the centre of gravity aft of or forward of the centre of buoyancy. The moment to change trim is then $\Delta \times GB$, and the change of trim $\frac{\Delta . GB}{''M}$.

Weights.

The total weight or displacement of a merchant vessel may be subdivided as follows :—

DISPLACEMENT
- Light Weight
  - Hull
    - Steel
    - Wood and fittings
  - Machinery
    - Engines
    - Boilers
- Deadweight

The *Weight of Steel* includes :—

Steel and iron plates and bars of all kinds.
Structural forgings and castings, whether of steel or of bronze.
Pillars.
Steel masts.

The *Weight of Wood and Fittings* includes :—

Carpenter work.
Joiner work.
Plumber work.
Wood masts and spars.
Upholstery.
Boats.
Smithwork.
Deck machinery and piping.
Electric lighting installation.
Refrigerating machinery.
Heating installation.
Ventilation.
Galley outfit.
Paint.
Cement.
Insulation.
Rigging and sails.
Canvas covers and awnings.
Sheet iron work.
Fixed ballast.
Deck Coverings. Etc.

The *Weight of Engines* includes :—

Main propelling engines.
Shafting and propellers.
Condensers and circulating pumps.
Air-pumps and bilge-pumps.
electric generators.
Sanitary and fresh-water pumps.
Floors, ladders, and gratings in engine spaces.
Feed water.
Oil in forced lubrication system and drain tank.
Piping in engine-room and tunnel.
Water in condensers and piping.
Tools, etc.
Blowers, silencers.
Air bottles.
Water in engine cooling systems.

The *Weight of Boilers* includes :—

Main boilers and donkey boiler.
Uptakes and funnels.
Fans and fan engines.
Piping in boiler rooms.
Feed-pumps.
Fuel pumps.
Hot water in boilers.
Floor-plates, ladders, and gratings in boiler-rooms.
Tools.
Etc.

Deadweight includes :—

Spare gear.
Crew and effects.
Passengers and baggage.
Coal in bunkers and on firebars.
Fresh water for ship's use.
Salt water for ship's use.
Reserve feed water.
Water ballast.
Bilge water.
Cooling water.
Mails and cargo.
Oil fuel in tanks and pipes.
Stores and provisions.
Bedding and napery.
Cutlery and crockery.
Glass and silverware.
Weights added by the owner.
Etc.

Hull weight may be estimated either in detail or as a whole. Detail calculation requires that complete plans of the vessel are available, together with reference lists giving weight-rates or actual weights of all varieties of fittings. An accurate record of the finished total weights of the various items of the outfit of previous completed vessels is of great value in this connection.

For preliminary work, where a rapid estimate of the weights is required, the method of coefficients is largely relied on. In this method the whole weight of the hull, or of certain sectional portions thereof, is compared with some arbitrary parameter or parameters of simple form, depending upon the principal dimensions of the vessel or of her sectional parts. For each vessel completed an accurate record of these weights is filed, and the values of the 'Coefficients of Weight' $\left(=\frac{\text{weight}}{\text{parameter}}\right)$ worked out. Then for any proposed new vessel the weight of the hull, or of any of its parts, is found approximately by multiplying the proper parameter by a suitable coefficient chosen from among those of similar finished vessels.

The most commonly used parameter is the *Cubic Number*, which is the product of the Length, Breadth, and Depth, divided by 100.

Cubic number $= N = L\,B\,D \div 100.$

where L = Length between perpendiculars, in feet,

B = Breadth moulded, in feet,

D = Depth moulded, in feet, taken to the highest deck to which the shell plating extends continuously or almost continuously throughout the whole length of the vessel;

then,

$$\text{Weight of Hull, } H = N \times \text{Coefficient of Weight of Hull}$$
$$= N \times C_h$$

Similarly,

$$\text{Weight of Steel, } S = N \times C_s$$

$$\text{Weight of Wood and Fittings, } W = N \times C_w.$$

The weights of deck machinery, refrigerating machinery, insulation, etc., may either be included in the wood and fittings coefficient or separately estimated.

The following table gives average values of these coefficients :—

Type of Vessel.	C_s	C_w.	C_h.
Torpedo Boat Destroyer	·24	·10	·34
Cruiser	·27	·10	·37
Battleship	·33	·12	·45
River Paddle Steamer	·19	·12	·31
Channel Steamer	·20	·16	·43
Coasting Steamer	·32	·20	·52
Cargo Tramp	·35	·10	·45
Intermediate Liner	·42	·17	·59
Fast Mail Liner	·38	·19	·57
Oil Tanker	·38	·09	·47

Other parameters sometimes used for estimating the weight of wood and fittings are

$$(L \times B) \text{ and } L\,(B + D)$$

Weight of Machinery.

In making preliminary estimates, the weight of machinery can be taken as proportional to the horse-power, the number of horse-power per ton of total machinery weight being recorded for all vessels completed and tried. A more accurate method is to record separately the weight of engines and the weight of boilers per horse-power, and to employ these rates when estimating the weights of proposed machinery of similar type. The weight of reciprocating engines may be taken as proportional to the product of the square of the diameter and the square root of the stroke of the high-pressure cylinder. For cylindrical boilers a suitable weight parameter is the product of boiler volume and square root of working pressure. For the greatest accuracy the weights of both engines and boilers must be made up in detail from the drawings and the detailed weight records of previous jobs. The machinery weight for Diesel-engined vessels varies considerably according to the type of engines installed. Generally the weight of the engine(s) proposed can be obtained from the makers. The complete weight of an installation of Diesel machinery is about $1\frac{1}{4}$ to $1\frac{1}{2}$ times the weight of the main propelling Diesel motors.

Deadweight.

The total deadweight may either be specified by the owner or must be made up in detail from the particulars of the service to be performed.

Spare Gear.—Make up the weight in detail from the specified list of spares or allow a total approximate amount by reference to the actual weights carried by similar steamers. The amount may vary from about two tons in river paddle steamers and fast Channel steamers to about 120 tons in an Atlantic liner.

Crew and Effects.—Allow about one ton per ten men.

Passengers.—Allow one ton for every fifteen persons in ordinary cases and for every twenty persons in excursion crowds including women and children.

Baggage.—For the hand baggage of excursion passengers allow about one ton for every 250 persons. For long voyages allow about three tons for every ten persons.

Fuel in Bunkers = horse-power at sea, including that of auxiliaries × pounds of fuel consumed per horse-power per hour ÷ 2240 × distance between fuelling stations farthest apart ÷ service speed, + fuel for galleys etc. + allowance for use in port + a margin of 10 to 20 per cent. for emergencies.

Coal on Bars.—Allow about 35 lbs. per square foot of grate area.

Fresh Water.—Allow about 3 tons per day for every 10 cabin or tourist passengers and about one ton per day for every 10 third-class or steerage passengers and crew.

Salt Water.—For sanitary purposes a tank containing from ½ ton to 5 tons according to the extent of the sanitary accommodation.

Reserve Feed Water.—Allow about 4 tons per 1,000 H.P. per 24 hours for short voyages and about half as much in vessels where distilling apparatus is carried.

Provisions.—Allow about one ton per day for every 75 cabin and tourist passengers and about one ton per day for every 250 third-class or steerage passengers and crew.

Water Ballast.—As may be required for purposes of immersion, stability or trim during the voyage. Say one-half of weight of consumables.

Bilge Water.—Is more or less inevitable, but should not amount to much if proper care in pumping is continually taken. Allow about one-fifth of the vessel's 'tons per inch.'

Mails, cargo, and stores.—As required by owners.

Bedding and napery, *Cutlery and crockery*, *Glass and silverware* } Together about one ton for every 15 persons on board.

Resistance of Ships.

The total resistance to the ahead motion of a ship is made up of four parts :—

1. 'Skin-friction resistance,' or 'skin resistance.'
2. 'Wave-making resistance,' or 'wave resistance.'
3. 'Eddy-making resistance,' or 'eddy resistance.'
4. Air resistance.

2 and 3 are usually considered together under the name 'residuary resistance,' or 'wave resistance' merely, since eddy resistance should be very small. An allowance of 10 to 15 per cent. is usually added to the resistance of the 'naked' hull to account for the resistances of bossings, bilge-keels, rudder and other appendages, and air resistance.

HORSE-POWER.

The horse-power required to overcome any of these resistances is proportional to the resistance multiplied by the speed, and is called the 'effective horse-power.' It is from about ·45 to ·65 of the corresponding indicated or shaft horse-power developed within the vessel, the remainder of the power developed being absorbed by engine and shaft friction and losses in the propeller. The ratio of effective horse-power to indicated or shaft horse-power is called the 'propulsive efficiency,' or 'propulsive coefficient.'

$$e = \frac{\text{E.H.P.}}{\text{I.H.P.}} \text{ or } \frac{\text{E.H.P.}}{\text{S.H.P.}}$$
$$= \text{from ·45 to ·60.}$$

Skin Effective Horse-Power.

Skin E.H.P. is directly calculable :—

$$\text{Skin E.H.P.} = \text{E}_s = \frac{\text{S}}{100} \times \text{F}_{10} \times \left(\frac{\text{K}}{10}\right)^{2·83}.$$

where,

S = wetted surface in square feet.
F_{10} = Skin E.H.P. per 100 square feet at 10 knots.
K = speed in knots.

The following table gives the values of F_{10} and of $\left(\frac{\text{K}}{10}\right)^{2·83}$ for various lengths and speeds :—

L.	F_{10}.	L.	F_{10}.	K.	$\left(\frac{\text{K}}{10}\right)^{2·83}$	K.	$\left(\frac{\text{K}}{10}\right)^{2·83}$
50	1·999	450	1·832	10	1·00	18	5·28
100	1·918	500	1·826	11	1·31	19	6·15
150	1·888	600	1·816	12	1·68	20	7·11
200	1·873	700	1·807	13	2·10	25	13·37
250	1·863	800	1·798	14	2·59	30	22·40
300	1·854	900	1·790	15	3·15	35	34·65
350	1·846	1000	1·782	16	3·78	40	50·56
400	1·839			17	4·49		

From these figures curves may be drawn upon squared paper from which intermediate readings can be obtained.

Wave Effective Horse-Power.

The wave effective horse-power is not directly calculable, although several empirical formulæ and curves of coefficients have been published.

Taylor gives :—

$$\text{Wave E.H.P.} = \text{E}_w = \text{B}_e \frac{\Delta^{\frac{2}{3}}}{\text{L}} \times ·0030707\ \text{K}^5$$

Where B_e = from ·35 to ·55, according to the degree of fullness of the vessel and her ratio of length to breadth.

The most accurate method of estimating E_w is by using the results of model experiments or of actually tried vessels, and Froude's 'Law of Comparison.' The Law of Comparison may be stated thus :—If two vessels of exactly similar form, but of different size, are run at speeds proportional to the square roots of their lengths, then their wave effective horse-powers will be proportional to the products of their displacements by their speeds. If, then, we are designing a vessel n times as long as a model or similar smaller ship whose speeds and corresponding wave E.H.Ps. we know, the following ratios which various elements of the design bear to corresponding elements of the prototype may be used :—

Linear dimensions	n	Speeds	$\sqrt{n}$
Surfaces	n^2	Wave resistances	n^3
Displacements	n^3	Wave E.H.Ps.	$n^3\sqrt{n} = n^{3·5}$

Speeds related as above are called 'Corresponding Speeds,' and the ratio $\frac{\text{K}}{\sqrt{\text{L}}}$ of speed to square-root of length is called the 'Speed-Length Ratio.'

Model experiments give us the total E.H.P. at any speed. From this the wave E.H.P. can be obtained by separately calculating the skin E.H.P., as above described, and deducting it from the total. The wave E.H.P. for the full-size ship will be $n^{3\frac{1}{2}}$ times that of the model, and the skin E.H.P. for the full-size ship can be separately calculated. The sum of these two results gives the

total E.H.P. for the full-size ship. It is then finally necessary to assume a suitable value for e, the propulsive coefficient, and calculate the I.H.P. or S.H.P. by dividing the total E.H.P. by e. If we are designing from the trial results of a similar completed vessel, the procedure is slightly different :—

I.H.P. or S.H.P. of Type Ship $\times e$	=	E.H.P. of Type Ship	at K knots.
$\frac{S}{100} \times F_{10} \times \left(\frac{K}{10}\right)^{2 \cdot 83}$	=	Skin E.H.P. of Type Ship	
Difference	=	Wave E.H.P. of Type Ship	
Multiply by		$n^{3\frac{1}{2}}$	
and so obtain		Wave E.H.P. for Design	at $\sqrt{n}$ K knots.
$\frac{S'}{100} \times F'_{10} \times \left(\frac{K'}{10}\right)^{2 \cdot 83}$	=	Skin E.H.P. for Design	
Sum	=	Total E.H.P. for Design	
Divide by		e	
To obtain		I.H.P. or S.H.P. for Design	

where,
S = wetted surface of type ship, in square feet,
$S' = n^2 S$ = „ design „
K = speed of type ship, in knots,
$K' = \sqrt{n}\,K$ = corresponding speed of design, in knots,
F_{10} = skin E.H.P. per 100 square feet at 10 knots for type ship,
F'_{10} = „ „ „ for design.

D. W. Taylor's book, 'The Speed and Power of Ships,' which contains a series of curves based on model experiments and covering a large range of proportions of dimensions and speeds, will be found to give reasonably accurate estimates of effective horse-power.

For fast, fine vessels, the paper 'On some Results of Model Experiments,' by R. E. Froude, published in the 'Transactions of the Institution of Naval Architects for 1904,' may be advantageously consulted. Many other experimental results have now been published in Great Britain, America and other countries.

Admiralty Coefficients.

The 'Admiralty Coefficient of Performance' $= \dfrac{\Delta^{\frac{2}{3}} K^3}{P} = C$

where,
Δ = displacement, in tons,
K = speed, in knots,
P = I.H.P. or S.H.P.

Conversely $P = \dfrac{\Delta^{\frac{2}{3}} K^3}{C}$, so that if C is known or can be correctly assumed, P can readily be calculated. In similar ships at 'corresponding speeds $\Delta^{\frac{2}{3}} K^3$ varies as $(n^3)^{\frac{2}{3}} (\sqrt{n})^3$, that is, as $n^{3\frac{1}{2}}$. Hence, if P were also to vary as $n^{3\frac{1}{2}}$, the values of C would remain the same. And also, if for any given vessel P varied as K^3, C would be the same for all speeds. For these reasons C is sometimes called the 'Admiralty Constant,' but as the above relations are theoretical assumptions only, and C is found to vary from speed to speed and from ship to ship, it is better to use the term 'Admiralty Coefficient.'

Although wave-making power follows the law of comparison, and therefore varies as $n^{3\frac{1}{2}}$ skin-friction power does not obey the same law and only increases at the lesser rate

$$n^2 \frac{F'}{F} (\sqrt{n})^{2 \cdot 83} = \frac{F'}{F}\, n^{3 \cdot 415}.$$

And hence larger ships have relatively less power and larger values of C than smaller vessels. In the case of large vessels, also, a lesser proportion of power is lost in overcoming internal friction and eddy-making, and a further increase of C is the result.

In spite of its known defects, the 'Admiralty Coefficient' is largely used for estimating horsepower, the assumed value being based upon that for a similar, or nearly similar, completed vessel at 'corresponding' speed, and suitable allowances made for variations in size and proportions.

If the prismatic coefficient of the type ship differs from that of the proposed vessel, we may still find suitable values for O by modifying the 'corresponding speed' of the type ship, thus:—

$$K = K'\sqrt{\frac{L}{L'} \cdot \frac{1 - O_P}{1 - O'_P}}$$

where K, L, and O_P are the corresponding speed, length between perpendiculars, and prismatic coefficient respectively for the type ship, and K', L', and O'_P the desired speed, length, and prismatic coefficient for the proposed vessel.

Admiralty Coefficient applied to Fuel Consumption.

A variation of the Admiralty Coefficient much used by superintendent engineers, when comparing performances at sea, consists in substituting the daily fuel consumption in tons for horse-power in the formula $O = \frac{\Delta^{\frac{2}{3}} K^3}{P}$. We thus get:—

$$\text{Fuel Coefficient of Performance} = \frac{\Delta^{\frac{2}{3}} K^3}{T}$$

Where Δ = mean service displacement in tons,
K = mean service speed in knots,
T = daily fuel consumption in tons.

Rudders.

The area of a vessel's rudder is determined by dividing the product of her length and load draft by about 30 to 40 in the case of warships, 60 to 70 for fast liners, 90 to 100 for slow cargo tramps. The most efficient angle is about 35° from the middle line, at which angle the total pressure upon the rudder in tons is about $\frac{AK^2}{700}$ for single screw ships, $\frac{AK^2}{1000}$ for twin screws, where A is the rudder area in square feet and K the ship's speed in knots. The centre of pressure at 35° is found by assuming that the pressure on each elementary horizontal strip is applied at a point four-tenths of the breadth of the strip from its leading edge, summing the moments for each strip, and dividing by the total area.

Let h be the distance of the centre of pressure from the axis in inches, and P the total pressure upon the rudder in tons. Then T, the twisting moment in inch-tons, = Ph, and

$$D = \text{diameter of stock, in inches} = \sqrt[3]{\left(\frac{16\,T}{\pi F}\right)}$$

where F = maximum allowable shearing stress in material in tons per square inch.

$$\text{If } F = 5{\cdot}1,\ D = \sqrt[3]{T}$$

In practice the diameters of the rudder stocks of all classified vessels are determined by the rules of the Classification Societies.

Estimate Sheet.

When making preliminary estimates of dimensions, weights, power, etc., the following form will be found convenient:—

Number of Inquiry. Type of Vessel. Owners. Date.

Conditions.

Length × Breadth.

Type. Length × Breadth × Depth. Cubic Number.

S Steel Weight. Steel Coefficient.
W Wood and Outfit Weight. W. & O. Coefficient.
M Machinery Weight. H.P. per Ton.
D Deadweight.

$$\frac{\text{Displacement}^{\frac{2}{3}}\ \text{Speed}^3}{\text{Admiralty Coefficient}} = \text{Horse Power.} \qquad \frac{\text{Length} \times \text{Breadth} \times \text{Draft full Keel}}{35} = \text{Block Number}$$

Block Coefficient

Displacement

For example, in figures:—

Inquiry No. 42. Cargo and Passenger Steamer. Messrs. Brown and Black. **3.3.47.**

To carry 6000 tons and steam 13¼ knots on 24 ft. draft. Class, 100 A1 at Lloyd's. Sixty first-class. 100 third-class.

	19200			
T.S.S.	400 × 48 × 30		5760	
S.	2200	·382		
W.	850	·147		
M.	750	4·54		24′ 0′
D.	6000			2″ keel

$$\frac{9800^{\frac{2}{3}}\ 13\tfrac{1}{4}^{3}}{313} = 3400 \text{ I.H.P.} \qquad \frac{400 \times 48 \times 23'\ 10''}{35} = 13080$$

$$\begin{array}{r} 13080 \\ \cdot 75 \\ \hline 9800 \end{array}$$

These figures are arrived at by a process of trial and error-correction until a satisfactory balance is reached between weights and displacement, block coefficient and speed length ratio, horse-power and weight of machinery. The block coefficient should not exceed 1·06 minus half the speed-length ratio. (Speed in knots divided by square root of length in feet.)

Elements of Propulsion.

If a vessel (without propeller) moving at a speed of V feet per minute experiences a total resistance of R pounds, a force R must be applied continuously by means of a thrust or tow-rope pull in order to overcome the resistance and maintain the speed. The power required is RV foot-pounds per minute and is called the Tow Rope Power or Effective Power.

$$\text{Effective Horse Power, E.H.P.} = \text{R.V.} \div 33000 = \cdot 00\dot{3}0\dot{7}\ \text{RK}.$$

If the thrust is delivered by a screw propeller working behind the ship, it is found that the action of the propeller increases the resistance. This increase is called the 'Augment of Resistance.' T, the total thrust of the propeller, must therefore be equal to R + Augment = R + A, and T − A = R. The augment of resistance can therefore be regarded as a deduction from the thrust: it is alternatively called the 'Thrust Deduction' and is written A = *t*T, where *t* is the 'Thrust Deduction Fraction.'

$$T - tT = T(1 - t) = R$$

$(1 - t)$ is called the Thrust Detection Factor.

If a vessel is being towed at a speed of K knots, the skin friction sets the adjacent water in motion, and at the stern there will be found a 'wake' current moving in the same direction as the ship at some speed W knots. The stern, and with it the propeller, will therefore be moving at a speed K − W relatively to the wake water, and this speed is called 'the Speed of Advance.' Froude compares wake speed with speed of advance, and writes $W = \omega(K - W) = \omega K'$.

Wake fraction	$\omega = \dfrac{W}{K - W} = \dfrac{W}{K'}$
Wake factor	$1 + \omega = \dfrac{K}{K - W} = \dfrac{K}{K'}$
Wake per cent.	100ω.

Taylor compares wake speed with ship speed, and writes—

$$W = \omega K$$

Wake fraction	$\omega = \dfrac{W}{K}$
Wake factor . . .	$1 - \omega = \dfrac{K - W}{K} = \dfrac{K'}{K}$
Wake per cent.	100ω.

If a propeller having a pitch P and making N revolutions per minute is working behind a ship and producing a thrust T with a torque Q in the shafting just forward of the propeller, the ship's speed being K and the speed of advance of the propeller K′ = K − W, the work obtained will be RK (E.H.P.) and the work expended 2πNQ, which we can call the 'Received Horse Power Behind' R.H.P.$_b$. The efficiency η is therefore $\dfrac{\text{E.H.P.}}{\text{R.H.P.}_b} = \dfrac{\text{RK}}{\text{R.H.P.}_b} = \dfrac{T(1 - t) \times K'(1 + \omega)}{\text{R.H.P.}_b}$.

TK′ is called the 'Thrust Horse Power' (T.H.P.) and the product $(1 - t)(1 + \omega)$ the 'Hull Efficiency,' being the product of one factor $(1 - t)$ representing the effect of the propeller on the hull and another $(1 + \omega)$ representing the effect of the hull on the propeller. If we write—

$$\text{Hull Efficiency} = h = (1 - t)(1 + \omega)$$

we have

$$\text{Screw Efficiency Behind} = \eta = \frac{\text{T.H.P.}}{\text{R.H.P.}_b} h.$$

Now if the same propeller be mounted on a shaft alone, without ship or model, and run in open water at the same revolutions N, and its speed of advance adjusted until it gives the same thrust

PARTICULARS OF NOTABLE LINERS.

Nationality.	British.				American.	Dutch.		French.	German.	Italian.
Owners.	Canadian Pacific Steamships Ltd.	Cunard Steamship Company.		Royal Mail Lines.	United States Lines Co.	Holland-America Co.	N.V. Stoomvaart Maatschappij, Nederland.	Cie de Navigation Sud-atlantique.	Norddeutscher Lloyd.	Italia.
Name	Empress of Britain	Queen Mary	Mauretania	Andes	America	Nieuw Amsterdam	Oranje	Pasteur	Bremen	Conte di Savoia
Built at	Clydebank	Clydebank	Birkenhead	Belfast	Newport News	Rotterdam	Amsterdam	St. Nazaire	Bremen	Trieste
Year	1931	1936	1939	1939	1940	1938	1939	1940	1930	1932
Length between perpendiculars	730′ 0″	965′ 0″	732′ 0″	630′ 0″	660′ 6¼″	700′ 0″	605′ 8″	656′ 2″	887′ 11″	777′ 0″
Breadth moulded	97′ 6″	118′ 0″	89′ 1″	83′ 0″	93′ 3″	88′ 0″	83′ 6″	88′ 0″	101′ 8″	95′ 9½″
Depth "	52′ 3″	74′ 7″	56′ 6″	47′ 6″	55′ 0″	55′ 0″	63′ 8″	47′ 10″	55′ 6″	62′ 10″
Gross tonnage	42,348	80,744	35,739	25,689	27,000	36,287	20,017	29,253	51,656	48,502
Load Draught	32′ 0″	38′ 10′	30′ 10½″	29′ 3″	32′ 8¾″	31′ 6¾″	28′ 11 11/16″	31′ 6″	33′ 7½″	30′ 6″
" Displacement (tons)	—	—	—	—	35,440	36,235	22,000	—	54,800	59,998
Passengers: Cabin	465	776	486	324	543	568	283	287	800	866
" Tourist	260	784	390	204	418	455	283	126	800	412
" 3rd Class	470	795	502	—	241	209	174 (inc. 4th)	338	600	922
Main Engines	Turbines	Turbines	Turbines	Turbines	Turbines	Turbines	Diesel	Turbines	Turbines	Turbines
Power transmission	Reduction Gears	Reduction Gears	Reduction Gears	Reduction Gears	Reduction Gears	Reduction Gears	Direct	Reduction Gears	Reduction Gears	Reduction Gears
Propeller shafts	4	4	2	2	2	2	3	4	4	4
Revolutions (per min.)	150 inboard 200 outboard	180	140	140	128	136	145	200	180	—
Boilers	8 Yarrow 1 Johnson 2 Cylindrical	24 Yarrow 3 Cylindrical	6 Yarrow	3 Babcock-Johnson 1 Howden-Johnson	6 Babcock & Wilcox	6 Yarrow	2 La Mont	10 Penhoët	20 Water-tube	10 Yarrow 3 cylindrical
Working press. lb./sq. in.	425	400	425	425	425	550	100	441	330	450
Heating surface (sq. ft.)	106,393	234,000	64,500	33,735	63,000	51,000	—	—	161,000	134,200
Superheat (degrees F.)	723	700	725	750	725	750	—	725	700	725
Air Heaters	Fitted	Fitted	Fitted	Fitted	Fitted	Fitted	—	Fitted	Fitted	Fitted
Draught	Forced	Forced	Forced	Forced	Forced	Forced	Forced	Forced	Forced	Forced
Fuel	Oil	Oil	Oil	Oil	Oil	Oil	Oil on Exhaust Gas	Oil	Oil	Oil
Horse-power (designed)	62,500	158,000	—	30,000	34,000	34,000	37,500	62,000	92,500	100,000
Speed on trial (knots)	25·5	Over 31 on service	25	23	24·7	22·8	26·3	—	28·5	29·5
Designed service spd. (knts.)	24·0	28½	23·0	21·5	22·0	20·5	21	25	26·25	26·25

T, we can fairly assume that the speed of advance so found must be the same as the speed of advance when in place behind the ship. We are thus able to determine the wake speed. The torque is not necessarily the same as when behind the ship, though it is usually not very different, so we call the power in the shaft 'Received Horse Power in the Open,' R.H.P.$_o$, and the 'Efficiency in the Open,' $\odot$, is $\dfrac{TK'}{\text{R.H.P.}_o} = \dfrac{\text{T.H.P.}}{\text{R.H.P.}_o}$.

The ratio $\dfrac{\text{R.H.P.}_o}{\text{R.H.P.}_b}$ is called the 'Relative Rotative Efficiency,' r.

Finally, therefore, we can write—

$$\eta = \frac{\text{T.H.P.}}{\text{R.H.P.}_b}h = \frac{\text{T.H.P.}}{\text{R.H.P.}_o} \times \frac{\text{R.H.P.}_o}{\text{R.H.P.}_b}h = \odot rh,$$

or screw efficiency behind = screw efficiency in open × relative rotative efficiency × hull efficiency.

S.H.P., the 'Shaft Horse Power' (or B.H.P., the 'Brake Horse Power'), is the power in the shafting just abaft the engine. The shaft line efficiency l is the ratio $\dfrac{\text{R.H.P.}_b}{\text{S.H.P.}}$. The mechanical efficiency of the engine m is the ratio of S.H.P. to the indicated horse power (I.H.P.).

$$\therefore\ \text{S.H.P.} = m \times \text{I.H.P.}$$

$$\text{R.H.P.}_b = l \times \text{S.H.P.} = l.m.\text{I.H.P.}$$

$l \times m$ is the 'Transmission Efficiency,' $f = \dfrac{\text{R.H.P.}_b}{\text{I.H.P.}}$

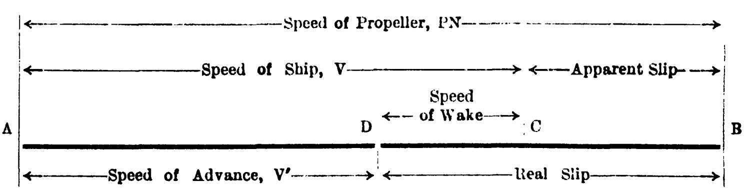
The overall propulsive efficiency or propulsive coefficient, ϵ,

$$= \frac{\text{E.H.P.}}{\text{I.H.P.}} = \frac{\text{E.H.P.}}{\text{R.H.P.}_b} \times \frac{\text{R.H.P.}_b}{\text{I.H.P.}} = \eta f = \odot rhf.$$

η is sometimes called the 'Quasi-Propulsive Coefficient.'

A propeller having a pitch P and making N revolutions per minute would advance a distance PN if there were no slip. On account of slip, however, it actually advances a lesser distance V' called the 'speed of advance.' The difference is called slip, so that slip = PN − V', and the 'slip ratio' $s = \dfrac{PN - V'}{PN} = 1 - \dfrac{V'}{PN}$. If the thrust is T and the torque Q, the efficiency $= \dfrac{TV'}{2\pi NQ} = \dfrac{TP(1-s)}{2\pi Q}$.

When the propeller is attached to a ship its apparent speed of advance is the ship's speed K knots or V feet per minute, where V = 101⅓K. The apparent slip is PN − V. On account of the presence of the wake stream, however, the true speed of advance of the propeller through the water surrounding it is K − W(= K') knots or V' (= 101⅓K') feet per minute, and the real slip is PN − V'. Let AB represent to scale the 'Speed of the Propeller':



AC the speed of the ship V, and CD the speed of the wake.

$$\text{Then apparent slip ratio} = \frac{CB}{AB}$$

$$\text{Real slip ratio} = \frac{DB}{AB}$$

$$\text{Froude's wake fraction} = \frac{DC}{AD}$$

$$\text{Taylor's wake fraction} = \frac{DC}{AC}$$

Law of Comparison for Model Propeller Experiments.

If a model propeller $\frac{1}{n}$th of the size of the ship's propeller is run at $\sqrt{n}$ times the revolutions and its speed of advance adjusted so that both screws are working with the same slip ratio, the thrust of the ship's propellers will be n^3 times that of the model, the torque n^4 times, the powers $n^{3\frac{1}{2}}$ times, and the efficiencies equal.

	Model Propeller		*Ship's Propeller*
Diameter	d		$D = nd$
Pitch	p		$P = np$
Revolutions	r		$R = n^{-\frac{1}{2}} r$
Speed of Advance	v'		$V' = n^{\frac{1}{2}} v'$
Slip Ratio, s	$1 - \frac{v'}{pr}$	=	$1 - \frac{V'}{PR}$
Thrust	t		$T = n^3 t$
Torque	q		$Q = n^4 q$
THP = U	$\frac{tv'}{33000}$		$\frac{TV'}{33000} = n^{3\frac{1}{2}} \frac{tv'}{33000}$
RHP = P	$\frac{2\pi rq}{33000}$		$\frac{2\pi RQ}{33000} = n^{3\frac{1}{2}} \frac{2\pi rq}{33000}$
Efficiency	$\frac{tv'}{2\pi rq}$	=	$\frac{TV'}{2\pi RQ}$
ρ_U (Rho-U)	$\frac{ru^{\frac{1}{2}}}{v'^{2\frac{1}{2}}}$	=	$\frac{RU^{\frac{1}{2}}}{V'^{2\frac{1}{2}}}$
ρ_P (Rho-P)	$\frac{rp^{\frac{1}{2}}}{v'^{2\frac{1}{2}}}$	=	$\frac{RP^{\frac{1}{2}}}{V'^{2\frac{1}{2}}}$
δ (Delta)	$\frac{dr}{v'}$	=	$\frac{DR}{V'}$

The last three functions are the same for all similar propellers working at the same slip ratio; ρ_U and ρ_P depend on revolutions, power and speed, which are given quantities in the initial stage of propeller design. The results of systematic propeller experiments are usually plotted on some function of Rho as base, since this is independent of diameter. For each of a series of pitch ratios, a curve of efficiency and a curve of some function (such as δ) involving the diameter are plotted, from which the most suitable diameter and pitch ratio can be found for the given power, revolutions, and speed of advance. Taylor has published four ' Rho-delta ' diagrams obtained by combining and averaging his own propeller experiments with those of Froude, Durand, and Schaffran. These were reproduced in *The Shipbuilder*, January 1925.

SECTION XXIX

PART II

MARINE ENGINES (STEAM)

(Revised by A. W. Davis, B.Sc.)

BASES OF ENGINE AND BOILER DESIGN.

Reciprocating Engines—Cylinder Ratios.

On account of its relatively low efficiency the reciprocating steam engine is seldom employed to-day except in certain cargo vessels and ships built for special purposes in which simplicity is of paramount importance. Triple expansion engines are the most popular for these services and the quadruple expansion engine, originally developed for large powers with minimum inefficiency, is now almost obsolete. The following table of cylinder area ratios is in accordance with modern practice for various types of marine engines:—

TABLE SHOWING CYLINDER AREA RATIOS.

Type of Engine.	Boiler Pressure. Lbs. per Sq. In.	Ratio of Cylinder Areas.			
		L.P. / H.P.	2nd M.P. / H.P.	1st M.P. / H.P.	H.P.
Quadruple expansion	200 to 250	8 9·5	4·0 4·5	2·0 2·1	1 1
			M.P. / H.P.		
Triple expansion	160 to 220	6·5 to 8·5	2·5 to 2·9		1
Compound engines, screw	100 to 140	4 to 5	—		1
,, ,, paddle	90 to 130	3·5	—		1

Mean Pressures and Expansion of Steam.

The total power developed in an engine may be varied by altering the point at which steam is cut off in the H.P. cylinder, and generally provision is made to enable the cut-off in each cylinder to be independently adjusted if necessary to equalise the powers developed in each cylinder. For the maximum designed power the cut-off in the H.P. cylinder takes place at from ·65 to ·75 of the stroke, and the total number of expansions of the steam is

$$= \frac{\text{Volume of L.P. cylinder}}{\text{Volume of H.P. cylinder at point of cut-off}};$$

$$\frac{\text{Ratio of } \dfrac{\text{L.P. cylinder area}}{\text{H.P. cylinder area}}}{\text{Cut-off in H.P. cylinder (expressed in fraction of stroke)}}$$

The total expansion of the steam is usually considered with clearance neglected and neglecting also the fact that release takes place before the end of the stroke. In estimating the mean pressure obtainable in any multiple-stage expansion engine it is referred to the L.P. cylinder, the mean pressure thus obtained being that necessary if the whole of the power were developed in the

L.P. cylinder. The actual referred mean pressure is, therefore, in the case of a triple expansion engine equal to

$$\text{Mean pressure in L.P. cylinder} + \frac{\text{Mean pressure in M.P. cylinder}}{\text{Ratio of L.P. to M.P. cylinder area}} + \frac{\text{Mean pressure in H.P. cylinder}}{\text{Ratio of L.P. to H.P. cylinder area}}.$$

The theoretical referred mean pressure may be calculated, assuming that the steam is moderately moist, the curve of expansion being a hyperbola. It depends upon the designed number of expansions and the absolute admission pressure of the steam.

Let Pa = absolute initial pressure,

Pm = absolute theoretical referred mean pressure,

r = total number of expansions,

$\frac{1}{r}$ = cut-off, or reciprocal of r,

then

$$Pm = Pa\,\frac{1 + \text{hyp. log. } r}{r}.$$

PRESSURE OF STEAM IN CYLINDERS OF COMPOUND AND TRIPLE-EXPANSION ENGINES.

The values of $\frac{1 + \text{Hyp. Log. } r}{r}$ for Various Expansions are given on page 73.

The initial pressure, Pa, should be taken 5 to 10 lbs. below the absolute boiler pressure to allow for the drop between boilers and engines, and the back pressure, which must be deducted from Pm as found above, should be taken at not less than 4 lbs. per square inch absolute.

Diagram Factor.

Owing to the losses which occur while the steam is expanding in the various cylinders the actual referred mean pressure obtained from indicator diagrams is less than the theoretical referred mean pressure, and consequently the latter must be multiplied by a diagram factor which takes into account the losses due to clearance, condensation, radiation, friction of ports, compression, drop in pressure between the cylinders, and other losses. The value of this factor is obtained by experience, and, when possible, should be derived from engines similar to the one projected, and working under similar conditions. The values for various types of engines are approximately:—

Quadruple-expansion engines	·63 to ·68
Triple-expansion engines	·6 to ·65
Compound engines	·63 to ·70

When cylinders are jacketed the diagram factors may be increased by about 5 per cent.

The power is usually divided equally among the cylinders of an ordinary compound, triple, or quadruple-expansion engine, and if N is the number of cylinders and P the referred mean pressure (*i.e.* theoretical mean pressure × diagram factor) the mean pressure in the various cylinders of a triple-expansion engine, for example, will be:—

$$\text{L.P. cylinder mean pressure} = \frac{P}{N}$$

$$\text{M.P. ,, ,, ,,} = \frac{P}{N} \times \frac{\text{L.P. cylinder area}}{\text{M.P. cylinder area}}$$

$$\text{H.P. ,, ,, ,,} = \frac{P}{N} \times \frac{\text{L.P. cylinder area}}{\text{H.P. cylinder area}}$$

In the case of four-crank triple-expansion engines the power developed in each L.P. cylinder is usually five-eighths that of the H.P. or M.P. cylinder.

Piston Speed.

The mean piston speed is usually reckoned in feet per minute and is equal to stroke in feet × 2 × revolutions per minute.

The power which may be obtained from a given weight of engine varies almost directly with the piston speed ; the wear and tear of the engine, however, is increased with the higher speeds on account of the increased inertia stresses involved.

Piston speed may be increased either by increasing the stroke and keeping the revolutions constant or by increasing the revolutions, keeping the stroke constant.

By adopting the former method the cost, dimensions, and weight of the engines are increased, whilst by adopting the latter method there is practically no increase in any of the items mentioned. It is evident, therefore, that in cases where weight and space are limited high piston speed must be obtained by high revolutions; these have a limit, however, on account of the stresses due to the inertia of the reciprocating parts.

The piston speed and revolutions of engines fitted in various types of vessels are as shown in the following table :—

Table showing Piston Speed and Revolutions of Reciprocating Machinery for Various Types of Steamships.

Description.	Number of Cranks.	Piston Speed in Feet per Minute.	Revolutions per Minute.
Cargo vessels	4	650 to 800	70 to 100
	3	550 „ 650	60 „ 100
Paddle steamers	3	550 „ 650	45 „ 55

(Also see remarks about propeller revolutions for geared turbines.)

Calculation of Cylinder Diameters.

In determining the diameter of cylinders required for an engine of a given I.H.P., the type of vessel for which the engine has to be designed to a certain extent determines the piston speed and cylinder ratios, due consideration being given to the boiler pressure, the number of expansions being increased as higher boiler pressures are adopted.

Having decided the piston speed and the mean pressure referred to L.P. cylinder as already described, the area of the L.P. cylinder may be calculated, and from the cylinder ratios decided upon the areas of the remaining cylinders are determined.

Let piston speed = S feet per minute;

referred mean pressure = P lbs. per square inch,

then,

$$\text{Indicated H.P.} = \frac{\text{Area of L.P. cylinder} \times P \times S}{33{,}000}$$

or

$$\text{Area of L.P. cylinder} = \frac{\text{I.H.P.} \times 33{,}000}{P \times S};$$

$$\text{Area of M.P. cylinder} = \text{Area of L.P. cylinder} \times \text{Ratio of } \frac{\text{M.P. cylinder area}}{\text{L.P. cylinder area}};$$

$$\text{Area of H.P. cylinder} = \text{Area of M.P. cylinder} \times \text{Ratio of } \frac{\text{H.P. cylinder area}}{\text{M.P. cylinder area}}.$$

Combined Turbo-Reciprocating Machinery.

The reciprocating engine is not suitable for obtaining work from very low pressure steam owing to practical difficulties such as the large cylinder diameters and port areas which the greater volume of steam would require. A turbine can make use of this low pressure work and much has been done in an effort to obtain a satisfactory combination of the systems whereby a low pressure turbine utilises the exhaust from the reciprocating engine.

Early experiments in which the turbine was direct coupled to a third propeller shaft only met with partial success on account of the low efficiency of a fast running propeller. It was fitted in several large liners but is now entirely displaced by the simpler and more efficient geared turbine installation.

Various systems have since appeared in which the turbine has been coupled to the reciprocating engine shafting. The principal difficulty of such an arrangement lies in reconciling the irregular angular velocity of the reciprocating engine with the steady velocity of the turbine and when the coupling is made through gearing a flexible member must be introduced. In the Bauer Wach system which has double reductional mechanical gearing, the flexibility is provided by a hydraulic clutch which also acts as a ready means of disconnecting the turbine for astern running. Where electric transmission is employed the exhaust turbine drives a generator, and a motor is directly coupled to the propeller shafting, thus dispensing with any gearing.

Another system embodies the Gotaverken exhaust steam, turbo compressor, fitted in conjunction with a reciprocating engine, and it has proved highly successful. In principle, exhaust steam from the L.P. cylinder drives a turbine which is coupled to a multi-stage rotary compressor. The exhaust from the H.P. cylinder is led to the compressor and is raised in pressure before entering the M.P. receiver.

The advantage of these arrangements lies in the facility with which an exhaust turbine can be installed in combination with the reciprocating machinery of existing vessels. Usually when a conversion of this nature takes place, other improvements are also incorporated with the reciprocating engine and boilers and a reduction in fuel consumption of about 20 per cent. is obtained.

TURBINES.*

The steam turbine is the mode of propulsion adopted for the largest passenger vessels and for all high-powered naval craft. It is also widely employed for intermediate and cross channel passenger vessels and for cargo liners. In the earlier stages of its adoption, when the turbine was directly coupled to the propeller shafting, the high revolutions necessary to obtain suitable blade speeds for low steam consumption without abnormally increasing the size and weight of the turbines entailed small propellers which were inefficient except for fast vessels, and even in these the turbines were run at an uneconomically low speed in order to improve the propellers.

The introduction of gearing between the turbines and propellers permitted the turbine and **propeller designers to select the revolutions most suitable for their individual purposes so as to obtain the maximum efficiency of turbine and propeller respectively. The revolutions of the most efficient propeller for a vessel depend on horse-power, speed and form of vessel, and should be worked out in conjunction with experimental results such as those published by Taylor and Froude, but the following method gives very satisfactory results for turbines, gearing and propellers:—**

If

R = revolutions of propeller per minute;
SHP = shaft horse-power of each shaft;
V = speed of vessel in knots on trial;

then,

$$R = C\sqrt{\frac{V^4}{SHP}}.$$

VALUES OF C.

—	Single Reduction Gearing.	Double Reduction Gearing.
Merchant vessels—ocean	7 to 8	5 to 6
" " channel	9 " 10	5 " 6
Battleships and aircraft carriers	7 " 8	—
Cruisers	9 " 10	—
Destroyers	7 " 8	—

Geared Turbines.

For many years the single reduction geared turbine unit was most generally adopted for both merchant and naval practice. In merchant practice it was usual to fit two turbines in conjunction with each propeller shaft for cross channel vessels where the propeller shaft revolutions were naturally fairly high but for ocean-going vessels it was more customary to employ three turbines per shaft, the steam passing in series from one to the other. To-day the double reduction gear has evolved into a unit of high reliability and for ocean-going vessels with relatively low propeller shaft revolutions it is common to employ such units in co-operation with two turbines per shaft working in series. A similar trend is to be found in naval construction.

For many years two principal types of turbines held the field, namely, the Parsons reaction turbine and the Brown Curtis impulse turbine. It is now quite general for high-pressure turbines to be of the impulse type and low pressure of the reaction type. Designs are prepared and developed and researches are progressed by the marine engineering industries' Research Association —Pametrada (Parsons and Marine Engineering Turbine Research and Development Association).

The tendency to increase boiler steam pressures and temperatures still persists, leading to economy which makes turbine machinery a successful rival to oil engines between powers of 3,000 s.h.p. per shaft and 10,000 s.h.p. per shaft, above which limit the turbine alone holds the field.

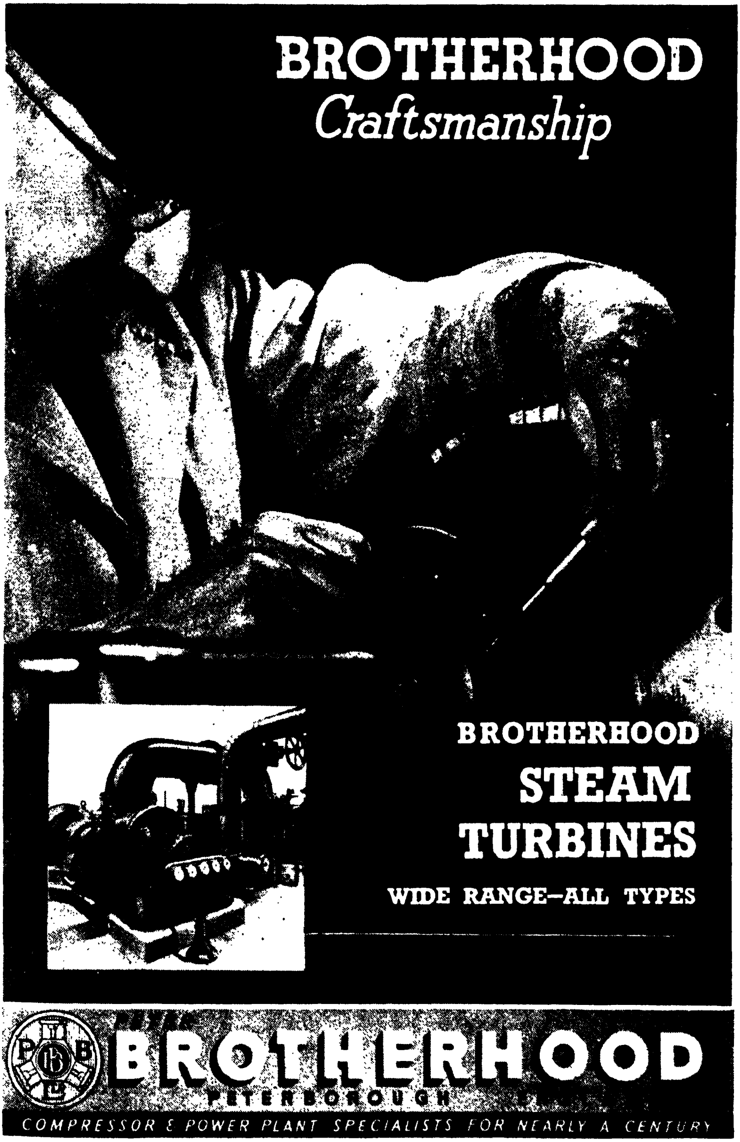
Turbine casings are of cast steel or fabricated construction where subjected to steam temperatures above 450° F., while for lower temperatures cast iron is usually employed. Blading material is usually of stainless iron or Monel metal, the latter often being regarded as more suitable for low temperature turbines.

Reduction Gear for Propulsion.

The single reduction type of gear consists of a large gearwheel having double helical teeth coupled to the propeller shaft and driven by two or more pinions coupled to the turbine through sliding couplings comprising toothed and sleeved members so arranged as to take up axial movement during the expansion of the turbine rotor under steam.

In the case of double reduction gears, a high-speed or primary train forms an addition to the unit described in conjunction with each of the turbine shafts.

*** See also 'Steam Turbines,' p. 103.**


BROTHERHOOD
Craftsmanship
BROTHERHOOD
STEAM
TURBINES
WIDE RANGE—ALL TYPES
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COMPRESSOR & POWER PLANT SPECIALISTS FOR NEARLY A CENTURY

F. 290

TURBINE GEARING.

Single Reduction.

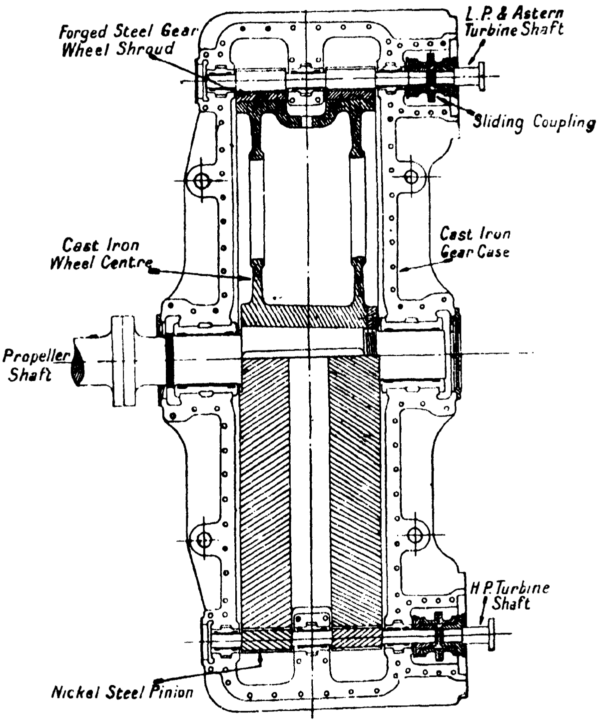


FIG. 1.—Plan with Top Casing removed.

The gearing is completely enclosed in a cast iron or fabricated steel casing with which the thrust block is sometimes arranged in integral construction.

Sprayers are provided to enable oil from the forced lubrication system to be directed on to the teeth adjacent to the zones of contact.

Pinions are normally made of 3 per cent. nickel steel, normalised, oil-hardened and tempered, having a tensile strength of about 45 tons per sq. in. and 25 per cent. elongation.

Gear wheel rims are of 0·3 per cent. carbon steel, normalised and having a tensile strength of about 35 tons per sq. in., with 26 per cent. elongation.

Extreme accuracy in cutting the gear teeth is essential and the hobbing machines employed for the work are the result of extensive and costly research both in design and construction, guidance on this subject being available in the form of a British Standard Specification.

The highest quality of gears are finished by a post hobbing process involving either lapping of the teeth with abrasive or the newly perfected system of selective shaving whereby, in the case of an accurately hobbed gear, the surfaces in contact on even a broad faced gear may be increased to as much as 90 per cent. of the total available tooth service.

The permissible pressure per inch width of gear face is a function of the diameter of the pinion and for normal ratios of face width to pinion diameter may be defined by the expression

$$150 D_e^{\frac{2}{3}} \text{ where } D_e = \text{pinion P.C.D.} \left(\frac{R}{R+1}\right)$$

where R in turn equals the gear ratio. The pitch of the teeth employed is a function of the pressure per inch width and may be given approximately as—

$$\frac{\text{Pressure in lbs. per inch width of tooth}}{2{,}000} \text{ inches.}$$

The normal depth of tooth employed varies between 20 per cent. and 45 per cent. in excess of the British Standard Form of involute rack for spur gears and the helical angle employed varies between 15° and 30°.

Electric Transmission of Power for Propelling Machinery.

The application of electricity for transmitting power between the prime mover and the propeller may be divided into two classes—viz. Turbo-electric and oil-electric.

TURBO-ELECTRIC DRIVE

The electrical equipment required is an alternating current generator suitable for direct coupling to a high-speed turbine, a motor of suitable speed for direct coupling to the propeller shaft, a direct current exciter or an auxiliary generator, from which direct current can be obtained, and suitable control gear. The whole can be regarded as a reduction gear having a ratio of reduction determined by the relative number of poles on the motor and generator. Thus with a two-pole generator and a sixty-pole motor the ratio of reduction would be 30 to 1, and, as with mechanical gearing, any variation in speed is obtained by varying the speed of the prime mover.

Two types of alternating current motors have been used for driving propellers :—

(a) Induction machines, both slip-ring and squirrel-cage types.
(b) Self-starting synchronous motors.

All things considered, the self-starting synchronous motor is the most suitable for this class of work on account of the large air gap, simplicity of construction, accessibility of windings, high power factor, and efficiency. When starting from rest the synchronous motor functions as a squirrel-cage motor with its field unexcited.

On account of its greater flexibility direct current is preferable for vessels up to about 3,000 s.h.p. on one shaft; beyond this power, owing to the limitations of voltage, the cables become excessively large.

The astern turbine is eliminated entirely in an electrically-driven vessel, power astern being available without reversing the direction of rotation of the turbine.

Manœuvring is very efficient with this type of power transmission, so that perfect speed control in heavy seas is maintained without danger of racing.

DIESEL ELECTRIC DRIVE.

Many types of small coasting vessels and vessels for harbour duties are particularly suited to propulsion by high-speed diesel engines operating through electric reduction drives.

BOILERS.

General Proportions.

Cylindrical Boilers.

Present-day economic conditions have led to the widespread use of oil fuel on shipboard instead of coal where the fuel is to be burned in cylindrical or watertube boilers. In designing cylindrical boilers for sea-going vessels it is, however, still customary to proportion them for coal-burning so that they may be converted at a later date if economic conditions change.

The weight of coal that can be burnt in the furnaces of a marine boiler depends on several factors.

In cylindrical boilers with natural draught some experiments were carried out with varying length of fire bar, and the results showed that the quantity of coal was practically the same with short and long bars and evidently depended more on the cross sectional area of the furnace than the length of grate. The height of the funnel is a controlling factor, and a useful rule for this type of boiler is as follows :—

$$\text{lbs. coal per hour per furnace} = D_m^2 \times \sqrt{F} \times C$$

where D_m = Mean diameter of furnace in feet;
F = Height of funnel in feet above firegrate;
C = Coefficient = 4 Welsh coal on short trial
3·5 „ „ long „
3 „ „ service.

The above values of C are a reasonable average, but may be increased or decreased, depending on the efficiency of the boiler. An extra large ratio of heating surface will take more heat out of the gases, thus decreasing their temperature and reducing the suction which causes the draught and consequently the rate of combustion.

With forced draught the quantity of coal that can be burnt in a cylindrical boiler varies more directly with the area of the grate bars and also depends on the available air pressure.

The air pressure in the ashpit measured in inches of water for various rates of combustion may be taken as follows :—

	Lbs. of Coal per Sq. Ft. of Grate Area per Hour.			
	20	25	30	35
Closed stokehold or closed ashpit with retarders but no air heaters	$\frac{5}{8}$ in.	$\frac{11}{16}$ in.	1 in.	$1\frac{3}{8}$ in.
Closed ashpit with retarders and air heaters	$\frac{1}{2}$ in.	$\frac{3}{4}$ in.	$1\frac{1}{4}$ in.	$1\frac{3}{4}$ in.

These pressures are also approximate as they are affected by the assistance received from the funnel. Due allowance must be made at the fans for the additional resistance of air trunks and the air side of the heaters.

With oil fuel the consumption in lbs. per furnace per hour varies approximately as D_m^2, and usually averages about $25D_m^2$ for Howden draught, but 30 per cent. more has been burnt successfully.

With regard to the heating surface the rate of evaporation in a boiler is the principal factor in determining the efficiency of this plant, but this also is subject to other factors, such as cleanliness, etc.

The rates of evaporation per square foot from and at 212° F. in ordinary practice for long trials are as follows :—

	Evaporation.	Boiler Efficiency.
Natural draught—		
Funnel height 50 feet . . .	5·0	73 per cent.
Howden draught—		
Ocean liners	6·5	80 „
Channel steamers	9·0	75 „
Closed stokehold—		
Channel steamers	9·0	70 „ with retarders
		65 „ without retarders

These rates of evaporation decrease on continuous service with coal owing to the surfaces getting dirty, cleaning fires, etc., and the efficiencies are about 5 per cent. lower. With oil fuel they can be maintained, especially if blowers are fitted for sweeping the tubes.

Cylindrical boilers of the return tube type as shown in figs. 2 and 3 (p. 295) are made single ended or double ended, and in the latter case the combustion chambers of furnaces axially in line with each other, are sometimes fitted with a common combustion chamber. This arrangement reduces the length and weight of the boiler but should not be used when burning coal with Howden or closed ashpit draught unless a high brick wall is provided between the furnaces to prevent flame being blown out of an open furnace door by the pressure in the opposite furnace.

The size and number of furnaces to burn the required quantity of coal and the area of heating surface to give the desired efficiency may be computed from the figures given above.

Furnaces and Firebars.

If forced draught is employed it is not desirable to have firebars longer than 6 ft. on account of the tendency for large quantities of air to pass through the firebars if the fire should burn into holes. With Howden's system 5 ft. 6 ins. is the length of firebar generally adopted. In naval practice with water-tube boilers working at full power under a moderate forced draught of 0·5 in. to 0·75 in. of air pressure, grate bars having a maximum length of 7 ft. are frequently employed; this, however, requires skilful firing, and extreme care to ensure the boiler efficiency being maintained at a maximum. For natural draught the length of grate bar should not exceed 6 ft. 6 ins., but increased boiler efficiency will be attained by using shorter bars, on account of the greater control over the fires. The diameter of furnaces for cylindrical boilers (figs. 7 and 8) should not be less than 32 ins., nor greater than 48 ins., and the number of furnaces which can be fitted in boilers of the following dimensions is:—

Boilers up to 9 ft. 0 ins. diameter	1 furnace	Boilers up to 15 ft. 6 ins. diameter	3 furnaces
,, ,, ,, 13 ft. 0 ins. ,,	2 furnaces	exceeding 15 ft. 6 ins. ,,	4 furnaces

The grate area of a cylindrical boiler is equal to:—Length of firebars × mean diameter of furnace × number of furnaces.

Length of Cylindrical Boilers.

The length of a cylindrical boiler is determined by the length of combustion chamber tubes or the furnace which is practically the same length as the tubes, the length of the combustion chamber and the water spaces at their backs. The combustion chamber measured horizontally should be as long as possible, and usually is about 30 ins. in single-ended boilers and from 50 to 57 ins. in double-ended boilers. The water space between the back of the combustion chamber and the back end-plate of the boiler should be from 5 to 7 ins. at the bottom and wider at the top, the taper being about ½ in. per foot on the combustion chamber back.

Heating Surface.

The heating surface of cylindrical boilers consists of the external surface of the tubes, the length being measured between the tube plates, the surface of the furnaces above the firebar level, and the surface of the combustion chambers. The surface of the front tube plate on the boiler end, although in contact with the hot gases, is not usually reckoned as effective heating surface, and should not be included in the total heating surface of the boiler. In boilers using oil fuel the total surfaces of the furnaces and combustion chambers is included. The heating surface of a water-tube boiler is the total external surface of the tubes in contact with the fire and hot gases.

Diameter of Tubes for Cylindrical Boilers.

In forced-draught boilers the combustion chamber tubes are generally 2½ or 2¾ ins. diameter, and in natural-draught boilers the diameter is 3 ins. to 3¼ ins., the increased sectional area being necessary to reduce the resistance to flow of the products of combustion.

Thickness of Cylindrical Boiler Shells.

The shell plates of cylindrical boilers must be of the following thickness to satisfy the requirements of the revised rules of the Board of Trade, Lloyd's, British Corporation, and Bureau Veritas.

CYLINDRICAL MARINE BOILER.

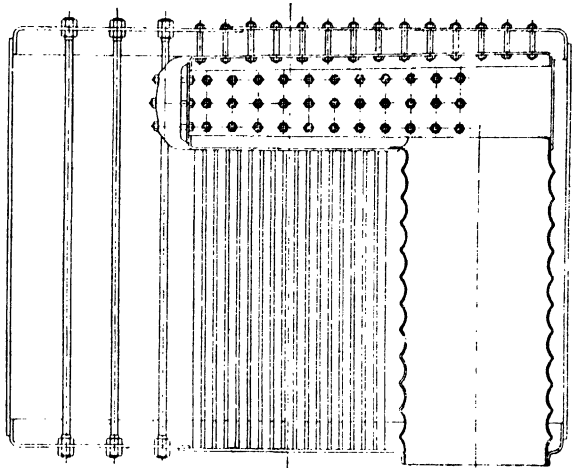


FIG. 3.

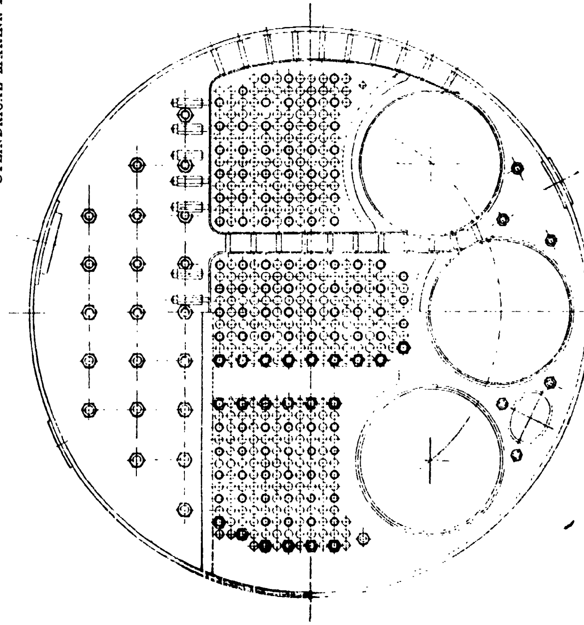


FIG. 2.

The thicknesses shown assume the steel to have a minimum ultimate tensile strength of 28 tons per sq. in., and the longitudinal joint to have double butt straps with an efficiency of riveting equal to 84 per cent.

THICKNESS OF SHELL PLATES IN INCHES.

Internal Diameter.	Working Pressure, lbs. per sq. in.							
	220	210	200	190	180	170	160	150
10 feet	1·027	0·983	0·940	0·896	0·852	0·808	0·765	0·721
11 „	1·124	1·076	1·028	0·980	0·931	0·883	0·835	0·787
12 „	1·220	1·168	1·115	1·063	1·010	0·958	0·905	0·853
13 „	1·316	1·260	1·203	1·146	1·089	1·023	0·975	0·918
14 „	1·413	1·352	1·291	1·229	1·167	1·106	1·045	0·984
15 „	1·509	1·444	1·378	1·312	1·246	1·181	1·116	1·050
16 „	1·606	1·536	1·466	1·396	1·326	1·255	1·186	1·116
17 „	1·703	1·629	1·554	1·479	1·404	1·330	1·256	1·182

Superheaters for Cylindrical Boilers.

The types of superheaters manufactured for cylindrical return-tube boilers consist in principle of a series of loops or elements of solid cold-drawn steel tubing inserted in the smoke tubes of the

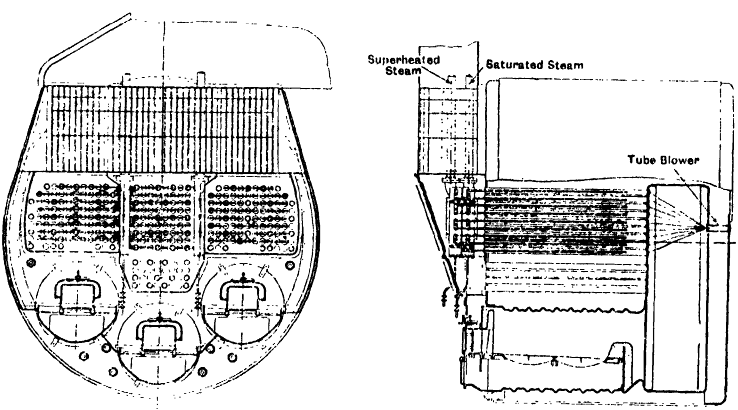


FIG. 4.—**Three-Furnace Boiler, Forced Draught, with Smoke-tube Superheater.**

cylindrical boiler and connected to suitable headers or collectors placed vertically in front of the boiler between the nests of smoke tubes. The saturated steam from the boiler stop valve enters the saturated steam header, the steam circulates through the elements and returns superheated to the superheated steam collector, and thence to the prime mover. The elements are, therefore, in the direct path of the gases flowing from combustion chamber of boiler to smoke-box. (See fig. 4.)

Draught.

The two systems of forced draught most commonly adopted are closed stokeholds and the Howden system with closed ashpits. In the former an air pressure is maintained in the boiler rooms by forcing in air with fans, all openings in the boiler room are closed down with doors, and air locks fitted for access purposes. With a Howden system the boiler room is open,

no air locks being required, and the air is forced by fans over the surface of a nest of tubes, or other form of air preheater situated in the uptakes, through which the products of combustion pass on the way to the funnel. The air is raised to a temperature of about 250° F., or over, by the air heater, and passes through passages on either side of the uptake into a reservoir surrounding the furnace fronts, from which reservoir the air is admitted to the furnaces by valves in the furnace fronts, at the sides and the top, to the ashpit and above the fire respectively. Valves are fitted to control the admission of air above and below the fires: these must be shut before opening the furnace door to fire the boiler, otherwise the flame would be blown out into the stokehold. The proper use of the top valve prevents excessive smoke without reducing the boiler efficiency or the amount of evaporation per lb. of coal. Special strips of twisted steel ribbon called 'retarders' are placed in each boiler tube, causing the hot gases traversing the tubes to take a spiral course and be retained longer in close contact with the tubes than they otherwise would.

The latest type of furnace fronts has the side valves controlled by one handle, and is also fitted with a safety guard, so that the furnace doors can only be opened when the air valves are shut, thereby preventing the possibility of any flame coming back through furnace door when it is opened.

Fronts have also been developed for firing powdered coal, in conjunction with various patentees of powdered fuel firing systems.

On account of the abstraction by the air of what would otherwise be waste heat passing up the funnel a considerable measure of increased boiler efficiency is attained compared with the closed stokehold system, and the uniform condition of furnace temperatures which Howden's system ensures tends to reduce the wear and tear of the boiler.

A more uniform efficiency of furnace can be maintained with forced draught and it may be readily controlled to suit any desired rate of combustion, being quite independent in this respect of prevailing atmospheric conditions.

The original air preheater was of the tubular type, but heaters are now made with surfaces of steel plates arranged to control the flow of air and gases so as to obtain the maximum heat transmission. The plates are assembled in sections to facilitate withdrawal for renewal or repair, and the space occupied by the heater is less than by a tubular heater of the same surface.

WATER-TUBE BOILERS.

The principal advantages of water-tube boilers as compared with cylindrical boilers are :—

(1) Ability to use higher steam pressures (above 250 lbs.).

(2) Rapidity with which steam can be raised.

(3) Capability to endure forcing to a great extent.

(4) Reduced weight.

(5) Reduced space occupied.

Many factors of great complexity enter into the design of a watertube boiler in ensuring satisfactory combustion of the fuel, proper circulation in the tubes and the desired rate of evaporation and superheating. The fundamental criterion of performance is the weight of oil to be burned in lbs. per sq. ft. of furnace radiant heating surface per hour, since this provides, when considered in conjunction with the temperature of the air supply, a measure of the furnace temperature and, consequently, of the resistance of the boiler to brickwork failures. Values adopted range from 30 to 50 for naval vessels and from 7 to 20 for merchant vessels.

The efficiency at which a watertube boiler will operate is largely dependent upon whether an economiser and/or air heater are fitted. If neither of these units is supplied, the efficiency is a function of the generating surface provided but when an air heater and or an economiser is fitted, the generating surface must in any case be so proportioned as to ensure that these units perform their proper function under all conditions of load. The maximum air temperature for supply to the furnace is about 450° F. and the maximum feed water temperature as supplied at the outlet from an economiser should not be above 50° F. less than the saturation temperature at the boiler pressure.

Typical boiler efficiencies are as follows :—

No economiser or air heater	75–80 per cent.
With economiser	84 ,,
With economiser and air heater or with air heater alone, depending upon type of boiler	88 ,,

The surface of the superheater depends upon the steam temperature desired and the temperature of the gases at the position in which the superheater is placed and, as these vary very greatly, no approximate rule can be given.

The principal watertube boilers now in use are the following:—

Yarrow three-drum type.
Yarrow five-drum type.
Fairfield-Johnson controlled superheat.
Babcock & Wilcox header type.
Babcock & Wilcox integral furnace type.
Foster-Wheeler 'D' type.
Foster-Wheeler two-furnace type.

Life of Marine Boilers.

The life of a marine boiler depends to a certain extent on the care and attention bestowed on it during the first few months of its existence, provided that the plates have been properly treated at the building of the boiler—that is, all mill scale removed and the plates kept dry and free from atmospheric corrosion. The mill scale, if not removed, has a very irritant action on the clean metal of the boiler, and is most hurtful to the plates if allowed to remain on them. The greatest care and attention is needed for the first period of working of new boilers to keep the new metal free from corrosion. With water-tube boilers especially, nothing but distilled water should be used, and it is very desirable that air should be excluded from all boilers to prevent corrosion. In order to achieve this result the feed arrangements should prevent the water coming in contact with the atmosphere as much as possible, and a deaerator for removing air from the water incorporated in the system.

Rules and Regulations with Reference to Boilers of Steamships.

Standard Conditions for the Design and Construction of Marine Boilers.

A set of rules has been framed and agreed to by representatives of the Board of Trade, Lloyd's Register of Shipping, the British Corporation for the Survey and Registry of Shipping, and the Bureau Véritas International Registry of Shipping, and recommended for favourable consideration and adoption. An abstract of these rules is given on pp. 33 *et seq.*

Fuel.

Coal.

Coal is burnt by one of three systems:—

Hand firing.
Mechanical stoking,
Pulverised with air injection.

The first system, that of hand firing, is suitable for most boilers, cylindrical or water-tube, and the design of the boilers has been evolved to suit the conditions necessary for this type of combustion.

Mechanical stoking devices have been produced for both cylindrical and water-tube boilers. In some of the arrangments the coal is fed into a hopper by hand and in others by a mechanical elevator. From the hopper it is led into the front of the furnaces and gradually worked towards the back of the furnace by the movement of the bars, which may have a reciprocating motion, or a continuous backward motion, as in the chain type grate.

In water-tube boilers, especially where there is a large radiant surface, and where the service of the vessel involves frequent or unexpected stoppages in the steam supply, provision should be made for damping the fuel in order to prevent excessive blowing off. This is not so important with the cylindrical boiler where the relatively larger steam and water spaces act as a steam accumulator.

Considerable attention has been given to pulverised fuel in recent years, and high boiler efficiencies have been obtained. Special provision must be made with extra large combustion chambers and furnaces in order to give time for complete combustion before the gases pass into the small spaces inside or between the tubes. Arrangements should also be made to prevent the ashes causing inconvenience to passengers.

Oil Fuel.

The advantages of fuel oil as compared with coal are as follows:—

(1) Reduction in weight and space occupied by the boiler installation.

(2) Increased evaporation of water per lb. of fuel in the ratio of about 14 : 9 as compared with good quality coal.

(3) Utilisation of remote parts of the ship and of constricted spaces, such as the double bottoms, for stowage of fuel.
(4) Less manual labour and greater rapidity in stowage.
(5) No reduction in speed due to cleaning fires, difficulty in trimming coal from remote bunkers, or exhaustion of stokers.
(6) Reduced wages bill for boiler room staff, no trimmers, and comparatively few attendants for the burners being required.
(7) A prompt and effective control of the steam output due to the mechanical supply of the fuel, permitting of more rapid changes of speed than is the case with coal.
(8) Absence of smoke if air supply is properly controlled.
(9) No trouble due to accumulation of ashes; ash expellers or ejectors are dispensed with.
(10) In the case of naval vessels, fuelling can readily be carried on at sea.
(11) Cleanliness of ship during fuelling and less inconvenience to passengers.

Both water-tube and cylindrical boilers are now adapted for burning oil fuel, the direct-pressure system being the one most generally in use for naval and mercantile service.

Pressure pumps fitted in the boiler rooms have suctions from the oil tanks, and discharge at a pressure of from 50 to 150 lbs. per sq. in. through cold filters having fine mesh wire gauze grids; from thence the oil passes through a tubular heater, the oil being inside the tubes and heating steam outside. The temperature of the oil is raised to about 150° Fah. in the heater, and before finally reaching the burner it is passed through a second, or so-called hot, filter.

It is necessary to heat the oil in order to reduce its viscosity so that it can be atomised and forced through the fine passages of the burner, which are so designed as to give it a rapid whirling effect due to centrifugal force when liberated through tangential passages at the lip of the burner, and thus reduce it to a fine spray of conical formation. The air supply to the burner passes through the directing slots cut in a steel truncated cone, 12 to 15 ins. diameter and about 16 ins. long. The burner is fitted at the centre of the entrance to the cone, and the admixture of the air passing through the slots with the oil spray from the burner produces almost perfect combustion.

In the case of naval vessels the closed stokehold system is adopted, the air pressure in the boiler rooms being controlled to suit the output of the burners. For merchant steamers the arrangement of burners and air cones is similar to that described above, but the air supply is heated by the waste gases, as in the ordinary Howden forced draught system applied to coal burning. The area of the furnace front limits the number of burners which can be fitted, and it is desirable that the volume of the combustion chamber should be as large as possible. Oil fuel is also burnt in closed stokeholds under forced draught, or under natural draught in the merchant service. (See also pp. 1351 (Vol. I) 312 (Vol. II).)

Fuel Consumption.

Approximate consumptions for all purposes for the various types of propelling machinery on service with medium quality coal and oil are as follows.

Reciprocating Steam Engines with slide valves. Installations with cylindrical boilers, feed heaters, steam-driven auxiliaries, also refrigerators for ship's stores, and an allowance for heating and domestic purposes in the passenger vessels.

Vessel.	I.H.P.	Engine.	Press.	Superheat.	Draught.	Fuel.	Fuel/ I.H.P.
Cargo	3,000	Triple dependent air pumps	180	Nil	Natural	Coal	1·7
					,,	Oil	1·2
					Howden	Coal	1·6
					,,	Oil	1·05
Cargo .	3,000	Reciprocating Steam Engines with cam operated valves	220	150° F.	Howden	Coal	1·4
					,,	Oil	0·9 Equiv. I.H.P.
Cargo	3,000	Combination triple and exhaust turbine	180	Nil		Coal	1·3
						Oil	0·85

Turbines.—Note the consumption for turbines is given per s.h.p., so that in comparing with reciprocating engines the mechanical efficiency of the latter must be taken into account, also any decrease in propulsive coefficient if resulting from higher propeller revolutions.

Type of Vessel.	Screws.	S.H.P.	Turbines.	Trans-mission.	Boilers.	Pres-sure.	Total Temp.	Fuel.	Fuel lbs./S.H.P. Hr. All Pur-poses
Cargo .	Single	3,500	Pametrada	D.R. gear	Cyl.	250	750	Oil	0·72
,,	,,	5,000	,,	,,	W.T.	400	750	,,	0·67
,,	,,	7,000	,,	,,	,,	450	825	,,	0·62
,,	,,	10,000	,,	,,	,,	550	825	,,	0·57
Channel Steamer	Twin	5,000	,,	S.R. gear	,,	300	600	,,	0·76
,,	,,	10,000	,,	,,	,,	350	750	,,	0·68
Passenger Liner	,,	15,000	,,	D.R. gear	,,	450	825	,,	0·64
,,	,,	20,000	,,	,,	,,	550	825	,,	0·60
,,	,,	50,000	,,	,,	,,	600	825	,,	0·56

Auxiliary Machinery

1. Reciprocating-Engined Vessels.

In the smaller vessels, driven by reciprocating steam engines, an arrangement of links and levers on one of the crossheads of the main engines drives the air pump, feed pumps, bilge pump, sanitary pump, evaporator feed pump, and in some cases the condenser circulating pump. This arrangement dispenses with the use of several independent units, which being small are very extravagant in the use of steam.

Provision has to be made in the size of the main engine cylinders for the power necessary to drive these dependent pumps, about 4 per cent. for the air, feed, bilge, and sanitary, and a further 3 per cent. for the circulating pump. In addition to these dependent pumps independent auxiliary feed, general service and ballast pumps are required for use in port and during emergency, the ballast pump also being used for circulating the auxiliary condenser.

A feed heater is very desirable, taking steam from the auxiliary exhaust range supplemented by steam bled from one of the main engine receivers.

There are two types of feed heater, the surface heater fitted on the discharge pipe between the feed pump and the boilers, and the contact heater fitted on the hotwell pump discharge, and placed high in the engine hatch so that the heated water has a gravity head to force it into the main feed pump suction. The main feed pump in this arrangement is independent, and is automatically controlled by a float so as to maintain a constant level of water in the heater. The contact heater facilitates the de-aeration of the feed water, which is very necessary with engine-driven air and hotwell pumps.

A feed filter is also desirable to prevent the lubricating oil used on piston rods and piston rings from entering the boilers.

In vessels on the home trade, where fresh water suitable for boiler feeding is available at frequent intervals, an evaporator is not necessary, but where the supply is unsuitable or the intervals are too great for the fresh-water storage capacity an evaporator should be carried for making up the loss of feed water. The evaporator should have a capacity of 3 tons per day per 1,000 h.p. for make-up feed, in addition to any requirements for distilling fresh water for drinking and other purposes for which latter a distilling condenser should be provided.

In a vessel fitted for oil burning the plant for supplying the fuel should consist of a fuel pump in duplicate, an oil heater, hot and cold oil filters, and probably a transfer pump for filling the oil fuel settling tanks from the bunkers every watch, or for transferring oil when trimming the vessel.

An independent circulating pump is desirable, and should be of the centrifugal type. It is capable of being regulated to give the desired vacuum with varying temperatures of sea water, and thus enables the condensate to be maintained always at the temperature which will give the greatest economy. In single-screw vessels, where there is only one circulating pump, it is good practice to fit two engines, one of which is idle and provides a standby.

It is also good practice to fit two fan engines if there is only one fan in a vessel fitted with forced draught.

In larger vessels the feed pumps are independent, and are fitted in duplicate.

In very large vessels the air, bilge, and feed pumps are all independent, and when the decrease in size and cost of the main engines is taken into consideration, the difference in first cost is not very great, and each auxiliary can be regulated to suit the requisite duty so as to obtain the greatest efficiency.

2. TURBINE-DRIVEN VESSELS.

The auxiliary machinery in connection with a turbine installation generally consists of independent units throughout, and is similar to that for reciprocating machinery with the addition of pumps for supplying oil to the forced lubrication system of the turbine and gear-case bearings and to the sprayers which supply oil to the teeth of the gearing. As a considerable amount of heat is transmitted to the turbine bearings from the hot casings and rotors by convection in addition to the heat generated by friction, oil coolers are necessary.

The oil pumps are always in duplicate, and the oil coolers usually, except in the smallest installations.

The circulating and air pumps are larger than those in a reciprocating installation in order to ensure the higher vacuum, which is an advantage with turbines.

The auxiliary exhaust from reciprocating auxiliaries is usually contaminated with oil from cylinder and piston rod lubrication, and while most of this impurity may be extracted by grease filters it is difficult to remove it completely. Turbines for driving auxiliaries do not require any internal lubrication, and on this account as well as being more suitable for using superheated steam they are now preferred for use with high-pressure water-tube boilers, especially for superheated steam where absolutely clean feed water is so desirable. The turbines being small are not so economical as direct-acting pumps, but since the exhaust is used for feed heating, this is not of great importance. Instead of the small turbines of each independent auxiliary it is more economical to generate the auxiliary power in the comparatively larger turbine of the main electric generating set, and drive the fans and the pumps for bilge, ballast, sanitary, fresh water duties, etc., by electromotors, and even the condenser circulating and forced lubrication pumps may be so driven with advantage.

In some cases the electric generators are driven by Diesel engines, and the fuel for driving the auxiliary machinery is reduced thereby. Where this arrangement is adopted the steam for feed heating is obtained by bleeding from one or more stages of the turbines.

Where cylindrical boilers are fitted the extraction of air and condensate from the condenser is usually performed by a reciprocating air pump with or without the assistance of a steam ejector. In other vessels the condensate is withdrawn by a rotary pump driven by turbine or motor, and the air removed by a two- or three-stage ejector. In this arrangement the feed water system from the condenser to the feed pump suction may be practically enclosed from the atmosphere and aeration of the water minimised to prevent corrosion of the boilers.

3. AUXILIARY MACHINERY—GENERALLY.

The capacities of the various pumps are determined to suit their respective requirements which can be readily determined in the case of feed and air pumps from the power and efficiency of the main machinery. The output of the circulating pump also depends on the size of the condenser and vacuum desired.

The number of the pumps which must be provided for bilge pumping in a passenger vessel is determined by the Board of Trade rules.

The diameter of the bilge pump suction pipe in inches is also determined by Board of Trade :—

$$\text{Diameter} = \sqrt{\frac{L(B + D)}{2{,}500}} + 1;$$

where L = length of vessel in feet.
B = breadth of vessel in feet.
D = moulded depth to bulkhead deck in feet.

The output of the pump in tons per hour must not be less than $3 \cdot 8 \times \text{diameter}^2$.

It may be observed that under conditions specified in their rules one of the bilge pumps is to be an emergency pump of a reliable submersible type, and the source of power should be available at all times and placed above the bulkhead deck. This emergency bilge pump is usually electrically driven, the current being taken from the emergency electric generator worked by an oil engine and placed on one of the upper decks.

It will be noted that in passenger vessels burning oil fuel the Board of Trade require that two or three fire pumps depending on the size of the vessel should be fitted having two-thirds the output of the bilge pumps. Independent bilge pumps would be accepted for this purpose if suitably connected, and not used for pumping oil fuel.

In vessels using oil fuel, leakage from the tanks to the bilge when pumped overboard contaminates the water in harbour, and as there are severe penalties for this offence in many ports, a separating tank is usually fitted on such ships to remove the oil from the bilge discharge. In addition to the purification of the bilge water this arrangement permits the oil to be recovered for use.

Sanitary pumps should have an output of from 50 tons per hour per 1,000 passengers and crew in emigrant ships, to 100 tons per hour in first-class liners, and fresh-water pumps about 20 per cent. of this amount.

The ballast pump, which may be included as a bilge pump for Board of Trade purposes, should have an output of about twice that of the bilge pump.

Where the exhaust from auxiliaries is used for feed heating the cylinders are proportioned to do their work when exhausting against a pressure in the exhaust line of about 5 lbs. gauge, which,

having a saturation temperature of 227° F., enables a feed temperature of 225° F. to be maintained with a contact heater and 205° F. with a surface heater, depending upon the ratio of surface to quantity of feed water.

In the case of important auxiliaries such as the lubricating oil pumps, where temporary stoppage might have serious results, it is good practice to legislate for two units being in service at all times, and instead of fitting two pumps, each capable of supplying the total quantity required, instal three pumps each having a capacity of 60 per cent. of the full duty, and work with two while the third is in reserve.

Auxiliary condensers are fitted principally for use in port, and the cooling surface is determined arbitrarily unless for the use of turbo-generators, where exact conditions of steam consumption are known and a definite vacuum maintained.

In cargo vessels with steam winches, 50 sq. ft. is usually allowed for each winch in the vessel, with additional surface for the electric generator, refrigerator, and pumps used in port. An auxiliary condenser so determined is usually more than ample for other purposes at sea.

If the steam consumption for auxiliaries is known an allowance of the following proportions may be adopted :—

Vacuum	Atmosphere	20"	20"
Sea temperature	85°	70°	85°
Lbs. of steam per sq. ft.	30	30	20
Lbs. of cooling water per lb. steam	20	25	35

It may be observed that the reduction in fuel consumption for all purposes at sea, when the auxiliaries are electrically driven and the generators are driven by Diesel engines, is about 3 per cent., as compared with auxiliaries driven by steam, which is subsequently used for feed heating but the saving in port is considerable, as it enables the boilers to be completely shut down.

Bearings.

The Michell Bearing.

Michell Bearings (thrust and journal) are largely used in modern marine practice, and in passenger and merchant ships the Michell Thrust Block is commonly used, both for reciprocating engines and steam turbines.

A standard thrust block used for light craft of all kinds is shown in fig. 5. This is self-lubricating, and can be arranged upon an extension of the engine-bed or upon a separate seating

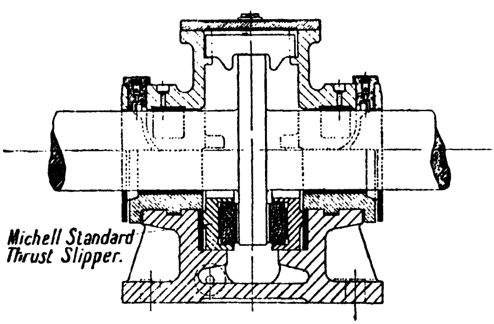


Fig. 5.—Michell Slipper Type Thrust Bearing.

immediately aft of the crankshaft coupling. The oil level in the thrust block is usually kept just above the under-side of the shaft. The bearing surfaces are lubricated by the thrust collar carrying oil upon its edge, and the oil is scraped from this by the scraper at top, thus giving a continuous cascade of oil on the pads. The pads are so pivoted that a film of oil is led into the space between pad and thrust collar, and this oil film can withstand a heavy pressure. A feature of the bearing is that as the speed increases, the pressure per square inch can be increased.

Michell thrust blocks have a single collar shaft with a series of pivoted pads arranged forward and aft, and it will be noticed from the illustration that the pads need only be in the bottom half of the bearing. Thrust blocks, however, can be designed to utilise an existing multi-collar thrust shaft.

Large engines are usually fitted with a thrust block arranged for continuous oil circulation and in these blocks the upper as well as the lower portion of the body sustains thrust load

Oil is introduced into the space behind the pads and compelled to trave radially across the thrust faces. This oil can be cooled and used over and over again. In these blocks the upper and lower parts of the body are keyed together, and liners for adjustment are fitted behind each of the rings which carry the pads.

The Michell principle is also applied to journal bearings, and fig. 6 illustrates one as fitted o a marine oil engine. When speeds are high the ring lubrication is replaced by a continuous

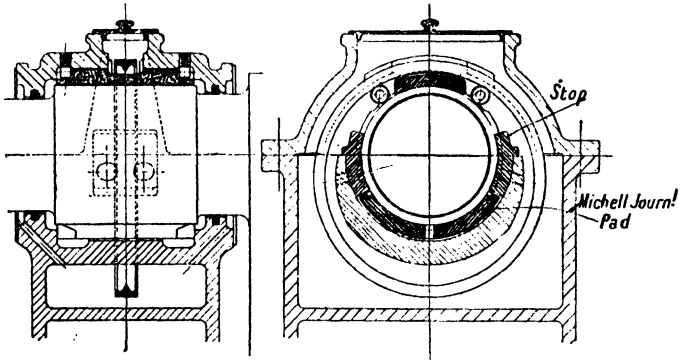


FIG. 6.—Michell Journal Bearing for Marine Oil Engine.

supply of oil from a pump. It will be noticed that the oil is taken on to the shaft by means of a ring, and this ring is always kept in a central position, even if the ship pitches, as the central part of the oiling ring is guided by grooves in rollers arranged upon the shaft.

Considerable economy in length is effected by this bearing, as a pressure of about 500 lbs. per square inch projected journal area is satisfactorily carried at any speed.

The pivoting principle, upon which all Michell bearings work, ensures, perhaps, the nearest approach possible to perfection in lubrication. Astonishingly high pressures are quite safely carried (under test as much as 5 tons per square inch has been successfully imposed), whilst speeds of any magnitude may be run by regulating the quantity of oil supplied. High speed is an advantage, inasmuch as oil film thickness increases with the speed.

SECTION XXIX

PART III

MARINE OIL ENGINES

(By A. P. Chalkley, B.Sc. Editor, 'The Motor Ship.')

The first large ocean-going motor ship, the *Selandia*, was completed in 1912. The war from 1914–1918 retarded further developments, but in 1919 rapid progress in the application of oil engines to ship propulsion was made. The following table shows the gross tonnage and machinery power of ocean-going motor ships built since 1935 :—

MOTOR SHIPS COMPLETED SINCE 1935.

Year.	No.	Gross Tonnage.	I.H.P. of Machinery.
1935	131	782,360	622,380
1936	167	1,075,480	848,860
1937	190	1,169,000	984,000
1938	223	1,461,000	1,198,000
1939	209	1,486,000	1,325,250
1946	110	625,000	572,000
1947	168	810,000	879,000
1948	272	1,250,000	1,100,000

During the period in question, the average tonnage of motor ships under construction throughout the world was 50 per cent. greater than that of steamers, and the last available Lloyd's Quarterly Returns (September 1948) showed that there were steamers of 1,610,732 tons gross and motor ships of 2,558,251 tons gross on the stocks. About 4½ million tons gross of motor ships were on order.

Whilst many different types of marine Diesel engine have been developed, about 90 per cent. of the motor ships built during the past few years have been equipped with six different designs, namely, Burmeister & Wain, M.A.N., Sulzer, Doxford, Werkspoor and Fiat. Since the building of the *Selandia* there has been a progressive improvement in marine Diesel machinery. The average fuel consumption has fallen from 0·42 lb. to about 0·36 lb. per b.h.p. hr., and the weight and space occupied are substantially lower. The *Selandia's* engines weighed 406 lb. per b.h.p., whereas modern two-stroke Diesel machinery does not exceed 130 lb. per b.h.p. Comparing the original *Selandia* with a second ship of the same name built 25 years later, the engine of the new *Selandia* weighs 30 per cent. less, develops 44 per cent. more power and gives the vessel a speed 32 per cent. greater.

In recent years oil engines have largely superseded steam machinery for the propulsion of coasters and other relatively small craft. Whereas with ocean-going ships, the economy in fuel and the increased deadweight capacity have been responsible for the increasing adoption of oil-engined propulsion, with coasters there is the further advantage that Diesel machinery provides greater flexibility of operation for the particular service involved. A coaster calls at a large number of ports with relatively short periods under way, and with an oil-engined vessel of this type the machinery can be shut down immediately the ship enters and started up at a moment's notice when required. Standby losses, which may be heavy in a steamer, are eliminated in a motor ship.

Diesel engines are being employed to an increasing extent in naval vessels. The only examples of motor battleships, however, were three German pocket battleships of the *Deutschland* class,

propelled by the highest-powered Diesel machinery yet constructed. It comprised eight two-acting two-stroke engines of 7,000 b.h.p. driving twin screws through reduction gearing. Apart from submarines, all of which are propelled by Diesel engines, there is an increasing application of this class of plant to auxiliary craft, such as escort and patrol vessels. In America a considerable number of destroyers are equipped with Diesel electric machinery and many submarine mother ships with Diesel generating plant and propelling motors were built during the 1939–45 War. In some of these the total output is 12,000 b.h.p. Even in certain classes of naval, commercial and pleasure craft where considerations of weight have hitherto rendered it necessary for petrol motors to be employed, high-speed Diesel engines weighing comparatively little more than the corresponding petrol type are now gradually being utilised, and, broadly speaking, there is no type of boat or ship requiring machinery from 50 b.h.p. to 50,000 b.h.p. that cannot be propelled by oil engines.

The number and tonnage of motor ships (of over 100 tons gross) in service in the various years from 1935 to 1948 are given in the following table. Later figures are not available :—

TONNAGE OF MOTOR SHIPS IN SERVICE.

Year.	Number.	Gross Tons.
1935	5,511	11,304,691
1936	6,128	12,290,599
1937	6,763	13,748,718
1938	6,912	15,232,953
1939	7,551	16,918,687
1948	7,119	16,789,084

TYPES OF MARINE OIL ENGINE.

All classes of oil engine have been utilised for ship propulsion, namely, four-stroke single-acting and double-acting, and two-stroke single and double-acting, but the four-stroke double-acting design is now no longer built, and the two-stroke engine has gained steadily in favour. In 1931, 46 per cent. of the motor ships built were equipped with two-stroke Diesel machinery, and in 1939 the percentage had risen to 80. The proportion is now approximately the same. In 1947 out of 84 ocean-going motor ships completed (one-half of the world total) 40 were equipped with Doxford type opposed-piston two-stroke engines. Eleven had Sulzer and 9 Harland-B. & W. two-stroke machinery, whilst 9 were equipped with Harland-B. & W. four-stroke engines mostly pressure-charged. Only 5 of the two-stroke engines were of the double-acting type. In Sweden, all of the 28 motor ships built in 1947 were propelled by two-stroke engines.

The utilisation of single-acting two-stroke engines has extended even to the highest powers, and three 12-cylinder Sulzer two-stroke single-acting engines of 12,500 b.h.p. each are installed in the motor ship *Oranje*. Many standard engines are constructed to develop an output of 1,000 b.h.p. per cylinder or more, as the following table shows. All are of the two-stroke design.

SIZES AND POWER OF LARGE MARINE OIL ENGINES.

	Bore. mm.	Stroke. mm.	Speed. r.p.m.	B.H.P. per cylinder.
Doxford . . (opposed piston)	725	2,250	120	1,350
Sulzer . . . (single-acting)	760	1,250	145	1,000
B. & W. . . (double-acting)	620	1,400	100	1,000
Harland-B. & W. . (single-acting)	750	2,000*	110	1,180
Fiat . . . (double-acting)	760	1,200	130	1,400
Kockum-M.A.N. . (double-acting)	720	1,200	115	1,000
Stork . . . (double-acting)	730	1,100	110	1,160

* Including exhaust piston.

All large four-stroke engines built for marine work are now pressure-charged. In some cases the Büchi exhaust gas turbo-charging system is employed, and in others the under piston system is utilised. Details of both of these are given later. It is found in practice, that the additional output thus obtained is at least 30 per cent. on a continuous rating basis, with a corresponding increase in mean effective pressure.

Mechanical injection of fuel is exclusively adopted with all classes of marine Diesel engine, a complete change having been effected in this direction during the past ten years. The elimination of the injection air compressor has represented an improvement of 5 to 7 per cent. in fuel consumption. The pump injection pressure is between 4,000 lb. per sq. in. and 6,000 lb. per sq. in. and there is a separate fuel pump for each cylinder. In some engines, exemplified in the Doxford type, the pumps are grouped together, and in others each pump is adjacent to the cylinder which it supplies, which has the advantage of equalising the length of piping to the various injection valves.

In addition to lower fuel consumption, mechanical fuel injection has the advantage of reduced maintenance charges since there is no injection air compressor, and with this in view, several systems have been developed for converting air injection Diesel machinery to the airless injection type.

Cylinder and piston cooling arrangements differ widely in various engines. In all of the Burmeister & Wain types, lubricating oil is employed for cooling the pistons and the same plan is adopted with the M.A.N. double-acting design and in some of the Sulzer engines. In the Doxford opposed piston unit, distilled water is invariably circulated around the pistons and in many Sulzer Diesel motors salt water is used. For cooling the cylinders and covers, most manufacturers adopt fresh water, and in the case of Doxford engines, distilled water.

Piston speeds vary between 1,000 ft. per min. and 1,200 ft. per min. Particulars of the average mean indicated pressures allowed under ordinary running conditions at sea are given in the table. The relatively high figure permitted for four-stroke pressure-charged engines is notable also the low mean indicated pressure of the two-stroke double-acting unit. Exception should be made, however, of the Burmeister & Wain two-stroke double-acting motor, in which the m.i.p. for normal output may reach about 90 lb. per sq. in. This is due to the fact that there is a 'through' scavenge. Scavenging air enters the cylinder through ports at the centre when uncovered by the piston and is discharged through exhaust valves at the top and bottom of the cylinder respectively. These valves, in the latest design, are of the same diameter as the cylinder, so that no cylinder covers in the ordinary sense are required. The valves are driven by eccentrics from the crankshaft. In all other double-acting two-stroke engines the exhaust gases are discharged through ports, which involve 'loop' scavenging, and with such an arrangement the permissible mean pressure is lower than with the through scavenge.

M.I.P. of Marine Diesel Engines.

Type.	M.I.P. lb. per sq. in.
Four-stroke (pressure-charged)	120–130
Two-stroke (single-acting)	90–95
Two-stroke (double-acting)	70–75

The advantage of the 'through' scavenge is obtained in some single-acting two-stroke engines by the employment of an exhaust valve in the cylinder head. This applies to the B. & W. type, also to the General Motors design, in which each cylinder has four exhaust valves. In the Doxford engine there is also a straight passage for the scavenging air, with the result that a higher mean effective pressure is employed than in the normal single piston engines with 'loop' scavenging. The pressure of scavenging air varies from about 1½ lb. per sq. in. to 2½ lb. per sq. in. In all designs, much experimental work has led to special shaping of the scavenging air inlet ports in order to provide a tangential flow of the air, and the necessary swirl which provides the best degree of combustion when the fuel is injected into the highly compressed air in the combustion chamber. It has been found that the shape of the scavenging air ports is of considerable importance in the attainment of the highest efficiency. In the early days of marine Diesel engine development, the scavenging air for two-stroke machinery in motor ships was usually supplied from separate electrically-driven blowers, and although this system is still sometimes maintained in high-powered vessels, it is more normal for the scavenging pumps, either of the piston or rotary type, to be driven from the engine.

In the general structure of marine oil engines, cast iron framework is usually adopted, sometimes with through bolts extending from the bottom of the bedplate to the top of the cylinders, to take the stresses arising out of combustion. In some recently built engines of large power, however, cast steel columns are adopted and a further development lies in the employment of a welded steel plate construction, of which the Doxford engine is a good example. It is probable that there will be a further development in this direction, in view of the increasing importance now being attached to the application of welding in all engineering industries. Fifteen years' experience with Doxford engines of this type has indicated that no troubles have arisen which can be traced in any way to the employment of the welded frame construction. In the latest B. & W. double-acting two-stroke engine with the exhaust pistons the same diameter as the cylinders, substantial use has been made of electric welding in the manufacture of the framework.

The trunk-piston engine has been found suitable for the propulsion of a considerable number of motor ships, although it is seldom adopted by British owners. On the Continent, such engines are built with outputs up to 6,000 b.h.p. in 12 cylinders, and they have been mainly employed where height is an important factor, as, for instance, in passenger or combined cargo and passenger liners. They usually run at a higher speed than the crosshead type engine, although the piston speed is about the same, namely, from 1,100–1,200 ft. per min. The main disadvantage has, hitherto, been the higher lubricating oil consumption in relation to the crosshead type, but some improvement has been effected in later designs. Both four-stroke and two-stroke trunk-piston engines are manufactured for propelling purposes, but they are mainly of the latter design.

The general features of construction of marine oil engines are similar to those of the larger stationary types. Cast iron is still largely employed for the cylinder covers and liners, although in some cases chrome steel with a tensile strength of 40–50 tons per sq. in. is adopted, and in other instances chrome molybdenum steel with a higher co-efficient of heat transmission. In order to reduce cylinder liner wear, which as an average for seagoing conditions may be taken as one-thousandth of an in. per 1,000 hrs., chrome hardening of cylinder liners has lately been adopted in a considerable number of ships. This system is costly but undoubtedly effects the required results, since operation with chrome hardened liners over periods ranging up to two or three years indicates that liner wear is reduced to an almost negligible figure. Unplated liners of normal good quality cast iron do not require grinding or polishing if turned to a good finish. A notable recent development is the employment of cast steel instead of forged steel crankshafts, although it cannot be said that complete success has yet been achieved in this direction.

It is now becoming common in marine installations to arrange for a tuned exhaust system. This is provided by designing the length of exhaust piping between the engine and silencer to give the correct periodicity to the exhaust gas waves, so that there is no interference, and, in fact, a suction effect is obtained. The requisite scavenging air pressure is thereby reduced, with a corresponding diminution in the output of the scavenging blower. In practice, with a correctly designed tuned exhaust system, the scavenging blower pressure is about 1 lb. per sq. in. less than with an untuned piston. It is usually necessary to subdivide the exhaust pipes in order to obtain the maximum effect. The principle of improved natural aspiration of the exhaust products from the cylinder through the exhaust pipe has been further developed by Kadenacy.

Auxiliary Machinery and Exhaust Boilers.

In the large majority of motor ships in which the propelling machinery installation totals over 2,500–3,000 b.h.p., the necessary auxiliaries, including lubricating oil, piston cooling and cylinder cooling pumps, are separately driven by electric motors. In single-screw vessels with engines up to the power mentioned, however, these pumps are in many cases engine driven. For instance, in the Doxford 2,500 b.h.p. three-cylinder unit, at the back of the engine is the scavenging air pump driven by levers from No. 2 cylinder crossheads. Below it, driven by links and a crosshead, are the forced lubrication pump (25 tons per hr.), the distilled jacket cooling pump (110 tons per hr.) delivering distilled cooling water to the cylinder jackets and pistons, and the sea water pump (160 tons per hr.) which circulates sea water for cooling the jacket water and lubricating oil. All of these pumps are of the piston type, but with the Werkspoor four-stroke pressure-charged engine used largely in single-screw tankers, all the auxiliary pumps are of the rotary design and are driven from two shafts in front of the engine, both of which are gear-driven from the crankshaft. When fresh water is used for cooling, it is delivered to coolers around which sea water is circulated. Where lubricating oil is employed for piston cooling, the whole of the lubricating oil, including that required for lubrication, is passed through a cooler in which the temperature is reduced by about 30–40 ° F. In order to diminish cylinder liner wear, which is considerably greater during the starting period than when running, it is usual to raise the circulating water to a temperature of about 120° F. before starting.

The employment of electrically driven auxiliaries is almost universal in ships in which the machinery is of moderate or high power. As stated, however, this does not, as a rule, apply to the blowers for the supply of scavenging air in two-stroke engines, nor to superchargers with four-stroke plant. Current is supplied from Diesel-driven generators and the tendency in recent years has been to install high-speed units, instead of the slow-running oil-engined dynamos which were provided in the earlier motor ships. Both two-stroke and four-stroke engines are utilised

for the purpose, the speeds being usually in the neighbourhood of 350–400 r.p.m., although higher revolutions are frequently adopted. For instance, in cross-Channel motor ships where reduction in weight is specially important, four-stroke Diesel engines running at 750 r.p.m. have been, on occasions, installed.

Scavenging pumps, when of the piston type, are either driven by beam lever at the back of the engine, or from the crankshaft, between two cylinders (Doxford type), or at the forward end of the engine. With Sulzer two-stroke engines it is now standard practice to have a separate side scavenge pump for each cylinder, driven by levers from the crossheads. Many four- and six-cylinder Doxford engines now have side scavenge pumps, two with a four-cylinder engine and three on a six-cylinder unit. The reduction in length is advantageous in a ship. Mechanical efficiency is perhaps slightly higher and the supply of scavenge air more regular.

Rotary scavenge blowers are usually of the positive displacement design with light cast-iron rotors bolted to the shaft. The speed of rotation is from three to four times that of the engine and the blowers are mainly chain-driven, although on small engines spur wheel drive from the crankshaft is sometimes adopted. There is a shock absorber between the blower coupling and the chain wheel, and the air is admitted from the engine room to the blower through a suction silencer. The clearance of the rotor vanes is about 24 thousandths of an inch with a tip clearance of 30 thousandths of an inch. Such blowers have an efficiency of about 85 per cent. and are usually divided into two separate units, which discharge into the scavenging air manifold through a change valve. The power absorbed in driving the blower is about 7 per cent. of the output of the propelling engine.

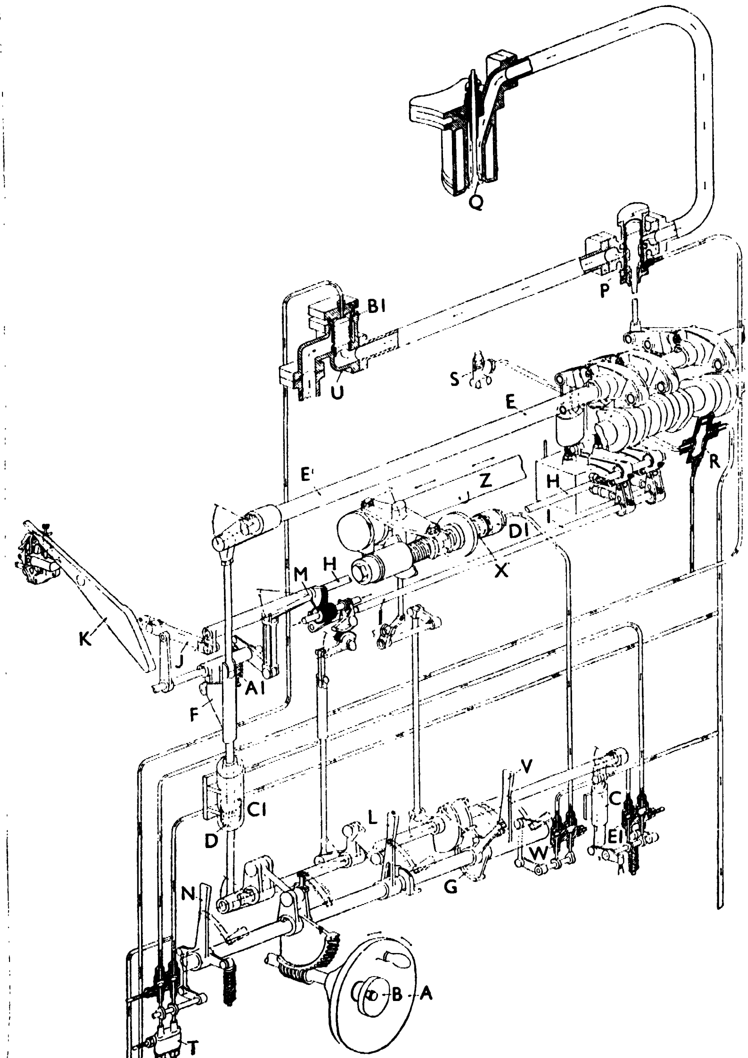
The inclusion of an exhaust gas boiler installation in a motor ship raises the overall efficiency between 5 and 8 per cent.; it is unusual for a vessel to be built in which such an installation is not made. The exhaust gas temperature from four-stroke engines is higher than that from the two-stroke type, consequently, the amount of steam generated is greater. With an average inlet temperature to the boiler of the exhaust of a four-stroke engine of about 750° F., or 400° C., approximately 1 lb. of steam is evaporated per b.h.p. at a pressure of 100 lb. per sq. in. With a two-stroke engine, the inlet temperature of the gases to the boiler being about 550° F. or 290° C., the steam production is usually under $\frac{1}{2}$ lb. per b.h.p. This figure has to be modified in the case of the Doxford engine, owing to the higher exhaust temperature, and a steam production of about $\frac{3}{4}$ lb. per b.h.p. is usually attained. Either simple or composite boilers are utilised, the latter being arranged for simultaneously firing the boiler with oil if the heat from the exhaust gases is insufficient. A further type is built in which the boiler can be fired separately by oil fuel or exhaust gases, but not by both together. If necessary, thermostatic control will enable the change-over to be made automatically. Waste heat boilers may be employed with four-stroke exhaust turbo pressure-charged engines, since the temperature of the exhaust gases falls but little in passing through the turbine before being discharged to the boilers.

In ships of moderate machinery power installations, it is usual to design the auxiliary plant so that sufficient steam is available from the exhaust boiler to drive all the steam-driven auxiliaries needed when the vessel is at sea. These include the steering gear, generator, bilge and ballast pump and compressor, the other auxiliaries being engine-driven. In this way the fuel consumption for the propelling engine represents the overall fuel consumption, including the auxiliaries, and the saving in fuel is substantial. In higher-powered ships (over about 3,000 b.h.p.) the steam is employed for heating the accommodation, in the galley, heating the fuel oil, and in some cases, for driving a stream turbo generator supplying part of the current required throughout the ship. With some installations in which the power is very large, the amount of steam generated is more than sufficient for these purposes, and it is used for distilling water. In the *Oranje*, with three 12,500 b.h.p. engines, 16,000 lb. of steam per hr. are raised from the propelling machinery at full output, and it is delivered to a distilling plant providing 300 tons of fresh water per day, which meets all the passenger requirements. In this instance, taking the exhaust heat recovery plant into consideration, the total thermal efficiency of the whole installation is raised from 31·6 to 54·2 per cent.

The Diesel engines installed in the machinery compartments of motor ships, coupled to generators for the supply of current, are of the normal stationary relatively high-speed type, the two-stroke and four-stroke designs being used to an approximately equal extent. The airless injection vertical type is invariably employed, but horizontal engines have, in some cases, been utilised in large refrigerated motor ships for driving the refrigerating compressors, the adventange being that they are of low height compared with the vertical unit.

Reversing Systems.

The basic principles underlying the systems of reversing marine Diesel engines are similar, although the manner in which they are worked out differ to a certain extent. Broadly speaking, the alteration of the timing of valves and pumps necessitated when reversing is effected either by partially rotating, or moving longitudinally, a shaft upon which the cams operating the necessary mechanism are mounted. Air or combined air and oil servo motors are utilised for providing the necessary movement, except in the case of the Doxford engine, in which, even in the largest izes, hand actuated mechanism is employed.



Reversing Mechanism of a Double-Acting Two-Stroke Marine Diesel Engine.

A typical example of starting and reversing mechanism for a large double-acting two-stroke marine Diesel engine is illustrated on page 310. The control wheel (A) with a small locking wheel (B) is rotated in a clockwise direction when it is desired to start up the engine. This moves the piston (D) in the oil cylinder of the servo motor, also the cam (F) in an upward direction. At the same time an angular movement is given to the control lever shaft (E) so that the fuel and starting levers are set in their correct position. Contact is made by the fuel pump lever with its cam (one lever for each valve) but the starting air lever remains clear of the cam. Through the upward movement of the control cam (F) an angular motion is given to the suction valve eccentric shaft (H) which alters the setting of the suction valve of the fuel pump. The starting air master valve (U) is kept shut, in the ' stop ' position.

The starting lever (N) is moved, causing air to be shut off from the master valve piston and the chamber (B1) above the valve (U) to be exhausted through the control box (T). This allows air to be delivered to the starting air operating valves (P) and (R) for the top and bottom cylinders. Meanwhile, air is supplied to the operating valve pistons through the starting air operating valve control box. In consequence, contact is made between the rollers of the operating valves and the starting air cams. Air is thus delivered to the starting air valves (Q) and (S) in the top and bottom cylinder covers respectively and the engine starts up. The handle (N) is then released and air is cut off from the operating gear. The rollers are lifted clear of the cams, since the springs come into operation. At this stage air is delivered to the chamber (B1) and the supply of compressed air to the engine is thereby shut off.

The engine is brought to the running position by a further movement of the control wheel (A) in a clockwise direction. This causes the suction shaft (H) to turn also the relief shaft (I), but this latter can be advanced or retarded independently through the lever (L) and the spiral gear (M). The engine is stopped by turning the control wheel (A) in an anti-clockwise direction. Fuel is cut off in consequence by the control cam (F) and the movement of the servo air cylinder control gear (O). The arrangement of this gear is such that air is admitted to the cylinder (D) at the position when the tappets lifting the levers are just in contact with the levers. When the ' stop ' position is reached air is cut off from the cylinder (D) and the chamber (O1) is exhausted by the control box (E1). The object of this is to avoid reverse loading whilst the engine is being started.

When reversing the control wheel (A) is brought to the ' stop ' position. There is an interlocking gear which prevents the engine being started unless this operation is carried out. The reversing lever (V) is moved, thus allowing air to enter the reversing cylinder (X) via the control box (W). The camshaft (Z) then moves in a longitudinal direction, bringing the respective cams in their correct positions for running in the astern direction. Movement of the lever (V) in the opposite direction cuts off the air supply to the reversing cylinder and the spring causes the cam shaft to move back to its original position. The Aspinall governor is provided to eliminate racing. The operating lever (K) is of cam shape, allowing contact with the roller at the end of the link (J). When the lever (K) is moved the link revolves the suction shaft (H) and the relief shaft (I) so that the fuel is shut off from the cylinders. When the engine is slowed down, the lever (K) is moved by the governor and the engine is set for normal running again by the operating spring (A1).

In the Doxford engine, with two horizontally disposed fuel valves for each cylinder, one at the back and one at the front, the former is used only when the engine is running ahead. There are two camshafts, at back and front respectively, and the reversing lever when moved causes the camshaft to slide longitudinally and brings the ahead or astern cams under the fuel and starting valve rollers as required. When the air starting lever is actuated, air is admitted to a relay cylinder and the starting air lever rollers come in contact with their respective cams, allowing starting air to enter the cylinders where the pistons are in a position to receive it. There is, finally, a manœuvring lever which controls the lift and timing of the fuel valves, and thereby the speed of the engine.

In the Burmeister & Wain two-stroke marine Diesel engine, an arrangement is employed by which, on reversing, the fuel cams remain stationary, the camshaft being turned through an angle of 120°. The cams are mounted on sleeves to allow this motion to take place, and as a result a single cam suffices both for ahead and astern rotation. The reversing lever, through the movement of which this operation takes place, also causes the cam disc of the starting air distributor to be set in the correct position for supplying air to the engine to start up in the reverse direction. In the Sulzer system for two-stroke single-acting engines, side-by-side cams are mounted on the cam shaft for ahead and astern operation of the starting air valves respectively. As a symmetrical cam is employed for actuating the fuel injection pump, no movement of this takes place during the reversing period. The actual operation of reversal is carried out by the actuation of the telegraph lever, which is a special feature of the Sulzer design. By its operation, pressure oil is delivered to a valve controlling the piston of a servo motor, according to the direction required. By the movement of this piston, the actual motion of the camshaft is effected.

In most four-stroke engines reversal is carried out on a similar principle, namely, by the longitudinal movement of the camshaft on which side-by-side ahead and astern cams are carried. In the Werkspoor four-stroke engine, however, the valve levers are pivoted on eccentrics which are fitted to the reversing shaft. This is rotated 180° and when the reversing lever is moved (through the action of a servo motor), the valve levers are raised. Due to the fact that they are mounted eccentrically at a slight angle, the levers move sideways and then, when lowered by the continued

movement of the servo motor, make contact with a second cam for operation in the astern direction. This motion involves the employment of two adjustable tappets for actuating the valves.

Air for starting and manœuvring purposes in marine Diesel engines is supplied at a pressure of 25 atmospheres, or about 355 lb. per sq. in. Lloyd's Rules call for compressed air receivers for starting air of sufficient capacity to permit 12 consecutive startings of each main engine without replenishment. The capacity of the starting air reservoirs differs substantially according to the ideas of the respective shipowners and varies so widely as from 4·5 cub. ft. per b.h.p. to 8·7 cub. ft. per b.h.p. An average figure of 6 cub. ft .per b.h.p. may be taken.

SUPERCHARGING.

It was stated previously that all marine Diesel engines of the four-stroke type are now pressure-charged. This development was rendered necessary in order to enable the engine to compete commercially with two-stroke machinery. Various supercharging systems have been utilised with marine machinery, but at present only two receive serious application. In each case the increase in continuous rated power is in the neighbourhood of 35–40 per cent., although in corresponding stationary installations with the Büchi exhaust pressure-charging system, an increase of 50 per cent. is accepted as a reasonable continuous rating. In this system, which has been adopted with engines developing up to 6,000 b.h.p., use is made of the heat energy in the exhaust gases which are supplied direct to a turbine coupled to a blower, from which combustion air for the Diesel engine cylinders is discharged at a pressure of from 5 to 7 lb. per sq. in above atmospheric. The exhaust and inlet valves are open simultaneously for short periods and the air pressure exceeds the average exhaust pressure above one-third of full load. Nearly one-third of the air supplied has a scavenging effect which cools the piston crown, cylinder cover and exhaust valve. The blower adapts itself automatically to changes of load. As the additional weight required for the turbo blower is not proportionate to the increased output, it is found that the total saving in weight of a supercharged engine compared with an unsupercharged type, is nearly 25 per cent. The inlet and exhaust valves must have a higher lift than those of a corresponding unsupercharged engine. As typical figures, assuming the opening of the inlet valve of an unsupercharged engine taking place at 21° before top dead centre and closing 21° after bottom dead centre, and having a lift of 62 mm., an exhaust turbo-charged engine would have inlet valves opening 70° before top dead centre, closing 25° after, and having a lift of 76 mm. The exhaust valves of the unsupercharged engine would open 41° before bottom dead centre and close 15° after top dead centre, and have a lift of 62 mm. The turbo-charged engine exhaust valves would open 20° before bottom dead centre, close 55° after top dead centre and have a lift of 76 mm.

In the Büchi exhaust turbo blower system, which has been very widely adopted, the exhaust pipes from the cylinders are subdivided and coupled up in such a manner as to prevent back pressure which might arise if the exhaust of all cylinders were discharged into one pipe. As an example a ten-cylinder 6,000 b.h.p. engine has four exhaust pipes, namely, from cylinders 1, 2 ; 9, 10 ; 3, 6, 7 ; 4, 5, 8 ; the firing order is 1, 4, 3, 2, 5, 10, 7, 8, 9, 6. With the engine in question, at full load (115 r.p.m.) and a mean indicated pressure of 8·25 kg. per sq. cm., or 117 lb. per sq. in., the exhaust gas quantity is 640 kg. per min. and the pressure 1·23 kg. per sq. cm. abs. The quantity of air delivered is 536 cub. metres per min. at a pressure of 1·30 kg. per sq. cm. abs. The lowest temperature at the exhaust valves is 390° C. and 450° C. at the turbine inlet. The increase in temperature after leaving the exhaust valves is to be noted. These figures relative to a Harland–B. & W. engine were given in a paper by Mr. C. C. Pounder before the Institute of Marine Engineers on 'Some Recent Diesel Installations and their Characteristics.'

An exhaust gas turbo-supercharging system has now been developed by D. Napier & Sons (English Electric Co.) and standardised units are being manufactured.

The alternative system of supercharging, under piston super-charging, has been applied to a number of different types of engines up to 5,000 b.h.p. with ten cylinders. The bottom of the main combustion cylinders is closed in, thus forming with the pistons low pressure compressors which supply supercharging air through the inlet valves. The suction and discharge manifolds are incorporated in the engine frame and there is a valve chest for each cylinder containing two suction and two discharge valves. By means of throttle valves between adjacent valve chests, the excess air on the downward stroke is pumped from one cylinder to another and low compression results. The supercharging pressure varies from 4 to 5 lb. per sq. in and there is an overlap between the opening of the exhaust and inlet valves, but to a smaller degree than is necessary with the exhaust turbo-charging system.

Supercharging has not, hitherto, been applied commercially with two-stroke engines, but experimental work has been carried out with this end in view, and it is probable that two-stroke supercharged engines will be built in future with an output 25 per cent. greater than that of unsupercharged engines of this class.

Diesel Engine Fuels.

Shipowners, oil engine manufacturers and oil suppliers are not agreed on the subject of fue oil specifications to give the most satisfactory operation. The characteristics of fuel vary in their influence according to the class of machinery, but three of the most important are: (1) viscosity, (2) ignition quality, (3) cleanliness.

The specific gravity, flash point, carbon residue, sulphur, pour point, and distillation range also have a marked influence. The difficulty of providing a standard specification that would apply to all engines may be illustrated by the fact that an oil which would comply with a manufacturer's demand for a certain carbon or sulphur content, may be far superior in its viscosity characteristic than is needed for that particular design. Most, but not all characteristics are subject to exact measurements, and among those which cannot be exactly specified are cleanliness, and in a certain degree, ignition quality, although methods have lately been developed for measuring ignition quality which may lead to standardisation. In regard to viscosity, the characteristics of two of the requirements are opposed, since the degree of viscosity should be such that the fuel will be properly atomised by the injection valves, and at the same time will penetrate thoroughly into the compressed air. Hence, minimum and maximum viscosity characteristics are sometimes specified for fuels. Ignition quality is more frequently specified in America than in the United Kingdom and may either be designated by Diesel Index or Cetane Number. There is no exact relationship between these two standards. As a rule, fuels with high specific gravities have a higher heat value than the lighter oils, to the extent of a maximum of 5 per cent., but the specific fuel consumption is somewhat higher, and the output per gallon is approximately the same. High sulphur content may, in some engines, give rise to corrosion, but it is substantially eliminated if cylinder liners are chrome hardened.

Standard specifications have been issued by the British Standards Institution as given below. The designation ' Heavy Diesel Fuel ' refers to the grade for slow-running direct-coupled Diesel engines in ocean-going ships, whilst the designation ' Marine ' applies to the engines of the faster-running type.

British Standard Specifications of Diesel Oil.

	Marine.	Heavy Diesel Fuel.
Flash point	150° F.	150° F.
Hard asphalt	--	4 per cent.
Ash	0·03 per cent.	0·1 per cent.
Viscosity Redwood No. 1	60 secs.	max. 750 secs.
Viscosity S.S.U.	67	840
Water	0·5 per cent.	1 per cent.
Pour point	30° F.	—
Conradson carbon	3 per cent.	8 per cent.
Sulphur	2 per cent.	—
Gr. Cal. value B.Th.U. per lb.	18,750	18,250

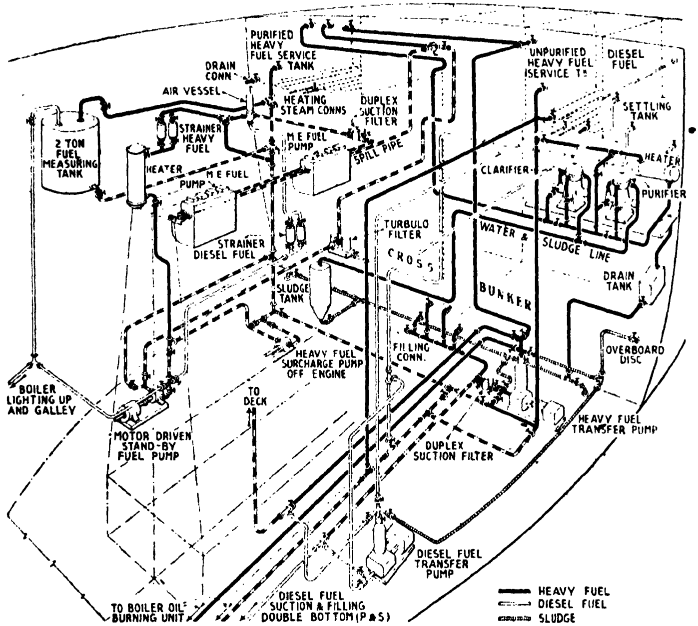
In all motor ships built in America for the United States Maritime Commission, a heavier residual oil is specified than is commonly employed by British shipowners, and the classification of the American Society for Testing Materials is adopted as a basis. The fuel is known as ASTM 4D and the following are the main characteristics specified :—

ASTM. Specification of Low-Grade Fuel for use in Diesel Engines.

Flash point ° F. minimum	140
Water and sediment per cent. by volume, maximum	0·5
Viscosity at 100 degrees F. Saybolt, seconds	140
Carbon residue per cent. by weight, maximum	3·5
Ash per cent. by weight, maximum	0·05
Pour point, ° F. maximum	35
Sulphur per cent. by weight, maximum	2
Cetane number, minimum	30

The usual practice in ships for the purification of the fuel oil is to pass it through a centrifugal separator, also as a rule through a fine filter, and in some cases, it is heated before being delivered to the fuel pumps. In modern American motor ships using the heavier grades now specified, the fuel is passed through a heater before reaching the separators, the temperature being raised to 180° F. After being discharged from the centrifuges, the oil is reheated, the temperature being thermostatically controlled, so that it is not below 180° F. when it enters the fuel valves. In the United States, a great deal of work has been carried out in connection with the use of heavier grade residual oils in Diesel engines, and the experience gained by engine manufacturers working on the problem indicates that the main factor limiting its use is its viscosity. The limit set is stated to be 1,000 seconds, Saybolt Universal, and it is claimed that fairly high percentages of carbon or residual asphaltum present no serious difficulties.

After prolonged experimental work a system has been adopted for the burning of boiler fuels in marine Diesel engines, in the machinery of the tankers of the Anglo-Saxon Co.'s fleet, based upon the investigations of Mr. John Lamb. The fuel is heated to a temperature of 180° F. and passed first through an ordinary De Laval centrifugal separator for the removal of water and suspended matter, and secondly, through a clarifier, when the last of the ash in the fuel is removed, the clarifier being a centrifugal machine with a modified form of bowl. The first ship to be so equipped, the *Auricula*, was placed in service in 1946.



Arrangement of system for using boiler oil in the machinery of the motor ship *Auricula*.

The arrangement adopted is seen in the illustration. The results after a year's working were published in a paper read before the Institute of Marine Engineers by John Lamb in December 1947,* and it was stated that the annual saving in the case of a 4,000 i.h.p. installation was £5,030 per annum. The cost of the equipment was £7,259, but this should be reduced. The mean wear of four cylinder liners was 0·648 mm. and the wear rate per 1,000 miles was 0·0093 mm. Mr. Lamb is confident that the amount of the liner wear compared with that in an engine burning Diesel oil is not excessive, and that the engine will operate over ten years before renewal of liners becomes necessary. The only stipulation made in regard to fuel is that the viscosity should not exceed 1,500 secs. Redwood 1 at 100° F.

Bearing on the question of the importance of various characteristics in Diesel engine fuel, it is to be added that detailed tests carried out at the Delft Laboratory showed that so long as the sulphur content of a fuel is not higher than about 1 per cent., the effect of this sulphur on cylinder wear is not important in comparison with that of other factors. As, however, the sulphur increases above 1 per cent., its effect on wear becomes more marked, and when it exceeds 2 per cent. the sulphur content has a strong influence on the rate of wear. On the other hand, in most cases a really high sulphur content is encountered only in heavier fuel, having also a high viscosity, also a measurable ash content. Hence, it is not always fair to attribute abnormally rapid liner wear solely to the sulphur compound present.

* 'Burning Boiler Fuels in Marine Diesel Engines.'

The Diesel fuel oil which has been mainly supplied for motor ships bunkering at British ports during the war, known as Pool Marine Diesel Oil, has the following approximate specification :—

SPECIFICATION OF POOL MARINE DIESEL OIL.

Specified gravity at 60° F.	max.	0·930
Closed flash point	min.	150 ° F.
Viscosity Redwood 1 at 100° F.	max.	85 secs.
Pour point	max.	30° F.
Sulphur	max.	1·8 per cent.
Conradson carbon	max.	2·5 per cent.
Water	max.	0·5 per cent.

It will be noticed that this specification agrees more closely with the British Standard Specification for Marine Diesel Fuel than that of Heavy Diesel Fuel. It will also be observed that there are various characteristics included in the American specifications which are omitted in the British, and vice versa. For instance, in this country no reference is made to ignition quality (Cetane Number). In America viscosity is almost invariably stated in terms of Saybolt Universal, and in Great Britain by Redwood No. 1. Roughly speaking, the following relationship holds good.

Redwood No. 1 = 1·2 Saybolt Universal at 100° F.

Weights and Sizes.

Diesel engines now built for installation in mercantile ships can be constructed with a weight 70 per cent. less than the earliest types in 1914. For instance, the four-stroke engines utilised at the latter date averaged 400 lb. per b.h.p., whereas the modern double-acting two-stroke engine weighs 115 lb. per b.h.p. Four-stroke single-acting pressure-charged machinery of modern design has a weight less than half that of corresponding engines manufactured 25 years ago.

In recent years the application of electric welding has substantially lightened Diesel machinery of various types. In the case of the Burmeister & Wain double-acting two-stroke engine, with the latest welded construction the weight has been brought down from 135 lb. per b.h.p. to 116 lb. per b.h.p., and in the welded frame Doxford opposed piston engine a reduction of 20 per cent. was effected. There will undoubtedly be new developments in the employment of welded construction in Diesel machinery, whilst it is probable that further diminution in specific weight will be effected by the increased adoption of light metals, even in slow running engines. For higher speed units this development will become even more marked.

The following table shows the average weights of various classes of marine Diesel engine as now manufactured. The weight of the engines of the pocket battleship *Deutschland* is included as a matter of interest, to indicate the extent to which reduction can be effected with special types of engine.

Trunk piston engines are considerably lighter than those of the crosshead design, the low figure of 130 lb. per b.h.p. being attained with the two-stroke single-acting class. Apart from other features, the considerably higher speed of rotation normally adopted with the trunk design leads to a marked weight reduction. It will be observed that there is little difference between the specific weights of two-stroke trunk-piston relatively high-speed engines and the modern two-stroke double-acting slow-running unit.

WEIGHTS OF MARINE DIESEL ENGINES.

Type.	Trunk or Crosshead.	R.p.m.	Weight per b.h.p.
Four-stroke single-acting	Crosshead	110	300
,, ,, (1914 type) . .	,,	115	400
,, ,, (pressure-charging) . .	,,	120	215
Four-stroke single-acting	Trunk	150	225
,, ,, (pressure-charging) .	,,	150	170
Two-stroke single-acting	Crosshead	110	150–160
,, ,, ...	Trunk	160	130
,, ,, ...	Opposed piston	100	170
Two-stroke double-acting	—	115	115–135
,, ,, (high-speed) . .	—	450	17·6*

* Machinery of *Deutschland* class of pocket battleship, 48·5 lb., including propellers, shafting, etc.

It has already been recorded that during a comparatively short number of years the size of marine Diesel machinery for given power has diminished substantially, and roughly it may be stated that engines may be installed in the same machinery space to-day as was required 30 years ago for machinery of half the power. The two-stroke double-acting engine is naturally the shortest of all types, and the following table gives particulars of some typical motor ships of comparatively recent construction, indicating the engine-room lengths, the power and the propeller speed. The question of height has also to come into consideration and the two-stroke double-acting design is at a disadvantage in comparison with the single-acting type. This may be of importance in passenger liners, and one of the reasons why single-acting engines were installed in the highest-powered motor ship yet built—the *Oranje*—was that the lower height of this type in comparison with the double-acting unit enabled an extra deck to be built above the engine room.

ENGINE-ROOM LENGTHS IN TYPICAL MOTOR CARGO SHIPS.

Ship.	Gross Reg. Tonnage.	Length b.p. ft.	ins.	Beam ft.	ins.	Speed Knots.	B.h.p.	E.R. Length ft.	ins.	Engine Type.	Propeller speed	Twin or single-Screw.
Trafalgar .	5,500	430	0	57	0	16	5,600	57	6	Stork double-acting	118	single
Delius .	6,065	430	0	62	0	14	4,000	54	0	Harland-B. & W. double-acting	110	single
Alex. Van Opstal	5,965	420	0	57	0	11½	4,800	53	0	B. & W. double-acting	100	single
Port Montreal	5,880	432	0	59	0	13½	4,250	60	0	Doxford opposed piston	108	single
Arendskerk .	7,890	480	0	63	0	17	10,000	72	0	Sulzer single-acting	120	twin
Selandia .	8,482	425	0	63	0	16	6,000	51	6	B. & W. double-acting	120	single
Waimarama	11,091	515	6	70	0	18	12,000	72	0	Harland-B. & W. double-acting	109	twin

GEARED DIESEL AND DIESEL-ELECTRIC DRIVE.

British shipowners have hitherto adhered exclusively to the direct drive in motor ships, the Diesel engines being coupled to the propellers without intermediary gear. In America, however, the large majority of ocean-going oil-engined tonnage is being equipped with geared machinery. On the Continent the application of the geared drive is steadily increasing and Diesel-electrically propelled ships are being used to a substantial extent. It should be added, however, that the direct drive is still most commonly employed.

The advantages claimed for the indirect drive are (1) reduction in weight, (2) lower height, (3) less space occupied, (4) parts, being lighter, are more easily handled, (5) increased standardisation is possible, and (6) one engine can be stopped for overhaul or repair without seriously reducing the ship's speed. Against these advantages is to be set the higher overall fuel consumption due to losses in the gearing and the slightly lower efficiency of the high-speed engine.

The usual arrangement with geared Diesel installations comprises two or four units running at about 250 r.p.m. coupled to a single or to two propellers through reduction gear with a ratio of about 2½–1, to give a propeller speed in the neighbourhood of 100 r.p.m. Either hydraulic or electro-magnetic slip couplings are fitted between the engines and the gearing, so that the latter is not subjected to shocks due to cyclic irregularity. In the hydraulic couplings, lubricating oil is used as the medium for transmitting the power from the engine shaft to the propeller. In the electric slip coupling, which is a comparatively new development, there is an air gap between the two rotary members, one of which is mounted rigidly on the engine shaft and the other connected to the gear. The couplings act as disconnecting clutches by means of which the engines can be connected to or disconnected from the propeller instantly. The external member coupled to the engine shaft is connected to the ship's direct current auxiliary power supply for excitation. The inner member, which has a squirrel cage winding, rotates inside this field. The mechanical rotation of the field or outer member creates a rotating magnetic field which induces current in the squirrel cage winding. The interaction of the resulting magnetic field creates forces which cause the squirrel cage to follow the field except for a small slip. The efficiency of the coupling is in the neighbourhood of 97½ per cent. During manœuvring period, the engines need not be stopped and this is advantageous from the point of view of economy in starting air and reduced cylinder liner wear.

The Diesel-electric drive has been employed in this country to a small extent for tugs and in America also, for fairly high powered vessels of this class, and special ships such as dredges. In these vessels the direct current system of transmission is adopted, but in the ocean-going cargo ships which have been built in Europe with electric propulsion, alternating current is utilised. The usual arrangement is to install two, three or four three-phase 2,000 volt alternators driven by single-acting or double-acting two-stroke engines of about 3,000 b.h.p. each, running at a speed of 250 r.p.m. The propeller is driven by a synchronous motor at about 125 r.p.m. for full ship's speed. The current for operating the auxiliaries is applied from the main generators, hence, in such ships, A.C. motors are utilised for driving pumps, compressors, etc. The system described has been applied to a number of fast cargo ships and passenger liners.

In America, in some Diesel-electrically propelled ships of fairly high power, the current is supplied to high-speed electric motors which drive the propeller or propellers through reduction gearing, the object being to utilise light-weight motors occupying comparatively small space.

It is the opinion of many engineers and shipowners that in view of the increasing experience which has been gained with high-speed Diesel machinery, the adoption of this class of plant in conjunction with gearing or electric transmission will be developed to a considerable extent in the future, and that substantially higher speeds than 250 r.p.m. will be employed. The 1,000 r.p.m. relatively high-powered engine is visualised, whilst Mr. H. R. Ricardo has proposed the installation of a large number of small oil engines running at 1,500 to 2,000 r.p.m. coupled to the dynamos supplying current to a single propelling motor.

The comparative advantages of different classes of machinery for ships have been the subject of two important symposiums, namely, 'The Engining of Post-war Cargo Vessels of Low Power' (*Transactions of the Institute of Marine Engineers*, July 1944), and 'The Engining of Cargo Vessels of High Power' (*Transactions of the Institute of Marine Engineers*, January 1948). In both of these, the question of the employment of Diesel-electric machinery is examined at some length for relatively low- and high-powered plant respectively.

A comparatively large number of high-speed engines both for naval and commercial service have been developed in recent years, ranging from about 400 b.h.p. to 1,000 b.h.p. The large majority are of the four-stroke type and the following are the average characteristics, subject to variations, according to the individual design.

MODERATE POWERED HIGH-SPEED MARINE ENGINES.

Engine speed . .	. **1400–1500 r.p.m.**
Piston speed . .	. **1700–1800 ft. per minute**
Mean effective pressure .	**80 lb. per sq. in. unsupercharged** **110 lb. per sq. in. supercharged**
Specific volume .	. **5 cu. ins. per b.h.p.**
Weight . .	. **12–18 lb. per b.h.p.**

By the adoption of a high degree of supercharge, giving 50 per cent. more power than when unsupercharged, and the employment of specially high speeds the weight can be brought down to 10 lb. per b.h.p. The maximum piston speed employed with fast running engines is 2,000 ft. per minute.

American high-speed engines utilised in Naval craft, also in some mercantile Diesel-electrically propelled ships, are mainly of the two-stroke single-acting design, but the speeds for units up to 1,200 b.h.p. do not as a rule exceed 750 r.p.m. A double-acting two-stroke high-speed engine was used in Germany for motor torpedo boats. This is a development of the M.A.N. slow running engine and has a maximum output of 1,200 b.h.p. at 1,200 r.p.m., with a normal rating of 900 b.h.p. The mean effective pressure referred to brake power at maximum output is 51 lb. per sq. in. and it is recorded that the weight is only 4 lb. per b.h.p.

SECTION XXX

AUTOMOBILES AND ELECTRIC ROAD TRACTION

(pp. 321–385.)

(Revised by F. I. C. Gillibrand.)

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AUTOMOBILES.

Trend of Design.

General.—The general trend of design both for motor cars and commercial vehicles is, of course, towards greater efficiency and the outstanding tendencies of present-day design may be exemplified by improved performance, *i.e.* acceleration and braking in respect of motor cars, and lightness of weight and economy of operation in respect of commercial vehicles. The British system of taxation has in the past encouraged the development of small, long stroke engines for motor cars, and large carrying capacity for a minimum weight with the maximum fuel economy in all commercial vehicles, whether used for the carriage of goods or passengers. Since the revision of the motor-car tax, however, engines in this category show a tendency to become more nearly 'square,' *i.e.* with a stroke/bore ratio approaching unity, and engine size is on the upward grade. Petrol engines are universal in motor cars, and although extensively used in the lighter commercial vehicles, compression ignition engines consuming fuel oil are virtually universal in the heavier commercial vehicles.

The number of models now produced under 1·5 litres capacity (approximately 12 h.p. rating) has decreased, although it is safe to say that the majority of cars produced fall within this category. The great majority of cars under 2 litres capacity have four cylinders and those above have six or eight cylinders, the latter both in line and V-formation. In the case of commercial vehicles the much larger horse-power rating encourages the use of more cylinders, and engines of 4, 5, 6 and 8 cylinders are common. In the interests of standardisation of parts and the lowering of production costs it is quite usual to employ cylinders of the same size throughout a range of engines having various numbers of cylinders.

Recent research has proceeded along the lines of investigation into the causes of, and providing remedies for, mechanical wear, particularly with regard to bearings and cylinder bores, together with the development of light alloys and steels having high strength/weight ratios.

With constantly increasing road speeds, the question of safety, particularly in regard to stability, steering and braking, receives more and more attention.

Tables or charts purporting to exhibit trend of design are generally highly misleading, because they fail to take into consideration the number of cars of each model produced; no estimate of the popularity of any particular feature can be formed unless the sales of the model in which it is embodied are known.

Considering private cars and commercial vehicles as a whole, the following tendencies will be observed:—

Cylinders.—These are generally cast in one block with the crankcase, even in the larger engine sizes, thereby providing greater rigidity. Generally speaking, side valves are used in the lower-priced vehicles, but overhead valves are virtually universal in engines of high performance, however small, and are, of course, universal in compression ignition engines. Amongst the higher-priced cars there is a move towards the employment of overhead inlet and side exhaust valves. Close-grained low nickel iron is still the most popular material for cylinder blocks, straight cast iron having been superseded for this purpose. Cylinder liners are used extensively in commercial vehicle engines, the latest practice being to employ a liner having a slip fit in the cylinder block with dry liners, thus simplifying maintenance problems. Inserted valve seats in heat-resisting alloy are also common in commercial vehicles, often being faced with stellite. Considerable advance has been made in the development of heat-resisting valve steels and both exhaust and inlet valves are often stellite faced and tipped.

Pistons.—These are universally of light aluminium alloy generally of proprietary make, and developments in manufacturing technique have tended towards the use of pressure casting in preference to gravity castings. Improvement has occurred in the design and treatment of piston rings, reducing piston and cylinder wear and consequently decreasing oil consumption.

Crankshafts.—Following American practice, the employment of cast semi-steel or innoculated cast-iron (somewhat misleadingly called cast-iron) crankshafts is increasing, and much research

in this material has been carried out by the National Physical Laboratory. It has long been recognised that a stiff and heavy crankshaft is essential to eliminate torsional vibration, hence a high tensile alloy steel has no advantage, and the newer semi-steel alloys show satisfactory resistance to wear with relatively low production cost. Forged steel crankshafts are almost universally nitride hardened in commercial vehicle applications and recent research has indicated the necessity for attention to the finish of pin and journal fillet radii.

Bearings.—Strip bearings, comprising a thin steel strip to which is bonded an exceedingly thin lining of bearing material, are almost universal both for car and commercial vehicle engines. The bearing material is invariably white metal or copper-lead, indium flashed on the bearing surface, but the discovery of the ideal alloy is still the subject of research. Bearing pressures have been much increased, particularly in compression ignition engines, as also have rubbing speeds, and the governing factor for bearing life appears to be adequate high pressure lubrication.

Clutches.—There has been little change in the design of clutches. In the majority of small and medium sized friction clutches, the single-plate type is preferred for its simplicity. These embody a very light free member having a bonded asbestos friction surface and, of course, are used dry; refinements include sprung centres and laminated steel plates to which the friction material is attached. Multi-plate clutches are only employed on engines of the highest horse-powers and are occasionally centrifugally assisted.

Transmissions.—Although a large number of epicyclic gear boxes are in use and are giving satisfaction, both in private cars and commercial vehicles, there is little doubt that the general tendency is towards a good type of syncromesh arrangement. This affords perfect ease in changing up or down, is not unduly costly, and has the great advantage of being easy to adjust and repair, besides being generally trouble free.

A fluid flywheel in conjunction with a so-called 'self-changing' gear box is employed by a number of manufacturers both on private cars and commercial vehicles of the passenger-carrying type.

Rear Axles.—The spiral bevel continues to be the most popular rear axle arrangement for private cars, and with suitable precautions in manufacture is satisfactory. Its main objection is the necessity for a tunnel in the body to accommodate the propeller shaft. The hypoid bevel gear, which permits the centre-line of the bevel pinion to be situated below the axle centre, reduces the tunnel height, but to make possible a flat floor in the rear of the car the only solution appears to be a worm drive, where the propeller shaft is at its lowest. There has, therefore, been some increase in the use of worm gears. With regard to commercial vehicles the worm gear is virtually universal throughout the heavier ranges being underslung for passenger vehicles and overslung for goods where a high ground clearance is necessary. Amongst the lighter and cheaper commercial vehicles the spiral bevel still retains its popularity although the hypoid bevel gear is gaining ground. The use of two-speed axles is on the increase in commercial vehicle applications.

Front Axles.—Independent suspension of the front wheels is now practically universal on motor cars. The various designs, described later, being suitable for laminated, helical and torsion bar springing. The conventional I-beam front axle of the reversed Elliot type is still universal on commercial vehicles generally embodying laminated springs of the semi-elliptic type.

Frames.—Although the use of rigid, relatively heavy, and often cruciform-braced frames continues, there is a strong tendency towards what is commonly known as 'chassisless' construction in which the body and its framework, or at least the lower portion of it, forms the main structure to which the various units are attached. The practice persists, however, of securing the utmost degree of stiffness in the frame itself and attaching all the units, engine and gear box, radiator, etc., by means of flexible connections in which rubber is incorporated, thus avoiding direct metallic contact between the components. This method of construction contributes largely towards the successful silencing of modern vehicles, by eliminating resonance in one member due to vibration in another.

Brakes.—These are mostly of proprietary make, viz. Lockheed, Girling and Bendix, except on commercial vehicles where often the manufacturers' own brakes are operated by a proprietary make of vacuum servo or compressed air system. In every case brakes are fitted to all wheels. The three systems named are illustrated later. Front brakes are usually of the two leading shoe type.

Suspensions.—Since the advent of independent wheel suspension, this important subject has received greater attention with consequent improvement in suspensions generally. Independent front suspension is now virtually universal on motor cars, and both helical and torsion bar springs are used successfully in the various designs. Laminated springs are also used but to a far less extent; they are usually confined to conventional rear suspension systems. It is now the practice to incorporate a torsion bar stabilising device in conventional rear suspension systems, both for motor cars and double-deck bus chassis.

There has been considerable controversy regarding the general chassis layout and the disposition of the various units; cars have been produced with the whole of the driving mechanism, engine, gear box, transmission and driving wheels, at the front of the vehicle and also in the reverse position, *i.e.* at the rear of the vehicle. The results, however, appear to be negative and it is unlikely that the present practice will change for some time. Much more ingenuity has been shown by Continental designers in this respect who are masters of the unconventional.

With regard to commercial vehicles the present tendency in this country is towards placing the power unit below floor level, with the cylinders inclined or horizontal, either at the front of the vehicle or amidships, and applies to both passenger and goods vehicles. This practice of mounting the power unit horizontally below floor level in the amidships position is common among American heavy passenger vehicle manufacturers, virtually the only varient being to mount the power unit transversely at the rear of the vehicle with the cylinders vertical; in both cases the drive being transmitted to the rear axle.

Notes on Thermodynamics.

Boyles Law. PV = a constant.

Relation between Pressure, Volume, and Temperature of a perfect gas. PV = CT

where T = absolute temperature deg. F.
P = absolute pressure in lbs. per sq. ft.
C = a constant for the gas.
C = 53·29 for air.

so for air PV = 53·29T.

Isothermal equation for a perfect gas. $PV = P_0V_0$

where P_0 and V_0 are initial pressure and volume.

Adiabatic equation for a perfect gas. $PV^{\gamma} = P_0V_0^{\gamma}$

$$\text{where } \gamma = \frac{\text{Specific heat of gas at constant pressure}}{\text{Specific heat of gas at constant volume}}$$

for air $\gamma = 1{\cdot}4$.

Adiabatic Compression.—In calculating curves of pressure and temperature the changes are adiabatic and PV^{γ} = a constant, and if T be absolute temperature $TV^{\gamma - 1}$ = a constant.

When air in a cylinder is compressed to one-fifth of its original volume (a compression ratio of 5/1)

$$P_1 = P \times \left(\frac{V}{V_1}\right)^{\gamma}; \text{ so } P_1 = 14{\cdot}7 \times 5^{1{\cdot}4}.$$

If the initial air temperature were 519° F. abs. then after rapid compression to one-fifth of its volume its final temperature would be :—

$$T_1 = T \times (5^{\gamma - 1}) = 519 \times 5^{0{\cdot}4}$$

and the converse of this is true.

In a *petrol engine* the heat is added and rejected at the ends of the stroke, while the volume does not vary, hence its operation is on a *constant volume cycle*.

The thermal efficiency of this cycle depends upon the compression ratio only. If E = thermal efficiency, $E = 1 - \left(\frac{1}{r}\right)^{\gamma - 1}$ where r = compression ratio.

TABLE I.

ENGINE EFFICIENCES FOR VARIOUS COMPRESSION RATIOS.

r = compression ratio. E = Thermal efficiency, Constant volume cycle.

r.	E.	*r*.	E.	*r*.	E.
3·8	0·4138	5·1	0·4788	6·4	0·5238
3·9	0·4201	5·2	0·4829	6·5	0·5270
4·0	0·4256	5·3	0·4868	6·6	0·5300
4·1	0·4313	5·4	0·4906	6·7	0·5328
4·2	0·4367	5·5	0·4944	6·8	0·5355
4·3	0·4420	5·6	0·4980	6·9	0·5382
4·4	0·4471	5·7	0·4984	7·0	0·5398
4·5	0·4521	5·8	0·5050	7·5	0·5534
4·6	0·4569	5·9	0·5084	8·0	0·5647
4·7	0·4615	6·0	0·5116	9·0	0·5740
4·8	0·4660	6·1	0·5149	10·0	0·6020
4·9	0·4704	6·2	0·5180	11·0	0·6170
5·0	0·4747	6·3	0·5211	12·0	0·6290

It will be seen from the above table that the efficiency increases rapidly with the increase in compression ratio until a ratio of about 5 to 1 is reached; thereafter the rate of increase falls off

until at a ratio of about 12 to 1 the advantage of increasing compression ratio becomes relatively slight.

In the case of compression ignition (diesel) or constant pressure type engines air alone is come pressed, usually to a ratio of 15 or 16 to 1. The fuel is subsequently injected at a pressure of several thousand pounds per sq. in. The duration of the injection over part of the working stroke is controlled, and the temperature of the air due to compression ignites the fuel, the oil consuming the oxygen from the air.

The heat is discharged at constant volume at the end of the stroke, so the working cycle is partly at constant pressure, and partly at constant volume.

$$\text{The heat added } H = K_p(T_2 - T_1)$$

$$\text{and the heat discharged } H_1 = K_v(T_3 - T)$$

$$\text{The efficiency } E = 1 - \frac{K_v(T_3 - T)}{K_p(T_2 - T_1)}$$

$$\frac{K_v}{K_p} = \frac{1}{\gamma}; \text{ therefore } E = 1 - \frac{(T_3 - T)}{\gamma(T_2 - T_1)}$$

where

T = initial air temperature (abs.).
T^1 = temperature after compression (abs.).
T_2 = temperature after combustion (abs.).
T_3 = temperature at exhaust (abs.).

And it follows from this that in this cycle the thermal efficiency will depend on the maximum temperatures as well as the compression ratio. With increase of temperature the efficiency decreases.

Note.—The compression ratio in any internal combustion engine, viz. r, is

$$= \frac{\text{volume swept by piston + combustion space}}{\text{combustion space}}.$$

Power Calculations.

The piston indicated horse-power developed in the engine is expended as in other prime movers by friction in the engine itself and by the resistance of the vehicle, the latter being composed of: (1) friction in the mechanical parts of the chassis; (2) the rolling resistance to the wheels on the road; (3) the surmounting of gradients; (4) the resistance of the air, and, when all these are accounted for, the remaining power is available for (5) acceleration. Item (1) consists largely of heating losses due to churning of the lubricant in gear box and rear axle. According to circumstances from 10 to 15 per cent. of the brake effective horse-power is thus accounted for, hence from 85 to 90 per cent. of total brake horse-power reaches the driving wheels.

Tractive Factor.—This is the ratio of the torque in pounds at the driving wheels/the gross weight of the vehicle in lbs., viz. $\frac{Tw}{W}$ and this ratio determines the whole performance of the vehicle.

Rolling Resistance.—For pneumatic tyres the resistances given in Table II are typical, but approximate only.

TABLE II.

ROLLING RESISTANCES, PNEUMATIC TYRES.

Concrete	9·5 lbs. per 2240
Asphalt	10·25 " " "
Stone sets	16·70 " " "
Good macadam	22·00 " " "
Poor "	46·00 " " "
Clay	66·00 " " "
Sand	200·00 " " "

Gradient Resistance.—A gradient of '1 in 7' means that in passing over 7 ft. a height of 1 ft is attained, hence G the gradient $= \frac{h}{s} = \sin \alpha$, where α = angle of slope and if P = force required to propel vehicle up the slope and W is the gross weight of vehicle in pounds $P = \frac{W}{G}$

Air Resistance.—This varies as the speed[2] thus

$$\text{Air resistance} = kV^2A$$

Where k is a constant varying from 0·0024 to 0·0006, and A is the frontal area in sq. ft.

The total tractive resistance thus becomes

$$T_R = W\left(\frac{R + 2240}{G}\right) + kV^2A$$

Here, W = gross weight in *tons*, R = rolling resistance lbs. per ton, G = gradient, V = velocity in ft per sec., and A = the projected area of vehicle (with k = 0·0024 to 0·0006).

Acceleration.—The force required $f = ma$, where m = gross weight in lb./grav. and a = acceleration required. Whence, $a = f/m$. f in this case being the tractive effort available at the road surface.

Tractive Effort.—The tractive effort at the road is equal to the torque of the engine in lb. ft. × total gear ratio between engine and back axle/radius of tyre in ft. × mechanical efficiency of chassis, *i.e.* 0·85.

Horse-power.—For a 4-cycle engine,

$$\text{H.P.} = \frac{2\pi RnW}{33{,}000}$$

where

H.P. = brake horse power.
R = radius of dynamometer arm in ft.
W = weight lifted on dynamometer arm in lbs.
n = revolutions per minute.

Horse-power may also be expressed as :—

$$\text{H.P.} = \frac{T_f \times 2\pi n}{33{,}000}$$

where

T_f = torque in lb. ft.

$$\text{or H.P.} = \frac{p \times V \times n}{33{,}000 \times 24}$$

where

p = brake mean effective pressure (b.m.e.p.) in lbs. per sq. in.
V = total swept volume of the engine in cu. ins.

Torque.—$T_f = \dfrac{5250 \times \text{H.P.}}{n}$ lb. ft.

$$T_i = \frac{63{,}000 \times \text{H.P.}}{n} \text{ lb. in.}$$

where

T_i = torque in lb. in.

Brake mean effective pressure.

$$p = \frac{262\cdot1 \times T_i}{d^2sN} \text{ lbs./sq. in. } (d \text{ and } s \text{ in cms.})$$

$$p = \frac{192 \times T_f}{d^2sN} \text{ lbs./sq. in. } (d \text{ and } s \text{ in ins.})$$

$$p = \frac{16\cdot53 \times 10^6 \times \text{H.P.}}{d^2sNn} \text{ lbs./sq. in. } (d \text{ and } s \text{ in cms.})$$

$$p = \frac{1\cdot009 \times 10^6 \times \text{H.P.}}{d^2sNn} \text{ lbs./sq. in. } (d \text{ and } s \text{ in ins.})$$

where

N = number of cylinders.
d = bore of cylinder.
s = stroke of piston.

In an internal combustion engine, the maximum torque is always delivered at considerably below the maximum speed, a graph of the torque showing a curve which first rises and then falls as the speed increases. As the sum of the vehicle resistances is a rising curve, it follows that the two curves must cross at some point beyond which speed cannot be increased.

Wheel Adhesion.—Apart from any question of engine power the performance of a vehicle is limited by the amount of adhesion between the driving wheels and the road surface. The coefficient of friction at this point is comparatively high, varying from about 0·7 on normal roads to over 1·0 in certain circumstances. The latter figure occurring with certain forms of tyre surface

and *slightly* yielding macadam roads. Except in special vehicles (four-wheel drive) only a portion of the gross vehicle weight is available for promoting wheel adhesion, hence the maximum acceleration is

$$a = \frac{w\mu_r \times 32\cdot 2}{W}$$

where w is the weight in lbs. on the driving wheels, μ_r is adhesion factor of wheels, and W is the gross vehicle weight in lbs.

The same remarks apply to the availability of the tractive effort generally which cannot exceed $w\mu_r$ in any circumstances.

Horse-Power Rating Formulæ of Various Countries.

(S.M.M. & T. and R.A.C.)

Notation :—

H = horse-power rating. d = bore of cylinder.
N = number of cylinders. S = stroke of piston.
V = swept volume of one cylinder. r = engine revolutions.

Australia (Queensland, South Victoria)
British Guiana
Canada
Columbia
Dominican Republic
Federated Malay States
Great Britain
Gibraltar
Hayti
Lebanese Republic
Northern Ireland
Porto Rico
United States of America

$$H = \frac{Nd^2}{2\cdot 5}$$
where d is in in.

$$H = \frac{Nd^2}{1613}$$
where d is in mm.

NOTE.—In Great Britain taxation is now £10 per car, irrespective of horse-power, plus the usual petrol tax.

Australia (Western).
$$H = \frac{Nd^2S}{12}$$
where d and S are in in.

Austria, Luxembourg.
$$H = 0\cdot 3Nd^2S$$
where d is in cm. and S is in meters.

Belgium.
$$H = Kd^2SNc$$
where d is in decimetres.
S is in decimetres.

Category.	Revs. per Min.	K.	C.
(1) Light vehicles . .	Under 1,500	4·25	1
(2) ,, ,, . .	Over 1,500 and under 2,400	4	1·5
(3) ,, ,, . .	2,400 and over	3	2·4
(4) Heavy vehicles including tractors light and heavy . .	—	4·25	1

NOTE.—When vehicles of (2) and (3) categories are over 6 years old (2) becomes (1) and (3) becomes (2).

Bulgaria.
$H = 0\cdot 5Nd^2S$ (for private cars)
$H = 0\cdot 3Nd^2S$ (for trucks)
where d is in cm. and S is in meters.

Cochin China, Tangier. $H = KNd^2Sr$

where
K = 0·00020 for 1 cylinder.
K = 0·00017 for 2 cylinders.
K = 0·00015 for 4 cylinders.
K = 0·00013 for more than 4 cylinders.

d and S are in cm.
r = maximum engine revs. per second.

Czecho-Slovakia.

NOTE.—Taxation is based on the capacity of the engine.

$$\text{Capacity in litres} = \frac{0 \cdot 785 N d^2 S}{1{,}000{,}000}$$

where d and S are in mm.

Denmark, Finland.

As Austria for four-stroke engines. Special factory ratings are adopted for two-stroke engines.

Eire. $H = \frac{Nd^2}{2 \cdot 5}$ (for engines having 1 piston per cylinder)

$H = \frac{Nd^2}{1 \cdot 6}$ (for engines having 2 pistons per cylinder)

where d is in in.

France. $H = KNd^2Sr.$

where $K = 0 \cdot 00015$, d and S are in cm.

r = engine revs. per second, which are fixed at :—

30 for cars and light vans up to 1,250 kgs.

25 for public service or goods chassis which, including tyres, weigh from 1,250 kgs. to 2,250 kgs. unladen.

20 for public service or goods chassis which, including tyres, weigh over 2,250 kgs. unladen.

NOTE.—All fractions of horse-power are taken to the nearest whole figure.

Germany.

NOTE.—Taxation is based on the capacity of the engine.

$$\text{Capacity in c.c.} = 0 \cdot 00078 N d^2 S$$

where d and S are in mm.

Hungary. $H = \frac{Nd^2}{20}$, where d is in cm.

Italy, Rhodes (Aegean Sea). $H = 0 \cdot 08782 N V^{0 \cdot 6541}$

where V is in c.c.

and $V^{0 \cdot 6541}$ is an exponential function of V.

Japan (Tokio). $H = \frac{Nd^2}{3}$, where d is in in.

Spain. $H = 0 \cdot 08(0 \cdot 785 d^2 S)^{0 \cdot 6} N$ (for four-stroke engines)

$H = 0 \cdot 11(0 \cdot 785 d^2 S)^{0 \cdot 6} N$ (for two-stroke engines)

where d and S are in cm.

NOTE.—$0 \cdot 6$ is an exponential function of the product in the brackets.

Sweden. $H = \frac{Nd^2Sr}{K}$

where

$K = 180{,}000$ (for Diesel engines).

$K = 200{,}000$ (for four-stroke engines using petrol or oil).

$K = 210{,}000$ (for two-stroke engines ignited by compression).

d and S are in cm.

r = max. engine revs. per minute.

South Africa. $H = \frac{Nd^2S}{200{,}000}$

where d and S are in mm.

Switzerland. $H = 0 \cdot 4 N d^2 S.$

where d is in cm. and S is in metres.

Syria (Aleppo). $H = 0 \cdot 06 N d^2$

where d is in cm.

Fuel.

Petroleum hydrocarbons are divided into four classes in the gasoline range, paraffins, naphthenes, aromatics, and olefins. They can be present in different proportions, straight run gasoline having mainly paraffins and naphthenes, when obtained from one source and a high proportion of aromatics when obtained from another.

Olefins are unsaturated hydrocarbons and are found in cracked gasolines.

Pool Motor Gasoline Specification.

Colour	orange.
Dye content	1·2 gm. per 100 gals.
Odour	merchantable.
Copper strip test	must pass.
Reid vapour pressure	10 p.s.i. max.
Sulphur	0·25 per cent. max.
Distillation :—	
10 per cent.	70° C. max.
50 ,,	125° C. ,,
90 ,,	180° C. ,,
Final boiling point (F.B.P.)	205° C. ,,
Residue	2 per cent. max.
Gum	10 mgm./100 mls. max.
Octane number	70 min.
T.E.L. content	1·2 c.c./imperial gal. max.

Typical Pool Motor Gasoline Analysis.

Specific Gravity	60° F/60° F.	0·746
Distillation :—		
(Initial B.P.) I.B.P.		33° C.
Temp. at 2 per cent.		49 ,,
,, 5 ,,		58 ,,
,, 10 ,,		68 ,,
,, 20 ,,		87 ,,
,, 30 ,,		100 ,,
,, 40 ,,		113 ,,
,, 50 ,,		125 ,,
,, 60 ,,		136 ,,
,, 70 ,,		149 ,,
,, 80 ,,		163 ,,
,, 90 ,,		178 ,,
F.B.P.		197 ,,
Total distillate		98 per cent.
Residue		1 ,,
Loss		1 ,,
Vol. to 70° C.		12 ,,
,, 100° C.		30 ,,
,, 110° C.		62 ,,
Colour		Orange-yellow.
T.E.L.		1·24 c.c./imperial gal.
Gum		1 ml./100 ml.
Octane number		71
Bromine number		2

Specific Gravity.—The specific gravity of a gasoline is expressed as the ratio of the weight of unit volume of the gasoline at 60° F. to the weight of unit volume of water at 60° F.

A difference in specific gravity of two gasoline samples does not necessarily mean that the lower value gasoline is of better quality except when it is known that both samples have been distilled from the same crude oil.

Specific gravities usually fall between 0·720 and 0·780.

Distillation.—The volatility of a gasoline can be determined by a distillation test.

Cold starting can be assessed from the '20 per cent. point,' whilst intercylinder distribution and crankcase dilution can be assessed from the '70 per cent point,' these being the temperatures at which the quoted percentages are recovered.

The remaining points of the distillation curve are not so important. I.B.P. and F.B.P. are determined and give an indication of the 'cut' of the gasoline. The I.B.P. cannot be determined with great accuracy. The F.B.P. shows if any heavy ends are present. 'Loss' is that proportion of light or volatile components which are not recorded, whilst 'residue' is that proportion remaining when the distillation is completed.

Octane Number.—The octane number gives the knock rating or detonation limited performance of gasoline and is determined by a standardised procedure and equipment. It is determined by referencing the test gasoline against a blend of iso-octane and heptane or known secondary reference fuels. The percentage of iso-octane, by volume, in the blend is given the term 'octane number.'

Tetra Ethyl Lead.—Tetra Ethyl lead ($Pb(C_2H_5)_4$) is used on a very large scale as an additive in gasoline to suppress detonation, thereby giving an appreciation in octane number.

The appreciation of octane number by the addition of T.E.L. varies with different gasolines dependant on their composition..

This appreciation diminishes with successive additions of T.E.L. until a point is reached where further additions are no longer justified on an economic basis.

Present-day motor gasoline has a T.E.L. content slightly in excess of 1 c.c. per imperial gallon.

Tetra Ethyl lead is combined with ethylene dibromide and dichloride in such proportions that when burnt in the cylinders the T.E.L. will be evacuated as lead bromide and chloride. It is poisonous and dangerous to handle.

Sulphur.—Gasoline is refined to remove both free sulphur and sulphur compounds. However, traces remain and if in a free state or as mercaptan or sulphuric acid derivatives, they can be corrosive. They also give an objectionable odour to the gasoline.

The corrosive properties of a gasoline are usually determined by the copper strip test. There must be no change in the surface of the copper strip after immersion at stated conditions in the fuel.

Gum.—Gum is undesirable in gasoline due to its effect on the functioning of the engine, such as valve sticking and deposits in the carburetter and induction system, causing incorrect metering of the gasoline. It is non-volatile and therefore remains when gasoline is evaporated.

The gum content of cracked gasoline may increase in storage due to the presence of unsaturated hydrocarbons in the gasoline.

Colour.—Unleaded gasoline is usually water white in appearance and will remain so unless deterioration, due to the presence of unsaturated hydrocarbons, takes place.

Leaded gasolines are usually coloured by the addition of a dye to distinguish the different grades. Present motor gasoline is coloured orange.

Bromine Number.—The bromine number is obtained as a laboratory test to determine the presence of cracked gasoline in any given sample.

It is necessary to know the value for the cracked gasoline itself before this assessment may be made, owing to the fact that this value varies considerably.

Vapour Pressure.—The vapour pressure of a gasoline is controlled to comply with considered safe limits. Too high a value may give rise to vapour locks in the fuel system. The maximum safe value is determined relative to the existing climatic conditions.

Fuel Oil.—Many attempts have been made in the past to employ very low grades of fuel in compression ignition engines, but although these are practical in emergency, the use of a standardised grade of fuel oil is now established. A thin oil of low viscosity with a specific gravity of 0·85 is typical, the viscosity would be in the range of 2·0–6·0 centistokes at 100° F. The calorific value of this fuel differs but little from that of petrol and approximates to 19,000 B.Th.U. per lb.

Petrol Engines.

Multi-cylinder engines are universal for cars and commercial vehicles and generally embody 4, 6 or 8 cylinders: occasionally 2, 3, 5 or 12 cylinders are used but these are the exception to the rule and are usually employed in special purpose vehicles. The popularity of the two-cylinder engine for motor-cycles is increasing, particularly the vertical transverse twin. The disposition of the cylinders and the consequent order of firing affects balancing and torsional stresses. Since automobile engines are generally operated on the 4-cycle principle, each cylinder fires every two revolutions.

In a two-cylinder engine having both cylinders on the same side of the crankshaft and with the crank-pins co-incident, the result is one impulse per revolution and the reciprocating parts entirely unbalanced. With two cylinders on the same side of the crankshaft and the crank-pins at 180°, there are two impulses in one revolution and none in the next, but the reciprocating parts are balanced. In the case of a two-cylinder engine in which the cylinders are on opposite sides of the crankshaft and the crank-pins at 180°, there is one impulse per revolution and perfect reciprocal balance.

Four-cylinder in Line Engine.—The crank pins are arranged at 180°, cranks 1 and 4 being coincident as also are cranks 2 and 3. Here there are two impulses per revolution 180° apart and the reciprocating parts are balanced. The firing order may be 1, 3, 4, 2 or 1, 2, 4, 3.

Six-cylinder in Line Engine.—In this case the cranks are arranged at 120° and there are, therefore, three impulses per revolution 120° apart. The firing order may be, 1, 5, 3, 2, 6, 4, in which case cranks 1 and 2 are in the same plane; cranks 3 and 4 in another plane, and cranks 5 and 6 in a third plane, or 1, 5, 3, 6, 2, 4, in this case the two end cranks 1 and 6 are in the same plane; cranks 2 and 5 in another plane and 3 and 4 in a third plane. A variation of the latter arrangement will provide a firing order of 1, 4, 2, 6, 3, 5. Firing orders of 1, 5, 3, 6, 2, 4 or 1, 4, 2, 6, 3, 5 are preferable to 1, 5, 3, 2, 6, 4 as explosions occur in alternate ends of the shaft and never in adjacent cranks as in the latter arrangement.

Eight-cylinder in Line Engine.—In this arrangement the cranks are disposed at 90° with the result that there are four impulses per revolution 90° apart. There are very many possible arrangements of firing order but for various considerations, which it is impossible to discuss here, one of the following orders is preferable to any of the others.

(i) 1, 3, 2, 4, 8, 6, 7, 5.
(ii) 1, 3, 7, 4, 8, 6, 2, 5.
(iii) 1, 3, 2, 5, 8, 6, 7, 4.
(iv) 1, 3, 7, 5, 8, 6, 2, 4.
(v) 1, 6, 2, 4, 8, 3, 7, 5.
(vi) 1, 6, 7, 4, 8, 3, 2, 5.
(vii) 1, 6, 2, 5, 8, 3, 7, 4.
(viii) 1, 6, 7, 5, 8, 3, 2, 4.

These arrangements are obtained by means of a crankshaft having the four centre cranks in the same plane and the four end cranks in a second plane at 90° to the first, the crank pins being coincident as follows :—

Nos. 1 and 8, nos. 2 and 7, nos. 3 and 6, and nos. 4 and 5.

The most favoured firing order is 1, 6, 2, 5, 8, 3, 7, 4.

Eight-cylinder 'V' Engine.—There are two possible crank arrangements for the V-8 engine, the first in which the cranks are all in the same plane, similar to the four-cylinder in line engine, with cranks 1 and 4 coincident, also cranks 2 and 3. In the second arrangement cranks 1 and 4 are 180° apart in one plane and cranks 2 and 3 are also 180° apart but in a second plane at 90° to the first. From the point of view of balance the latter arrangement is to be preferred.

With the 180° or flat crankshaft the firing order may be 1R, 4L, 3R, 2L, 4R, 1L, 2R, 3L, or 1R, 4L, 2R, 3L, 4R, 1L, 3R, 2L.

With the 90° crankshaft the firing order is 1R, 1L, 4R, 4L, 2L, 3R, 3L, 2R.

Balancing.—It is impossible to discuss here the many and complicated problems relating to balancing; briefly, however, the main sources of vibration or unbalance are the cyclic torque variations which act upon the crankshaft assembly due to varying piston loads and connecting-rod obliquity, the unbalanced effects of centrifugal forces on rotating members, such as the crankshaft and camshaft, and the unbalanced effects caused by the reciprocating members of the engine, such as the pistons and upper parts of the connecting-rods, the valves and springs. The reciprocating forces produce rocking moments about some point in the length of the crankshaft, these are modified by the angularity of the connecting-rods which produces harmonic vibrations of 2, 4, 6 or more times the fundamental frequency. The higher orders of harmonics above the second need not be considered, but analysis of the second harmonics shows that with two cylinders on the same side of the shaft and the cranks 180° apart, fundamental vibrations are balanced but octave harmonics are not. In a two-cylinder horizontally opposed engine with cranks 180° apart, fundamentals and octaves are balanced.

In a four-cylinder in line engine, the fundamentals balance but the octaves do not; while in a six-cylinder engine both fundamentals and octaves are truly in balance.

The second harmonics of twice engine frequency act along the cylinder axis and produce a force :—

$$F = \frac{Mr\omega^2}{g}\left(\frac{r}{l}\cos 2\theta\right)$$

where θ = crank angle.
M = reciprocating mass (piston + small end).
r = crank radius.
l = length of connecting rod.
ω = angular velocity of shaft.

hence F is of maximum value when $\cos 2\theta = \pm 1$, *i.e.* at 0°, 90°, 180° and 270°.

Crankshafts.—The greater the number of cylinders, and consequently the longer the crankshaft, the greater is the tendency to torsional vibration, and this can only be overcome by increasing the dimensions, and the weight, of the shaft. The result is a shaft in which the specific stresses are low and for this reason the use of a low grade of cast semi-steel is justifiable. It is usual also to employ a friction or bonded rubber damper attached to the free end of the shaft to damp out the torsional vibration, but it is, nevertheless, impossible to eliminate it altogether, hence the use of flexible engine mountings embodying rubber cushioning blocks, to insulate the engine from the chassis frame.

For the reasons stated above the 'V'-cylinder formation is to be commended for engines having 8 or more cylinders; not only does this arrangement permit the use of a comparatively short stiff crankshaft but it enables the overall dimensions of the engine to be retained within reasonable limits for installation purposes.

From the symmetrical form of the crank arrangement it follows that the centrifugal forces due to crank-webs, crank-pins and big ends are balanced, but where adjacent crank-pins are

coincident the centrifugal and inertia forces impose a very heavy load on the common bearing which must be of adequate dimensions to support this. If adjacent cranks are placed 180° apart the mutual balance of opposing cranks is more direct, and bearing loads are much reduced. The rear main bearing must also be of adequate dimensions in order to withstand any misalignment between the flywheel and clutch.

Bearings.—Generally speaking, bearings are the product of specialists and, in view of the very high specific surface pressure imposed by modern design, 'Thin wall' or 'Strip' bearings are becoming universally popular both for big-end and main bearings. These bearings comprise a thin steel shell only about one-sixteenth of an inch thick lined with white metal, the thickness of which is kept to a minimum consistant with proper adhesion about 0·008 in. They are finished to extremely fine limits and with the perfection of surface finish that may now be obtained by diamond turning and boring operations, it is possible to assemble the shells without any further treatment whatever, the flexibility of the shell enabling the true form to be maintained by the rigid and accurately machined big-end or bearing housing. The peripheral length of the bearing must be very carefully controlled in order to provide the necessary 'nip' on assembly.

For heavy duty loading, such as occurs in high-speed compression ignition engines, copper lead or 'lead bronze' lining materials have been developed, the bearing surface being treated with an electrically-deposited flash of lead and/or indium, thermally infused into the lead.

These bearings are located by means tangs, pressed out of the shells at the bearing joint, fitting in machined slots in the big-end or bearing housing and the respective bearing caps, alternatively cylindrical dowels may be used. The crankshaft is located by means of flanged centre or rear main bearings, alternatively split bronze washers retained against rotation by means of tabs or dowel pegs may be used. The end thrust imposed by clutch withdrawal is also taken on the rear-most bearing flange or pair of split washers.

Bearing clearances are usually of the order of 0·001 in. per 1 in. shaft diameter.

The value of the PV or load factor, which is the product of bearing pressure and rubbing velocity, determines the behaviour of the bearing and is limited by the coefficient of friction and the viscosity of the oil.

Cylinders.—In the interests of rigidity, in addition to economy in manufacture, the cylinder block is usually cast in one with the upper half of the crankcase, the only disadvantage of this form of construction being that of weight, but even this is outweighed by the advantages gained.

Owing to the complicated design a free flowing iron is essential and as phosphorus is objectionable a high silicon content is used. Nickel chromium iron, of which the following analysis is typical, is in general use :—

Chromium Alloy Cast-Iron.

Total carbon	3·15
Combined carbon	0·55
Manganese	0·98
Nickel	0·09
Chromium	0·25
Sulphur	0·10
Phosphorus	0·25
Silicon	2·20

In addition to nickel and chromium, molybdenum is employed for producing close fracture grey cylinder irons having excellent wearing qualities. The percentage composition being usually within the following limits :—

Total carbon	3·1 –3·4
Graphitic carbon	2·5 –2·8
Combined carbon	0·5 –0·7
Silicon	1·8 –2·4
Manganese	0·5 –0·8
Phosphorus	0·12–0·20
Sulphur	0·10–0·12
Chromium	0·10–0·50
Nickel	0·25–1·50
Molybdenum	0·10–0·70

The tensile strengths of these irons vary from 15·5 to 18·0 tons per sq. in.

Where cost is not the primary consideration the use of detachable cylinder liners offers many advantages. A much harder material can be used such as a very hard cast iron centrifugally cast to ensure homogeneity, or nitrogen hardened steel.

Liners may be of wet or dry types, the latter having the advantage that they may be pre-finished to such a degree of accuracy as to provide a slip-fit in the cylinder block, thus enabling the liners to be fitted by hand with the engine *in situ*, without the need for special tools.

Cylinder Heads.—The contour given to the inside of the cylinder head determines the form of the combustion chamber, which itself has a profound effect on the running of the engine. A homogeneous mixture of air and petrol vapour is essential and gas turbulence during and after

compression is therefore of paramount importance. There is no unanimity of practice, but for side valve engines a variation of the form introduced by Ricardo is generally adopted (fig. 1). The combustion chamber more or less of D form, extends over the valves and something less than one-half of the cylinder bore ; the sparking plug being more or less equidistant between the axes of the cylinder and the two valves.

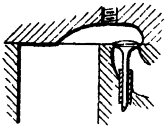


FIG. 1.

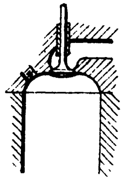


FIG. 2.

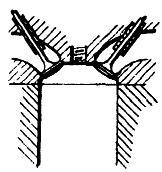


FIG. 3.

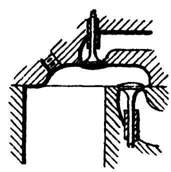


FIG. 4.

There are innumerable variations of the overhead value arrangement in which the combustion chamber approximates to a hemispherical form with the valves either vertical or inclined and the sparking plug either to one side or in the centre. Fig. 2 illustrates a typical commercial arrangement and fig. 3 the arrangement usually adopted in high performance engines.

The overhead inlet and side exhaust valve arrangement has some supporters and has the advantage of permitting the use of large inlet valves without increasing the cylinder centres ; it also enables a high compression ratio to be combined with a good form of combustion chamber and excellent scavenging. Fig. 4 illustrates this arrangement.

In all cases it is essential to provide adequate cooling for the exhaust valves and care must be taken to ensure that the water passages completely surround them circumferentially, care must also be taken to avoid any pockets where steam could accumulate.

Adequate depth of the cylinder head is necessary to ensure rigidity, ample water passages between cylinder and head are desirable and very careful attention should be paid to the positioning and quantity of cylinder head studs, a general guide for the latter being ' as many as possible.'

Cast iron is still the most popular material for cylinder heads although aluminium alloy is not uncommon.

Poppet Valves.—The effective valve area governs the entire performance of the engine and the mean gas velocity through the valve ports ranges from 250 to 370 ft. per sec. for maximum engine speeds of 3,000 to 4,000 r.p.m.

The mean gas velocity may be obtained from the following formulæ :—

$$Vg = \frac{VsN}{360Av} \text{ ft. per sec. . . . . (1).}$$

Where:

Vs = swept volume of one cylinder in cu. in.
N = crankshaft r.p.m.
Av = valve area in sq. in.

$$\text{or } Vg = \frac{VsN}{911 \cdot 4Av} \text{ . . . . (2).}$$

where Vs and Av are in c.c. and sq. c. respectively.

If the gas speeds are calculated at a crankshaft speed of 2,000 r.p.m. these formulæ become :—

$$Vg = \frac{5 \cdot 55Vs}{Av} \text{ ft. per sec. . . . (1a).}$$

$$\text{or } Vg = \frac{2 \cdot 19Vs}{Av} \text{ ft. per sec. . . . (2a.)}$$

The effective valve area can be calculated, assuming the usual face angle of 45°, as follows :—

Effective area = $3 \cdot 1416(0 \cdot 707Dh + 0 \cdot 3535h^2)$ sq. in.

where D = port diameter in ins. and h = valve lift in ins.

Valve Acceleration.—The question of the inertia forces due to the weight and acceleration (or deceleration) is of considerable importance in valve-spring and cam design. Acceleration and spring strength can be obtained as follows :—

If n = r.p.m. of crankshaft.
n_2 = r.p.m. of camshaft.
E = effective throw of cam (fig. 5).
θ = instantaneous angle of cam and tappet.
W = weight of reciprocating parts, lbs., *i.e.* valve, tappet, spring cotter and collar, and ½ the weight of the spring.
g = 32·2 ft. per sec.²
A = acceleration at any angle θ.
F = inertia force at θ.

then $A = \dfrac{4\pi^2 n^2 E \cos\theta}{3{,}600 \times 4 \times 12}$

$= \dfrac{n^2 E \cos\theta}{4{,}378}$ ft. per sec.^{2}

and $F = \dfrac{WA}{g}$

$= \dfrac{Wn^2 E \cos\theta}{141{,}000}$ lbs.

At full lift $\theta = 0$ and $\cos\theta = 1$.

whence $A \doteq \dfrac{n^2 E}{4{,}378}$ ft. per sec.^{2}

$F = \dfrac{Wn^2 E}{141{,}000}$ lbs.

Whereas a nickel-chrome high tensile steel is usually quite suitable for inlet valves, exhaust valves, in view of the higher operating temperatures, require the use of special heat-resisting steels, such as silicon-chrome, stainless steel, cobalt-chrome or the high nickel-chromium steels belonging to the austenitic class.

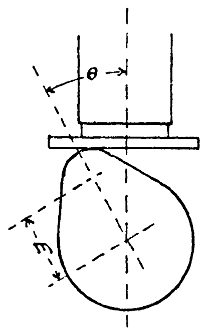


FIG. 5.

In the case of austenitic valves it is necessary to give the value stems a high surface finish in order to avoid valve guide 'pick-up' and to provide hardened tips for the tappet or rocker contact area. It is usual to weld a button of hardened steel or to stellite the tip of the stem. For heavy duty engines requiring long life a flash of hard chrome is deposited along the stem and the seating face is stellited. While valves must be light, they must possess sufficient material to conduct away the heat.

Valve guides are of centrifugally cast iron, and tappets usually of the flat 'barrel' type, the latter are often offset in relation to the cam in order to promote rotation of the tappet and thus equalise the wear; for the same reason the valve and tappet are frequently offset in side-valve engines.

Valve seat inserts in heat resisting alloy are common, particularly in commercial vehicle engines where they are usually stellite faced. A recent development for aluminium cylinder heads is the use of a high expansion coefficient nickel-chromium-manganese steel for valve seat inserts.

Valve Springs.—Valve springs must be heat treated after coiling, apart from the maximum **stress, the stress range must be kept as low as possible to avoid effects of fatigue. This should be about one-half the maximum stress and the latter may be from 20,000 lbs. per sq. in. to 60,000 lbs. depending on the material used. Chrome vanadium steel gives the best results. Failures often occur through resonance in the spring where its natural frequency coincides with the forced vibrations. When the forced vibration is a simple multiple of the free wave-length of the spring a condition of resonance occurs. If the wave-length is long, and such that an integral number of free wave-lengths of the spring are completed during the cycle of the valve, the forces will be** in phase and the resulting wave in the spring becomes cumulative and destructive. The natural frequency of a helical spring is

$$N = 589 \cdot \sqrt{\frac{R}{W}} \text{ cycles/minute.}$$

where **R = rate of spring in lbs. per in.**
W = weight of active portion of spring in lbs.

The following formulæ can be used :—

d = diameter of wire in ins.
r = mean radius of coil in ins.
N = number of coils.
C = a constant based on modulus of transverse elasticity = 180,000.
K = stress per sq. in., *e.g.* 20,000 lbs. per sq. in.
Safe load = Kd^3/r.

Load to deflect spring 1 in. (rate per in.) = Cd^4/Nr^3.

In high speed engines it is usually necessary to use two helical springs to each valve, the springs being nested; besides permitting a lower stress in the steel, this arrangement tends to damp out resonant vibrations.

TABLE III.—CONSTANTS OF ROUND STEEL WIRE FOR HELICAL SPRINGS.

S.W.G.	Equivalents in.	Equivalents mm.	Cube	4th Power
7/0	·500	12·699	·12500	·06250
6/0	·464	11·785	·09990	·04635
5/0	·432	10·972	·08062	·03483
4/0	·400	10·159	·06400	·02560
3/0	·372	9·448	·05148	·01915
2/0	·348	8·839	·04214	·01467
0	·324	8·229	·03401	·01102
1	·300	7·620	·02700	·008100
2	·276	7·010	·02102	·005803
3	·252	6·400	·01600	·004033
4	·232	5·892	·01249	·002897
5	·212	5·384	·009528	·002020
6	·192	4·876	·007078	·001359
7	·176	4·470	·005452	·0009595
8	·160	4·064	·004096	·0006554
9	·144	3·657	·002986	·0004300
10	·128	3·251	·002097	·0002684
11	·116	2·946	·001561	·0001811
12	·104	2·641	·001125	·0001170
13	·092	2·336	·0007787	·00007164
14	·080	2·056	·0005120	·00004096
15	·072	1·828	·0003732	·00002687
16	·064	1·625	·0002621	·00001678
17	·056	1·421	·0001756	·000009834
18	·048	1·218	·0001106	·000005308
19	·040	1·016	·00006400	·000002560
20	·036	·9140	·00004666	·000001679
21	·032	·8124	·00003277	·000001049
22	·028	·7109	·00002195	·000000615
23	·024	·6093	·00001382	·000000332
24	·022	·5585	·00001065	·000000234
25	·020	·5078	·000008000	·00000016

Valve Timing.—The valve timing must be such as to furnish optimum results in respect of charging the cylinder with explosive mixture, and also of scavenging the burnt gases. It is largely a question of compromise between disadvantageous circumstances.

A typical timing arrangement would be as follows :—

Inlet opens	5 to 15° before T.D.C.
Inlet closes	40 to 45° after B.D.C.
Exhaust opens	45 to 55° before B.D.C.
Exhaust closes	5 to 15° after T.D.C.

It is thus seen that inlet and exhaust valves are *both* open at top and bottom dead centres, with an overlapping period of 10° to 30° at the top, and from 85° to 100° at the bottom. The period of overlap being greater for high- than for low-speed engines and, in the case of high-speed racing engines may be as much as 80° at T.D.C. and 160° at B.D.C.

Pistons.—These exhibit great variety of design. Lightness combined with strength is the first essential, and apart from transmitting the piston pressure to the crankpin, the functions of the piston include dissipation of heat to the cylinder walls, preventing passage of oil into the combustion chamber, and the provision of adequate support for the gudgeon pin. They are generally relieved on the sides to reduce friction, and the skirt is ground tapered and oval. Three or four rings are usually fitted above the gudgeon pin the bottom of which is an oil control or scraper ring, occasionally a second oil control ring is fitted below the gudgeon pin. It is quite common in commercial vehicle engines to provide a groove for a bottom oil control ring below the gugdeon pin but to omit the ring when the engine is originally built. The ring being fitted by the operator as dictated by oil consumption.

Piston rings are of cast iron, concentric, and hammered to equalise the spring effect throughout the circumference. They are ground on the outside diameter and both sides and fit closely into the grooves.

Owing to the high coefficient of expansion of the aluminium alloy used for piston manufacture a liberal clearance is necessary at the piston crown which is decreased as described above, as the skirt is approached. The actual amount of clearance is dependent on the general design both of piston and cylinder, the service for which the engine is intended, and other considerations.

Piston Inertia.—A simple method of calculating piston inertia is due to Ricardo, as follows :—

p = inertia pressure in lbs. per sq. in of piston area.

w = total weight of reciprocating parts (including connecting rod) in lbs. per sq. in. piston area.

n = r.p.m.

s = stroke in ins.

r/c = ratio of crank to connecting rod.

then at top of stroke

$$p_t = 0{\cdot}0000142 wn^2s\,(1 + r/c).$$

and at bottom of stroke

$$p_b = 0{\cdot}0000142\, wn^2s\,(1 - r/c).$$

It generally happens that the nertia pressure is greater than the gas pressure. The former occurs four times in two revolutions and the latter only once in the same period, the combined effects are thus best shown by means of a polar diagram.

Piston Acceleration and instantaneous velocity are determined by the usual methods.

Gudgeon Pins.—Made of casehardened steel of very light tubular section, ground and externally lapped; they always float in the piston and frequently in the small end of connecting rod as well. In the pistons the pin runs direct on the aluminium and generally with only oil vapour lubrication.

Fully assembled with gudgeon pins and rings, pistons must be held closely to limits of weight; a maximum tolerance of $\pm\frac{1}{4}$ ounce is permissible, but the best practice is less than this to avoid a rough running engine.

Connecting Rods.—Connecting rods are of steel, stamped to required form, but when lightness is essential duralumin or hyduminium stampings are used, alternatively the steel rod is machined all over. They are of H section with the small end either solid or semi-split, the big-ends in halves and with only two bolts. Steel rods are bronze bushed for the gudgeon pins ; light alloy rods not.

For the big-end bearings, see ' Bearings.'

Camshafts.—These, as in the case of crankshafts, are not subject to high specific stresses as they must be of adequate diameter to ensure rigidity ; this is the more easily attained since the journal bearings must be large enough to pass the cams through.

The design of cams is too complex a subject to be treated here, and involves prolonged study to reconcile antagonistic factors, *e.g.* adequate lift of valve without prohibitive acceleration,

rapid closing without noise, etc.; again, base circles must not be so large as to cause too high rubbing speed; finally it must not be of such a contour that it cannot readily be ground. See Section XXVIII (III), page 219.

Camshaft Drives.—**With a side valve engine the camshaft is inside the crankcase and a simple 2 to 1 reduction gear, either spur or helical, suffices. The same applies to overhead valves where the shaft remains in the crankcase, and operates the valves through push rods. Where, however, the camshaft is in the cylinder head more complex gears are necessary. Dr. H. E. Merritt (*Proc. I.A.E.*) remarks as follows:—**

With side-by-side valves there are two general types—(*a*) chain; (*b*) spur or helical gear. Silence of operation is important, and the chief obstacle to this centres round the uneven torque of the drive. This may result in a wave motion in a chain drive, introducing considerable stresses and limiting the allowable velocity of the chain. With tooth gear the silence of operation depends largely on the accuracy of manufacture. In many cases helical teeth are used in place of the straight tooth, with improved performance. This introduces, however, an end-thrust on the crankshaft and camshaft, and a transverse torque on the intermediate gear. In correctly designed helical teeth the degree of overlap should not be less than one pitch, and with double helical gear end-thrust is eliminated.

Camshaft brakes may be used to secure uniformity of driving torque, in which case the breaking torque must be at least equal to the maximum forward accelerating torque, due to a closing valve spring.

Another method is to drive an auxiliary, such as the fan, water, or oil pump from the free end of the camshaft, thereby providing the necessary constant resistance.

The majority of camshaft drives are taken from the front end of the crankshaft with the object of accessibility, but in this position crankshaft torsional vibrations may produce severe fluctuations in tooth pressure. For minimising gear noise, the use of non-metallic gears is very common. Due to their resilience and high inter-molecular friction, the energy of tooth impact is absorbed, which prevents the bounce of the teeth.

Owing to the greater centre distance between an overhead camshaft and the crankshaft a variety of drives is possible, and representative arrangements are shown in fig. 6.

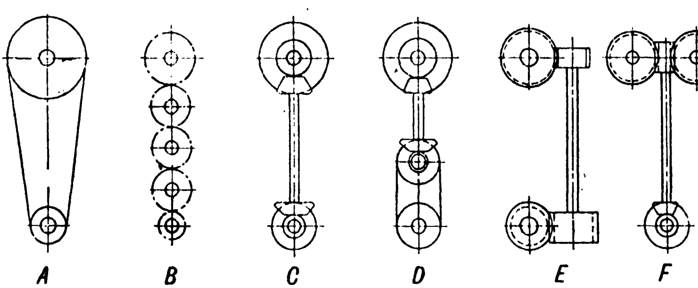


FIG. 6.—Types of Overhead Camshaft Drive.

A. Chain. D. Chain-bevel.
B. Spur Gear. E. Helical gear.
C. Bevel. F. Bevel-worm.

The train of spur gears (B) has a large number of points at which wear may occur, and is liable to be noisy. The bevel drive demands care in layout. Bevel gears require very accurate location in every direction, means for axial adjustment, and thrust bearings. Straight-tooth bevel gears are the most difficult of all gears to get to run quietly, and the tendency is to use spiral bevels, but this type calls for the greatest accuracy in cutting and assembly.

The unequal expansion of the cylinder block and the vertical shaft has an important effect on gear mesh and tappet clearance, and this can be provided for by the use of a telescopic vertical shaft. With the spiral and worm drives a wide range of gear diameters is possible. Another advantage is that there is no necessity for precise adjustment for end float on either crankshaft or camshaft, or expansion of the vertical shaft. On the other hand, the contact between spiral gears is only point contact and the intensity of tooth pressure is very high.

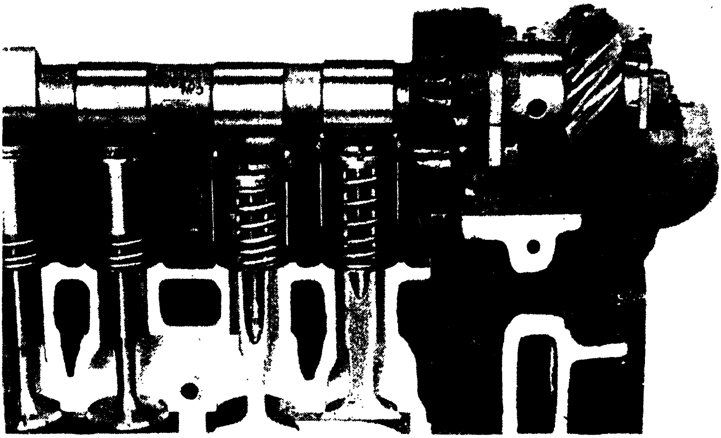


FIG. 7.

Fig. 7 illustrates the overhead camshaft and valve gear on the Wolseley ' Four Fifty ' and ' Six Eighty ' engines in which the drive is provided by means of spring-loaded split wormwheels.

Engine Efficiency.—**The mechanical efficiency can best be determined by running on a dynamometer. The torque is first measured over the whole speed range and from this the speed at maximum brake h.p. is determined. The engine is then run at this speed with one of each of its cylinders cut out in turn. The h.p. missing under each of these conditions is the indicated h.p. of the cylinder cut out, and their sum is the total indicated h.p. Thus if**

T = full torque at maximum power lbs .ft.

T_1 = mean torque with one cylinder cut out.

N = number of cylinders.

T_2 = indicated torque, lbs. ft.

$T_2 = N(T - T_1)$ from which total i.h.p. can be calculated by the formula

$$\text{i.h.p.} = \frac{nT_2}{5252} \qquad n = \text{r.p.m.}$$

Volumetric Efficiency can be measured by connecting the carburettor air intake to an accurate meter, *e.g.* an accurately balanced gas holder, and comparing the measured volume drawn in with the swept volume of the engine.

It can also be shown that the density of the charge in the cylinder at the end of the suction stroke is a direct measure of its volumetric efficiency ; it is, therefore, only necessary to compare the density of the charge with the density of the ambient air in order to obtain the volumetric efficiency.

The volumetric efficiency representative of the average modern engine is 0·75 to 0·85.

Thermal Efficiency.—**This is the proportion of the heat liberated by the combustion of the fuel which appears as B.Th.U. on the piston, *i.e.* i.h.p. × 42·4, compared with the calorific value of the fuel consumed. The following are typical thermal efficiencies observed.**

Compression	ratio	4·0	observed	efficiency	28·2	per cent.
,,	,,	5·0	,,	,,	32·2	,,
,,	,,	6·0	,,	,,	35·3	,,
,,	,,	7·0	,,	,,	37·7	,,

TABLE IV.

ENGINE CHARACTERISTICS, 1949. UNDER 1·500 LITRES.

Make and Type.	Valve Position.	No. of Cyldrs.	Bore. mm.	Stroke. mm.	Capacity Litres.	Compression Ratio.	Stroke Bore Ratio.	Max. b.h.p.	Max. r.p.m.	Max. BMEP	Max. Torque.	Speed at Max. BMEP and Torque.	Power Capacity Ratio. b.h.p./litre.	No. of Main Bearings.	Piston Speed at Max. r.p.m. ft./min.
Austin, 'A40'	OH	4	65·5	89	1·200	7·2	1·36	40	4,300	121	59	3,000	33·4	3	2,510
Ford 8, 'Anglia'	SS	4	56·6	92·5	0·933	6·3	1·63	23·4	4,000	96·5	36·4	2,300	25·1	3	2,430
" 10, 'Prefect'	SS	4	63·5	92·5	1·172	6·16 *6·6	1·456	30·1	4,000	97·8 *109·5	46·4 *52	2,400	25·8	3	2,430
Hilman, 'Minx'	SS	4	63	95	1·185	6·3	1·51	35	4,100	112·5	54	2,400	29·6	3	2,555
H.R.G., '1100'	OHC	4	60	95	1·074	7·75	1·58	44	5,200	123	51	3,500	41	3	3,240
" '1500'	OHC	4	68	103	1·496	7	1·515	61	4,800	128	77	3,000	40·8	3	3,240
Jowett, 'Javelin'	OH	4	72·5	90	1·485	7·2	1·24	52·5	4,500	126·5	76	2,600	35·2	3	2,660
Lanchester	OH	4	63·5	101·6	1·287	7·4	1·60	40	4,200	115	59·7	2,000	31·1	3	2,795
Lea-Francis, 12 h.p. Sports	OH	4	69	100	1·496	7·25	1·45	70	5,100	131	79	2,500	46·8	3	3,340
" 12 h.p. Saloon	OH	4	69	100	1·496	7·25	1·45	50	4,700	116	71	2,000	33·4	3	3,080
M.G. Midget (Series 'TC')	OH	4	66·5	90	1·250	7·25	1·353	54·4	5,200	125	63·7	2,600	43·5	3	3,068
" " (Series 'Y')	OH	4	66·5	90	1·250	7·2	1·353	46	4,800	116	58·5	2,400	36·8	3	2,835
Morris Minor (Series 'MM')	SS	4	57	90	0·919	6·5	1·58	29·5	4,400	113	44·6	2,400	32·1	3	2,600
" Oxford (Series 'MO')	SS	4	73·5	87	1·476	6·8	1·185	40·5	4,200	112	67	2,000	27·4	3	2,395
Riley, 1½ litre	OH	4	69	100	1·496	6·8	1·45	55	4,500	128	76	2,800	36·8	3	2,950
Sunbeam Talbot '80'	OH	4	63	95	1·185	6·88	1·51	47	4,800	127	61	3,200	39·7	3	2,992
Vauxhall, 'Wyvern'	OH	4	69·5	95	1·442	6·4	1·37	35	3,600	119	68	2,000	24·3	3	2,245
Wolseley, 'Four Fifty'	OHC	4	73·5	87	1·476	7	1·185	51	4,400	120	71·6	2,700	34·6	3	2,510

* 'De Luxe' Model.

TABLE V.

ENGINE CHARACTERISTICS, 1949. OVER 1·500 LITRES.

Make and Type.	Valve Position.	No. of Cyldrs.	Bore mm.	Stroke. mm.	Capacity Litres.	Compression Ratio.	Stroke Bore Ratio.	Max. b.h.p.	Max. r.p.m.	Max. BMEP	Max. Torque.	Speed at Max. BMEP and Torque.	Power Capacity Ratio. b.h.p./litre.	No. of Main Bearings.	Piston Speed at Max. r.p.m. ft./min.
Aston Martin '2-litre Sports'	OH	4	82·55	92	1·970	7·25	1·114	90	4,750	135	107·5	3,000	45·7	5	2,860
Alvis 'T.A. 14'	OH	4	74	110	1·892	6·725	1·487	65	4,000	112	95	3,000	34·4	3	2,890
Armstrong Siddeley '16'	OH	6	65	100	1·991	7·0	1·54	70	4,200	118	95	2,500	35·2	—	2,760
Austin '16' and 'A.70'	OH	4	79·4	111·1	2·199	6·85	1·4	67	3,800	130	112	2,200	32·9	3	2,770
,, 'A.90'	OH	4	87·3	111·1	2·660	7·5	1·273	88	4,000	134	140	2,500	33·1	3	2,920
,, 'A.135'	OH	6	87·3	111·1	3·992	6·8	1·273	130	3,800	132	212	2,400	32·6	4	2,770
Bristol '400'—85A.	OH	6	66	96	1·971	7·5	1·455	80	4,200	131·7	105	3,000	40·6	4	2,650
,, '400'—85B.	OH	6	66	96	1·971	7·5	1·455	85	4,500	—	—	—	43·2	4	2,830
Citroen 'Light 15'	OH	4	78	100	1·911	6·2	1·283	55·7	3,800	—	—	—	29·2	3	2,490
,, 'Six-Cylinder'	OH	6	78	100	2·867	6·2	1·283	76	3,800	—	—	—	26·5	4	2,490
Ford 'V.8'	SS	8	66·04	81·28	2·228	6·6	1·23	60	3,500	104	94	2,500	26·9	3	1,865
,, 'V.8' Pilot	SS	8	77·79	95·25	3·622	6·15	1·225	85	3,500	102·4	150	1,500	23·5	3	2,188
Humber 'Hawk Mk. III'	SS	4	75	110	1·944	6·4	1·466	56	3,800	122·5	96·6	2,000	28·8	3	2,742
,, 'Supersnipe' and 'Pullman'	SS	6	85	120	4·086	6·25	1·412	100	3,400	120	197·7	1,200	24·6	4	2,675
Jaguar 'XK 100'	2-OHC	4	80·5	98	1·995	7·0	1·218	105	5,400	140	113	3,000	52·7	3	3,470
,, 'XK 120'	2-OHC	6	83	106	3·442	7·0	1·277	160	5,400	140	195	2,500	46·5	7	3,760
,, 'Mk. V—2½ litre'	OH	6	73	106	2·664	7·3	1·453	102	4,600	125	136	2,200	38·3	7	3,200
,, 'Mk. V—3½ litre'	OH	6	82	110	3·486	6·75	1·342	125	4,250	130	184	2,000	35·9	7	3,070
Jensen '4-litre-Straight 8'	OH	8	85	85	3·860	6·25	1·00	130	4,300	123·5	193	2,500	32·8	9	2,400
Lagonda '2½ litre'	2-OHC	6	78	90	2·580	6·5	1·154	105	5,000	130	135·6	3,250	40·7	4	2,953
Lea-Francis '14 Saloon'	OH	4	75	100	1·767	7·25	1·335	70	4,700	132	94	2,250	39·7	3	3,085
,, '14 Sports'	OH	4	75	100	1·767	8·0	1·335	87	5,200	160	114	3,500	49·3	3	3,410
Morris Six—Series 'M.S.'	OHC	6	73·5	87	2·215	7·0	1·184	70	4,600	115	103	2,400	31·6	4	2,626
Riley '2½ litre'	OH	4	80·5	120	2·443	6·8	1·49	100	4,500	135	134	3,000	41·0	3	3,540
Rolls-Royce, 'Silver Wraith'	OHI-SE	6	88·9	114·3	4·256	6·4	1·287	126	3,750	132	228	1,700	29·6	7	2,810
Rover '60'	OHI-SE	4	69·5	105	1·595	7·1	1·51	50	4,000	130	84	2,000	31·4	3	2,760
,, '75'	OHI-SE	6	65·2	105	2·100	7·25	1·61	72	4,000	130	110·7	2,000	34·3	4	2,760
Standard 'Vanguard'	OH	4	85	92	2·088	6·7	1·082	68	4,200	131	110·8	2,300	32·6	3	2,530
Sunbeam Talbot '90'	OH	4	75	110	1·944	6·59	1·467	64	4,100	128	100	2,300	32·9	3	2,958
Triumph '1800 Saloon'	OH	4	73	106	1·776	6·7	1·453	63	4,500	—	—	—	35·5	3	3,130
Vauxhall 'Velox'	OH	6	69·5	100	2·275	6·75	1·44	55	3,300	116	106	1,200	24·2	4	2,165
Wolseley 'Six Eighty'	OHC	6	73·5	87	2·215	7·0	1·184	72	4,600	118	105	2,400	32·5	4	2,626

Explosion Pressures.—The general expression for the explosion pressure is :—

$$P_s = \frac{0\cdot3\,(r-1)}{1-r^{-\cdot3}}\left\{P_i + P_o \cdot \frac{r}{r-1} \cdot \frac{r^{\gamma'-1}-1}{\gamma'-1}\right\}$$

where

r = compression ratio.

P_s = explosion pressure, lbs. per sq. in. absolute.

P_o = pressure at the end of the suction stroke, which is assumed to lie between 12 and 14 lbs. per sq. in. absolute.

P_e = brake mean effective pressure, lbs. per sq. in.

γ = ratio of specific heats ; this for the explosion curve is assumed to be 1·3, and for the compression curve, where the ratio is denoted by $\gamma' = 0\cdot64 + 0\cdot05P_o$.

P_i = theoretical mean indicated effective pressure, and may be written $P_i = \frac{P_e}{\eta K}$ where η is the mechanical efficiency, and K the diagram factor representing the proportion of the actual indicator diagram to the sharp-cornered constant-volume cycle diagram assumed in the calculations.

The above equation may be reduced to the equivalent form :—

$$P_s = (0\cdot72 + 0\cdot482r)\left\{\frac{P_e}{\eta K} + 5\cdot427 + 0\cdot780\,P_o + r(0\cdot642\,P_o - 4\cdot801)\right\}$$

which agrees with above exact equation to within one part in 400.

A well designed engine may be expected to show 120 lbs. per sq. in. b.m.e.p. running under good normal conditions, but (see Supercharging) far higher pressures are frequently attained.

Compression-Ignition Engines.

Much of the foregoing mechanical data relating to petrol engines is applicable to the compression-ignition engine (also known as the oil engine, or popularly though somewhat erroneously as the diesel engine). Parts subject to forces arising from gas pressure will usually be of heavier construction, however, as higher maximum cylinder pressures are encountered.

C.I. engnies operating on both four- and two-stroke cycles are in common use for stationary and marine purposes, but the four-stroke cycle is almost universal (with a few notable exceptions) for the higher speed engines used in road transport vehicles. The fuels used vary from heavy end cuts for the large slow-speed engines to light gas oil distillates for the high-speed road transport engines. The term ' heavy oil engine ' is thus a misnomer in the latter case.

In all modern engines, the so-called ' solid injection ' principle is used, in contradistinction to the ' air blast injection ' system used on earlier engines. The fuel is sprayed towards the end of the compression stroke, through an injector into the air contained in the combustion space, by means of an engine-driven pump, which usually serves both to time the injection and meter the precise quantity of fuel required to carry the engine load (see under Fuel Injection Equipment). Compression ratios vary between 12 and 20 to 1, giving rise to compression pressures of from 400–600 lb./in.^{2} The injection pressure may rise to as high as 10,000 lb./in.^{2}, and it is the conversion of the pressure energy into velocity at the injector nozzle which produces the required atomisation of the fuel into a finely divided spray, able to penetrate the dense, highly compressed air in the combustion chamber and seek out the oxygen required for complete combustion. It is the high expansion ratio associated with these high compression ratios which gives rise to the greater thermodynamic efficiency of the C.I. engine in comparison with the petrol engine, in which compression ratio is limited by the onset of detonation. There is no similar limitation in the case of the C.I. engine although ratios higher than these quoted are not used as any further slight gain in thermodynamic efficiency is offset by reduction in mechanical efficiency.

Following upon the commencement of injection at a point just before top dead centre on the compression stroke, combustion proceeds in three more or less distinct stages, as first put forward by Ricardo. First there is a delay period during which no tangible combustion takes place, but in which the particles of fuel first injected are being raised in temperature by contact with the hot air, enabling surface vaporisation to take place. The length of this delay period varies with the ignition quality of the fuel and the temperature and density of the air into which it is injected, also to some extent on the degree of atomisation of the fuel and the turbulence of the air. The second stage follows upon the initiation of combustion at the end of the delay period, and in it combustion spreads almost instantaneously through the mixture already present in the cylinder, producing a rapid rise in pressure to a value approaching the maximum which will be reached during the cycle. It is this high more or less uncontrolled rate of pressure rise which produces the so-called ' diesel knock ' commonly obtained with these engines. The high rate of pressure rise and the associated knock can be reduced by keeping the quantity of fuel introduced during the delay period to a minimum, either by shortening the delay period by the use of high ignition quality fuels or by reducing the rate of, or interrupting, fuel injection during the delay period. The third stage of combustion is that in which fuel which may still be issuing from the injector is burnt under comparatively controlled conditions according to the rate of fuel feed, and serving to maintain pressure near the maximum until fuel injection is cut off.

The cycle is thus ideally a mixture of a constant volume (or Otto) cycle and the constant pressure (or Diesel) cycle. In practice the second and third stages tend to merge indistinguishably, particularly in high-speed engines where the length of the delay period in terms of crank angle

necessitates a large proportion of the fuel being burnt at constant volume. The diagram (fig. 8) showing pressure on a line crank angle base is typical of the latter engines.

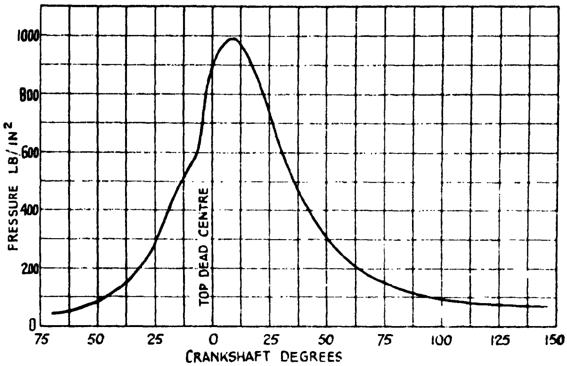


FIG. 8.

Maximum pressures usually occur shortly after top dead centre, and range from 600–1,600 lbs. per sq. in. depending on compression ratio and on the relative proportions of constant volume and constant pressure combustion. This mixed cycle has an efficiency intermediate between that of the Otto and Diesel cycles, and which will be higher with the greater proportion of combustion taking place at constant volume. It follows that the efficiency of the cycle tends to increase with reduction in quantity of fuel injected as engine load is reduced since the constant pressure stage becomes progressively diminished or even eliminated. This improvement in efficiency with reduction in load, modified of course by an increasing proportion of mechanical losses, is a further factor in the overall improvement in the efficiency of the C.I. engine in comparison with the petrol. The petrol engine is restricted in the range of air-fuel ratios in which spark-ignition can take place; hence both air and fuel charges must be reduced (by throttling the mixture) when load is reduced, and no corresponding improvement in cycle efficiency takes place.

Combustion Chambers.—Almost every manufacturer has his own particular variety of combustion chamber, but these may be grouped into four main categories, with however, numerous borderline and overlapping examples. The diagrams (fig. 9) show typical examples of each group.

(1) *Open Chamber or Direct Injection.*

The combustion chamber is formed by a recess provided in the cylinder head or in the crown of the piston, or partly in both, into which fuel is directly injected. In view of the necessity of accommodating the values in the cylinder head it is usual for the chamber to be situated in the piston crown.

(2) *Precombustion or Antechamber.*

A portion of the air is transferred during the compression stroke into a chamber separated from the engine cylinder by a restricting orifice or orifices, and in which injection and the initial stages of combustion take place.

(3) *Swirl Chamber.*

Somewhat akin to (2), but the connecting passage is relatively unrestricted, and is so arranged as to give an ordered rotation or swirl to the air transferred to the chamber during the compression stroke.

(4) *Air Cell.*

Air is again partly transferred to a separate chamber, but fuel is not injected directly into the latter but is arranged to spray from outside into the air passing through the connecting passage during the compression and expansion strokes.

The aim of all these combustion chambers is to bring the particles of fuel issuing from the injector nozzle into contact with the air in the combustion space in such a manner that each particle seeks out its own proportion of unused air and is thus enabled to burn completely without the mixture at any part becoming over rich. It will be realised that this ideal is very difficult to achieve as fuel usually enters from only one point in the combustion chamber, and it becomes progressively more difficult for the particles entering later to reach air which has not been consumed by those burning in the earlier stages.

For this reason, it is seldom that more than 70 per cent. of the air available can be utilised without signs of local incomplete combustion, as indicated by the release of free carbon or smoke in the exhaust.

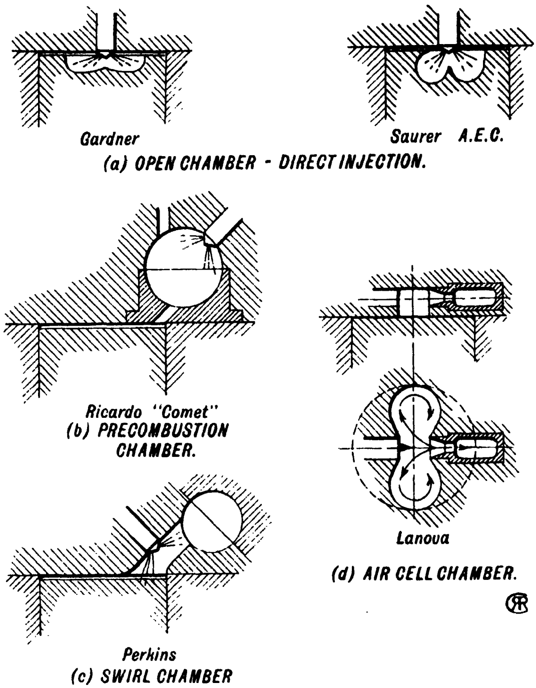


FIG. 9.

The relative importance of air movement and fuel droplet dispersion in bringing about the maximum utilisation of air varies in the four categories of combustion chamber. Type (1) can produce only a limited degree of air movement, and therefore relies greatly on suitable atomisation, penetration and direction of the fuel sprays. Types (2) and (4) rely almost entirely upon air turbulence, in these cases of a more or less indiscriminate nature, while in Type (3), air movement. although of an ordered nature, can have insufficient intensity to play a major part in the process. These later types can therefore operate with less refined injection equipment, but suffer in efficiency in comparison with type (1) owing to increased thermal and pumping losses brought about by the greater areas of exposed surface available for cooling losses, and by the transfer of air through restricting passages.

In Great Britain and Europe, owing to the high importance attached to fuel consumption, the direct injection engine has become increasingly popular, but in the U.S.A. the indirect types (2, 3 and 4) still hold sway, partly because they can generally accept a wider range of fuel qualities and partly because they can give good combustion over a wider range of rotational speed. The increase of fuel consumption of the indirect over the direct type is of the order of 5 to 15 per cent.

Two-stroke engines.—Compression ignition is particularly suitable for the two-stroke cycle, as there is no problem of direct loss of mixture during the scavenging process as with the carburettor engine. Inlet ports are piston controlled, and exhaust ports similarly in the case of the 'Loop Scavenge' engines, or by mechanically operated exhaust poppet valves in the head in the case of 'Uniflow' engines. Sleeve valves have also been employed. By careful study of the timing of inlet and exhaust events and of the scavenging process, both systems have attained a high degree of efficiency in recent years, and develop outputs closely approaching twice that from four-stroke engines of the same capacity. A blower has to be provided to obtain such results, however, partly offsetting the size and weight advantage, and the two-stroke engine presents more severe problems of waste heat disposal and injection equipment design.

Construction.—Owing to the inherent higher gas pressures, the scantlings of compression-ignition engines must be more generous than with spark-ignition engines of similar duty, and piston and bearing areas must be adequate for the increased gas and inertia loadings. The weight therefore tends to be correspondingly greater but adoption of improved materials has kept the increase to a minimum, and except for high duty applications such as for aircraft, C.I. engines are seldom objected to on this score. Specific output for a given capacity and speed of rotation is now little below that for spark-ignition engines, so that unless the more limited speed range is a factor, C.I. engines are not notably larger in size.

Fuel Injection System.

The fuel injection equipment of a ' solid injection ' type oil engine comprises the fuel injection pump, the injector nozzles and nozzle holders, and a suitable fuel filter. In addition, a fuel feed pump is included where there is insufficient gravity head from the fuel tank or where the latter is below injection pump level. A governor may be fitted to the injection pump as required for the particular duty. The fuel filter is a very necessary part of the system, as the finest dust or other abrasive material can ruin an injection pump if allowed to pass through with the fuel.

Fuel Injection Pump.—The great majority of oil engines are equipped with injection pumps manufactured by specialists in this field. The class of fit and finish demanded by the pump elements, delivery valves, etc., is of an exceptionally high standard, and calls for special skill and manufacturing facilities quite beyond those of the general engineering shop. The function of the injection pump is to meter the fuel to the highest degree of accuracy and to deliver it into the engine cylinder in quantities proportional to the load at any moment, carefully timed and at a pressure of 1,200 to 4,500 lb. per sq. in. (80 to 300 atm.) or even higher, to secure adequate atomisation and penetration.

The ' jerk ' type injection pump is almost universal; single and multiple element pumps are in common use, each cylinder being supplied by its own element. There are two main types of pump, the enclosed camshaft type complete with tappet mechanism, and arranged for drive from a suitable shaft or point on the engine, and the flange-mounted type for engines in which the camshaft forms a part of the engine proper.

Size of Injection Pump Required.—In making a preliminary selection of the size of fuel injection pump to employ for any given engine, a number of factors have to be taken into account; the following formula is suggested by C.A.V. Ltd., to enable the pump delivery to be determined :—

$$F = \frac{\text{b.h.p.} \times 3{,}520}{\text{r.p.m.}}.$$

Where F = required fuel delivery in cu. mm. per cycle per cylinder.
b.h.p. = maximum output per cylinder.
r.p.m. = camshaft speed tor four-cycle engines and crankshaft speed for two-cycle engines.
3,520 = a constant based on average fuel consumption and specific gravity of gas oil.

In selecting a pump size by means of this formula, it should be borne in mind that for average engine applications the useful output from any fuel pump plunger is usually not more than 70 per cent. of the maximum value calculated from the pump dimensions. Fig. 10 illustrates, in part section, a typical four-cylinder enclosed camshaft injection pump.

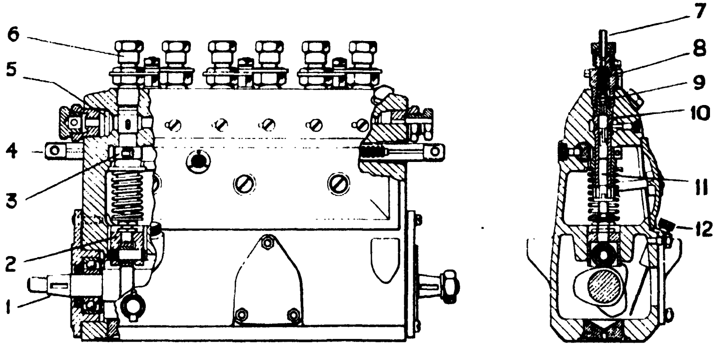


FIG. 10.

1. Camshaft.
2. Tappet.
3. Toothed quadrant.
4. Control rod.
5. Oil gallery.
6. Delivery valve nut.
7. Delivery pipe.
8. Delivery valve spring.
9. Delivery valve.
10. Pump barrel.
11. Pump plunger.
12. Dipstick.

Governors.—Various types of governor are fitted to compression ignition engines according to the duty; all of these control the speed and power of the engine by regulating the amount of oil injected. The usual type of C.A.V. mechanical governor for road vehicles regulates both maximum and minimum (idling) speeds within predetermined limits, while still leaving the control of the engine between these points directly under the influence of the accelerator pedal. This governor comprises centrifugally actuated weights complete with suitable linkage, which transmits the motion of the weights to the injection pump control rod. For marine, agricultural and certain industrial duties, a variable speed governor is often fitted. Another governor in extensive use is the pneumatic type; the pump control rod in this case is operated by a diaphragm, the movement of which is effected by means of inlet manifold pressure variation. A flexible pipe couples the diaphragm chamber with a venturi unit in the manifold, and the induction pressure differences are thus utilised to operate the pump control rod.

Fuel Oil Filters.—A typical fuel oil filter is shown in section, in fig. 11; here the filter medium consists of a felt pack element, and with relatively clean fuel and filter element a flow capacity of 3·5 to 4·5 pints/min. is available. The fuel, on entering through the inlet connection in the top

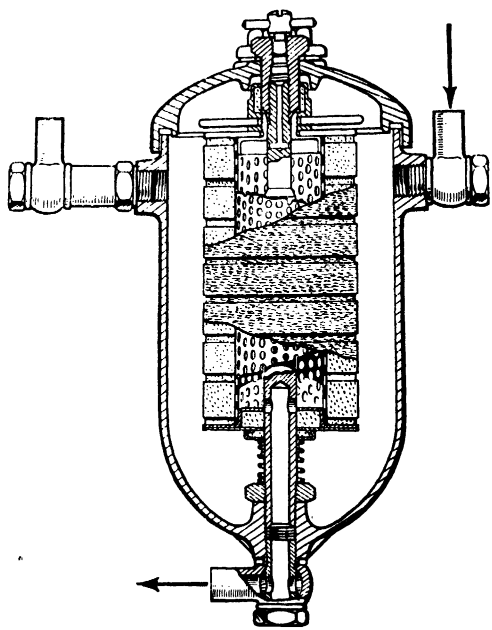


FIG. 11.

of the bowl, passes through the felt pack element and the perforated cylinder. It then flows into the stand pipe, and out into the main pipe line through the outlet connection in the base of the bowl. The filter can be dismantled with ease for cleaning and inspection by unscrewing the cap nut and removing the cover. The filter element can then be easily withdrawn from the bowl. Both 'downflow' and 'crossflow' types of filter are commonly supplied.

Injector Nozzles and Nozzle Holders.—The type of nozzle used depends on the requirements of the combustion chamber; typical types of chamber have been referred to earlier, and it will be clear that a considerable variety of nozzle designs is called for. A number of typical types is illustrated.

Fig. 12 (C) shows a single hole nozzle; the fuel is sprayed through the single central orifice when the needle is lifted by pump pressure. The hole can be any diameter from 0·2 mm. (0·008 in.) upwards. A variation of this type is the conical end nozzle, as at D, fig. 12.

In this the hole is at an angle to the centre line as required. It will be noted that the fuel reaches the nozzle tip through holes drilled from the groove of semi-circular section in the nozzle face to the oil gallery just above the needle valve seat.

Multi-hole nozzles, a typical example of which is seen in fig. 12 (B), can have any number of holes up to seven drilled in a small ' dome ' or ' teat ' formed under the nozzle mouth ; the holes are usually arranged radially in a circle about the axis of the nozzle, the number, size and ' hole angle ' of the holes depending on the requirements of the engine.

In direct-injection combustion chamber engines where, owing to the restricted space between valves in the cylinder head it is not possible to provide cooling of the nozzle in the usual way, an alternative form of nozzle has been developed. This type is known as the ' long stem ' nozzle

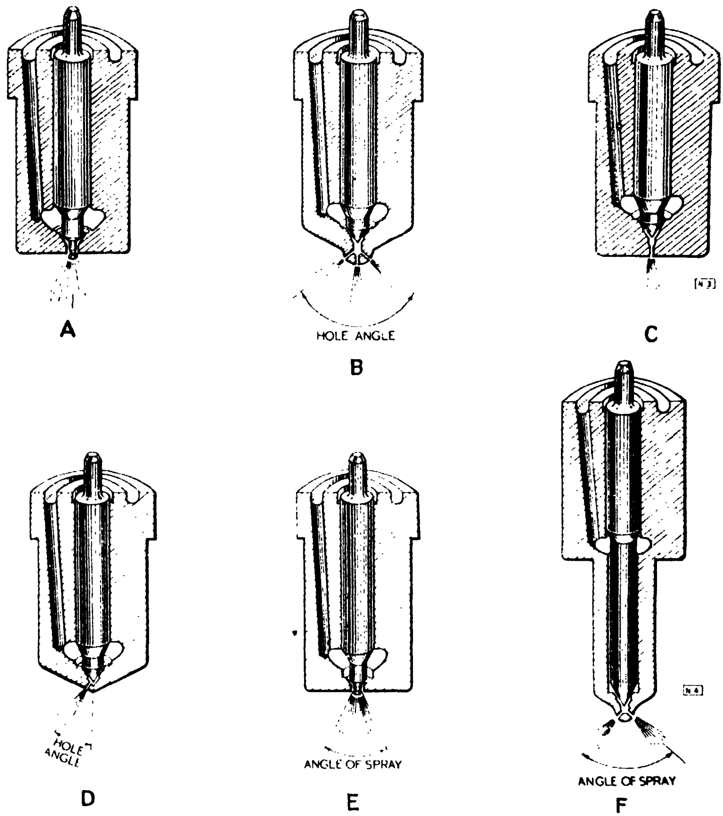


Fig. 12.

(F, fig. 12), and has an extended body at the tip of which is provided the valve seating and dome for the spray holes. The valve stem is a clearance fit in the body, but is located on the lapped portion remote from the seat.

Engines of the ' air-cell ' or pre-combustion chamber type frequently use a pintle nozzle (E, fig. 12). The needle valve is extended to form a pin or pintle which protrudes through the mouth of the nozzle body. The angle of the cone of spray can be varied by varying the size and shape of the pintle. A further variety of nozzle is the ' delay ' type, in which the pintle is modified so that the rate of injection increases towards the end of the delivery. An example is shown at A, fig. 12 (the details of these nozzles are necessarily drawn somewhat out of scale to enable them to be seen clearly).

Many types of nozzle holders are in use, some designed for bolting to the cylinder head, some for screwing in; they carry the valve spring against which the nozzle valve opens, and means of adjusting this to provide varying injection pressures. A typical example will suffice to illustrate the general design features.

Fig. 13 shows a C.A.V. nozzle holder and nozzle in position in a cylinder head. The principal parts are indicated. The slight leakage of fuel which accumulates within the nozzle holder and which serves to lubricate the nozzle, can be led away by a pipe connected to the leak-off nipple provided.

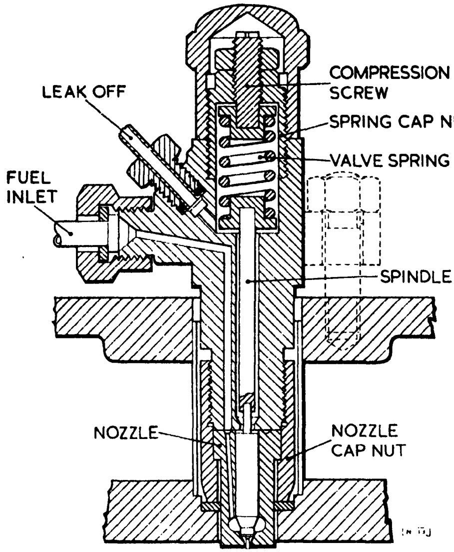


FIG. 13.

Engine Cooling.

In water-cooled engines the heat is carried away from the hot surfaces of the cylinder, cylinder head, valve pockets, etc., by a moving stream of water which gets rid of the heat to the atmosphere by means of a radiator. In air-cooled engines the heat is carried outwards from the cylinder by means of metal fins over which currents of air are passed. In some engines where so-called evaporative cooling is used, the cooling medium is a nearly stationary body of water which is allowed to boil and pass off steam to a surface condenser. The condensed water is returned to the cylinder jacket by a pump, but the circulation is very slow as the water vapour passing to the condenser carries off 20 to 100 times the heat as compared with the same weight of water in a normal system. The condenser is made in the form of an ordinary radiator. With this method of evaporative cooling an invariable engine temperature is maintained—viz. 212° F.—for sea-level conditions. With air cooling higher temperatures will be obtained, and with water somewhat lower.

In a well designed petrol engine with pump circulation the water outlet temperature is about 140° F. In a Diesel engine this may be raised to 160° F. With thermo-syphon cooling higher temperatures up to 205° F. are common. The actual wall temperatures of the cylinder are considerably higher. Herreshoff (*Proc. S.A.E.*) gives the following figures:—

Type of Engine.	Heat to Cooling Medium.
Aviation	30 B.Th.U. per b.h.p. per min.
Valve in head	50 ,, ,, ,,
Side valve	70 ,, ,, ,,

The variations are due to different types of combustion chambers and sizes of valves; the figures are on the high side.

In a Diesel engine the figure is about 33 B.Th.U. per b.h.p. per min. As a safe figure 60 to 70 B.Th.U. for a petrol engine and 40 to 50 B.Th.U. for a Diesel may be taken, depending on the load factor which varies from 30 per cent. in a private car to 100 per cent. for a heavy lorry.

The true mean water temperature is not the arithmetic mean of the inlet and outlet temperatures but something lower. Thus if these are respectively 100° and 200° the true mean temperature is about 132° F.

For a honeycomb radiator the performance was found by Miall to be in accordance with the following equation :—

$$B = 0\cdot 007\ AV^{0\cdot 856}\,(1 - e^{1\cdot 77t});$$

where

B = B.Th.U. per minute per degree temperature difference between air and true mean of water temperatures.

A = frontal area in sq. ft.

V = velocity of air in ft. per min.

***t* = effective thickness of core in ft. (length of tubes between bulges at ends).**

***e* = base of Napierian logarithms.**

For a tubular radiator the following equation applies :—

$$H = (0\cdot 00012 - 0\cdot 0000089\ R)\frac{V^{0\cdot 85}}{d^{0\cdot 32}}$$

where

H = B.Th.U. dissipated per sq. ft., per second, per degree F. difference.

V = Air speed ft. per second.

R = Ratio $\frac{\text{Air surface}}{\text{Water surface}}$

***d* = Bore of tube in ft.**

It is found that with pump circulation the tube surface is always more efficient owing to greater turbulence in the water stream, and a radiator for thermo-syphon cooling requires to be nearly twice the size of one for pump circulation.

The most suitable pump is a centrifugal one, and the speed of circulation over the tube surface should not be less than 1 ft. per sec. In round figures a delivery of about 0·33 gal. per b.h.p. of engine per minute is required.

The fan for air circulation through the radiator is of the first importance, but little attention as a rule is paid to it. As a result, although a large volume of air is handled by the fan, it is at the expenditure of considerable power which falls to be deducted from the effective power of the engine.

Reliable data on this subject is largely wanting, but for maximum efficiency, blades of aerofoil section should be used, the fan should be fully cowled and the radial blade tip clearance reduced to a reasonable minimum. The fan blade tip speed should also be maintained at a minimum in order to reduce the noise level.

Supercharging.

The object of supercharging is to enable the engine to draw in a greater quantity of fuel mixture during the suction stroke, thus maintaining the volumetric efficiency at high speeds and increasing the power output. For very high output performance in petrol engines the supercharging is carried to a point where considerably over 100 per cent. volumetric efficiency is obtained. It may be remarked here that the supercharger is nearly always arranged to draw through the carburettor, thus handling a petrol-air mixture. The result is perfect admixture of the fuel and air, and cooling of the supercharger by the evaporation of the petrol.

Three types of blower are now employed :—

(*a*) The Roots Blower.

(*b*) The Eccentric Vane Type.

(*c*) The Centrifugal Fan.

(*a*) is difficult to make and above 10 to 12 lbs. discharge pressure is extravagant in power; (*b*) is used for engines of racing cars, etc., and (*c*) mainly for aviation work as it is by far the lightest and its noise is immaterial. Its speed is usually from 20,000 to 25,000 r.p.m.

Fig. 14 shows two examples of eccentric vane blowers of a type suitable for 20 to 30 lbs. per sq. in. discharge pressure. They are usually coupled direct to the crankshaft at the front end for engine speeds up to 7,000 r.p.m.

Dealing first with supercharging petrol engines. The effect is to increase the pressure and to some extent the temperature of air in the cylinder after compression.

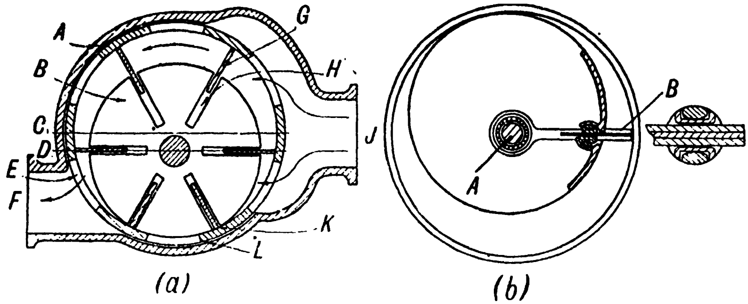


FIG. 14.—Eccentric Vane Blowers.

A. Casing. B. Rotor. E. Rolling drum. F. Outlet. G. Blade. J. Inlet.

A. Layshaft central to casing, eccentric to rotor. B. Blade.

It follows that if detonation is to be avoided, the ratio of compression must be reduced. Using 83 octane value Ricardo found the following relationship between supercharge pressure and h.u.c. (columns 1 and 2).

Supercharge in Hg.	h.u.c.	b.m.e.p.
0 in.	6·55	145
2 ins.	6·30	158
4 ins.	6·10	170
6 ins.	5·90	182
8 ins.	5·70	195

In obtaining the third column readings the octane number of the fuel was raised from 78 to 88 to compensate for increased tendency to detonate running at *constant* compression and not at figures given in column 2.

In a petrol engine it is thus shown that the limit to supercharging is detonation (assuming the engine is designed to withstand the high pressures reached).

In a Diesel engine this limitation is absent as there is no fuel in the cylinder during compression, there is also more rapid ignition when it starts.

O. Thornycroft considers that in a Diesel engine working at 6 ins. Hg. supercharge, maximum b.m.e.p. is increased by 13·5 per cent., with fuel consumption unchanged; combustion knock reduced, and total weight of engine increased by 5 per cent.

Thus the supercharged power/weight ratio would be greater.

Note.—In supercharged petrol engines used for racing an output of 187 b.h.p. per litre has been recorded.

Size of Supercharger.—For a Roots or Vane type the following formulæ apply :—

$$P = \frac{B \times D_e \times S_e \times 14 \cdot 7}{V/2}$$

$$B = \frac{P \times V/2}{D_e \times S_e \times 14 \cdot 7}$$

where

B = blower capacity litres per revolution.
V = engine capacity litres per revolution.
D_e = discharge coefficient.
S_e = suction coefficient.
P = absolute discharge pressure lbs. per sq. in.

D_e is a figure experimentally determined which varies from 0·7 at 2,000 r.p.m. to 0·9 at 7,000 r.p.m

$$S_e = \left(\frac{P_s}{14\cdot7}\right)^{0\cdot715} = \sqrt[1\cdot4]{\frac{P_s}{14\cdot7}}$$

where P_s is absolute suction pressure.

Carburettors and Manifolds.

The function of the carburettor is to provide an approximately correct mixture of two very dis-similar substances, an oily liquid and air, in an average ratio of about 1 to 17·5 by weight respectively, at any engine speed. The function of the manifold is to distribute this very unstable mixture to each cylinder in equal quantities. Except when a supercharger is interposed between the carburettor and the manifold this condition seldom obtains. The effect of the supercharger by its beating action, is to heat and thoroughly mix the two ingredients, which are then conducted under pressure to each cylinder. With defective distribution wide variation will be found between the power developed in each cylinder. In a case recorded by O. O. B. Beale the indicated power varied from 11·8 to 21·2 between cylinders, and in another case, by gas analysis the following figures were recorded.

Fuel Distribution to Cylinders.

TABLE VI

Tests made on a standard four-cylinder 22·5 h.p. car showed the ollowing values, the latter being determined from gas analysis.

Cylinder No.		CO_2 in Exhaust. Per Cent.	Air Fuel Ratio.	Distribution Per Cent.	Excess or Deficiency. Per Cent.
A.	1	8·1	10·9	28·5	+14·0
	2	8·4	11·1	27·4	+ 9·6
	3	12·0	13·5	21·8	−12·8
	4	11·6	13·3	22·3	−10·8
B.	1	9·0	11·5	27·2	+ 8·8
	2	9·5	11·9	26·6	+ 6·4
	3	11·4	13·1	23·2	− 7·2
	4	11·9	13·4	23·0	− 8·0

A = tested at 15 m.p.h. up a 4 per cent. gradient. B = tested at 30 m.p.h. up a 4 per cent. gradient.

The tests show that the two front cylinders (Nos. 1 and 2) are subject to 'loading-up,' particularly at the lower speed.

With a 'wet' mixture there is difficulty in keeping droplets of fuel in suspension which depends on (1) the speed through the pipe, (2) the fineness of the petrol spray, (3) the pressure, and (4) the mixture temperature.

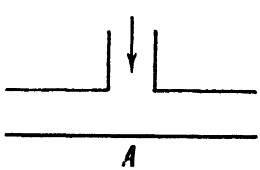


FIG. 15.

Where the direction has to change, which must happen at least once for each cylinder, a round bend is undesirable, since speed round the inside of the bend may amount to no more than say one-half the speed round the outside. Deposition of liquid is bound to occur; angular branches may prevent this.

With a downdraught carburettor there is a short pipe leading to the horizontal distributing pipe. In a four-cylinder engine the mixture has to oscillate to right or left once for every revolution. The result is that some deposition of petrol takes place at A (fig.15) and if allowed to accumulate will build up and ultimately be sucked into one or other of the arms with resulting disturbance

of running. The remedy is to apply local heat to the point A, which is achieved by forming a hot-spot, either by bolting the inlet and exhaust manifolds together at this point or casting them integrally. The hot-spot may be thermostatically controlled. The same conditions will apply to six- and eight-cylinder engines although the problems of manifold design assume such complexity that they cannot be dealt with here.

The mixture ratio to give the best results is by no means constant over the whole range of engine speed; at idling speed the mixture must be rich, then normal throughout most of the range, increasing in strength slightly at the highest speed where maximum power is needed.

Carburettors.

Fig. 16 illustrates the Solex downdraught carburettor Models 26 AIC and FAI which represent the simplest form of modern 'static' carburettor and in which compensation is effected by means of an air correction jet in conjunction with an emulsion tube.

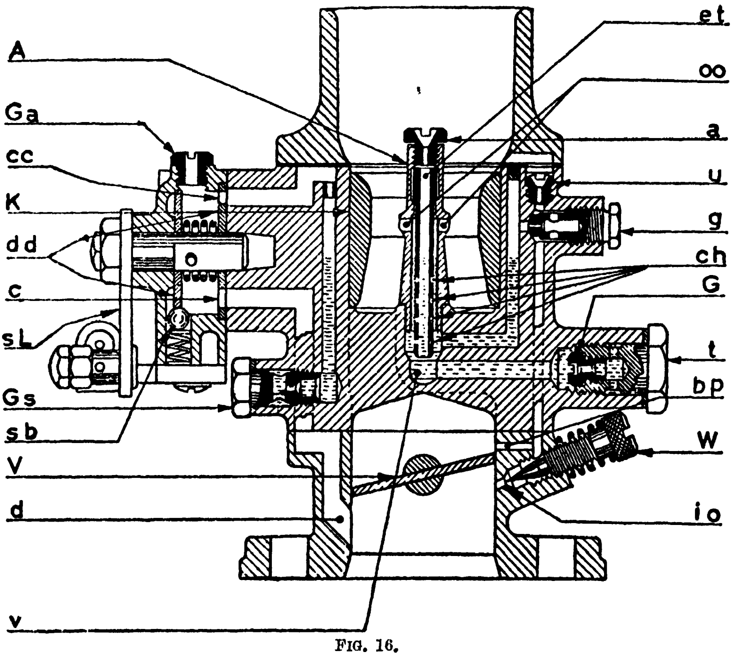


FIG. 16.

These carburettors embody the Solex 'Bi-starter' unit which is a small auxiliary carburettor integral with the main carburettor and ensures easy starting from cold.

The 'Bi-starter' consists of a disc valve controlled mixing chamber fed via the petrol jet (Gs) and the air jet (Ga), it is operated by the lever (sL) which rotates the discs until the drillings in the right hand disc register with the entry duct (cc) and exit duct (c), through which the petrol and eventual mixture flow respectively. The mixture finally passing through the delivery duct (d) below the throttle. When the lever (sL) is operated throughout its full travel, a rich mixture is provided for starting under very cold conditions. With the lever (sL) at the intermediate position the mixture provided is less rich, due to a very much smaller drilling in the inner disc coming into operation, its effective position being located by the spring loaded ball; thus a weaker mixture is produced on which the car can be driven away from cold without fear of over-dosing. When the engine is fully warmed up the lever (sL) is returned to the 'off position' by pushing the dashboard knob fully home. The starter is then isolated from the main carburettor, allowing the latter to function normally.

The main jet (G) is secured in the carrier (t) and meters the fuel drawn from the float chamber; the metered fuel then passes into the spraying well (A) via the reserve well (v) where it meets air drawn downwards through the calibrated air correction jet (a). This passes out through the emulsion holes (ch) into the annulus, where an emulsion is formed with the petrol, and the resultant mixture rises to the four spraying orifices, of which two are shown (oo) in the waist of the choke tube (K). Here the emulsion is absorbed by the main air current and passes down to the induction pipe of the engine via the Butterfly throttle (V).

It will be seen that as the air flow is increased by opening the throttle, the level of the petrol within the emulsion tube (et) drops until the highest of the emulsion holes (ch) are uncovered; as the level continues to fall, further emulsion holes are uncovered, thus decreasing the suction in the metering orifice. These emulsion holes may, therefore, be regarded as compensators, and by suitable determination of the size of the air correction jet (a), any desired degree of weakening or enriching the mixture can be obtained. The pilot jet draws its petrol from the main jet and thus ceases to operate when the main jet is in normal operation.

The Solex downdraught carburettor Model 32PBI–2 is the most elaborate of this type of carburettor and incorporates a two-phase acceleration and economy device. The main carburettor and ' bi-starter ' are as described above.

The acceleration device consists of a box-like structure situated at the side of the float chamber and fitted with a jet (Gu) on the exterior—see extreme right of fig. 17. The internal components comprise the following :—

1. A pump shaft ultimately actuating a ball valve (H), in front of which is a light spring. The shaft passes through and is affixed to the centre of a flexible membrane (Mm).

(2) To the right of the membrane (Mm) is a chamber containing a compression spring (r). This chamber is subject to induction depression via the duct having its exit at a point in the throttle chamber on the engine side of the throttle (V).

(3) Further still to the right is another chamber, one wall of which consists of the flexible membrane (M), spring loaded, and centrally from which protrudes a short rod in clearance contact with fulcrumed lever (L). The lever (L) slides on to a spring loaded horizontal rod, the effective length of which is determined by the adjustment of the nut (e) or by a split pin. The rod is fixed at its end to a small lever fitted to one end of the carburettor throttle spindle.

The chamber through which the pump shaft passes to the left of membrane (Mm) is in communication with the chamber containing the compression spring to the left of the membrane (M). Both chambers are filled with petrol, via the ball valve at the base of the assembly, prior to the functioning of the device, and the contents of both are equally affected thereby.

Function and Economy Action.—When the butterfly (V) is nearly closed and the vehicle travelling at very low speed, the central chamber containing the spring (r) is subject to high induction depression. The membrane (Mm) to which the pump shaft is affixed, subject to the same depression, flexes to the right, compressing the spring (r) and the ball valve (H) is on its seating, being held firmly thereon by pressure of the light spring to the left of it.

At this stage the lever (L) is lightly in contact with the end of the rod affixed to the centre of the membrane (M).

On opening the throttle to accelerate, the immediate sequence of events is as follows :—

The lever (L) forces the rod, and consequently the membrane (M) to the left, compressing the spring on the inner side and discharging the petrol contained in its chamber, through to the chamber on the left of the membrane (Mm), so displacing the petrol contained therein. This displacement forces the ball valve (H) off its seating, and the petrol is discharged via the ' pump ' or ' Speed jet ' (Gp), from the Injector tube (i) into the central air stream passing down the choke tube (k).

Next, the induction depression rapidly falls in the central chamber as the throttle opens, and the spring (r) expands, flexing the membrane (Mm) to the left thereby discharging the residue of petrol contained in the chamber to the left of it, to emerge via the injector tube (i) as just described.

Now let us consider the function of the jet (Gu) often referred to as the ' economy jet ' since such is its effect in the final stages of carburation.

Consider first of all what takes place at high speeds with the throttle fully, or almost fully open.

The depression in the induction manifold is low and consequently negligible in the central chamber containing the spring (r). Thus the ball valve (H) is pushed off its seating and petrol is free to flow from the float chamber through the chamber on the left of the membrane (Mm) and so past the ball valve (H) to the injector tube (i).

The calibration of the speed or pump jet (Gp) is determined to ensure the best acceleration, and the main jet of such a size that its output supplemented by that of the speed jet is equal to the requirements of the engine at major throttle openings.

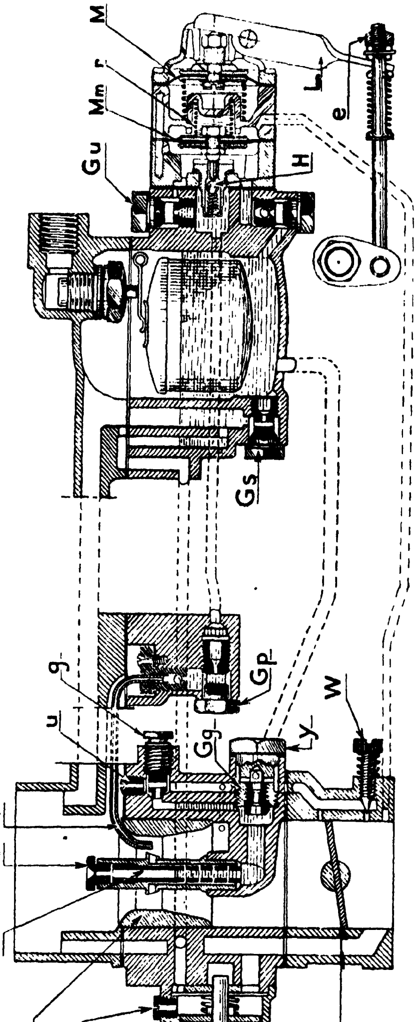


FIG. 17.

At lower throttle openings, when the depression in the central chamber is high and the ball valve (H) in consequence seated, the main jet may be too small for optimum carburation efficiency, and supplementary petrol is therefore needed to maintain the correct standard of output.

Thus the economy jet (Gu) is installed, and it will be obvious that with the pump valve closed, petrol is free to flow from the carburettor float chamber through the valve chamber and thence via the economy jet (Gu) and the speed jet (Gp) to emerge at the injection tube (i).

On the size of (Gu) therefore depends the fuel flow needed by the engine for good performance and maximum economy in the circumstances described viz., at ' cruising ' speeds.

The Zenith ' V ' type vertical model is illustrated in fig. 18, in which compensation is effected by means of an additional jet. The main and compensating jets feed into the choke through the emulsion block, and it will be seen that as the petrol in the well of the capacity tube is consumed,

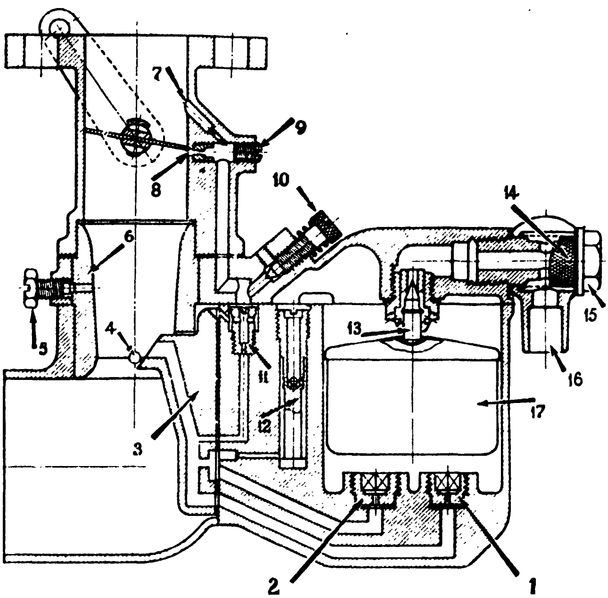


Fig. 18.

1. Main jet.
2. Compensating jet.
3. Emulsion block.
4. Distributor bar.
5. Choke Retaining Screw.
6. Choke Tube.
7. Slow running outlet.
8. Progression jet.
9. Progression jet cover or plug.
10. Slow running air screw.
11. Slow running jet.
12. Capacity tube.
13. Needle.
14. Filter.
15. Filter union.
16. Petrol union.
17. Float.

and as the top of the well is open to the atmosphere, petrol issuing from the compensating jet will be air bled from atmosphere. As depression increases the compensating jet supplies a weaker mixture, whilst the main jet delivers more petrol. The fuel issuing through the main jet will meet the emulsified petrol from the compensating jet in the common channel. This will tend to break up the petrol from the main jet also, so that when the supply from both sources is eventually drawn from the emulsion block nozzle into the choke tube complete atomisation is assured. On certain models the jets are removed from underneath the float-chamber after first removing cover plugs.

The operation of the Zenith horizontal and downdraught 'V' type carburettors is similar to that described above. The slow running jet draws its supply of petrol from the main and compensating jets, and therefore ceases to function when the latter are in normal operation. Two types of automatic starting device are available for the Zenith 'V' type carburettors, the air valve type illustrated in fig. 19, and the dip tube type in which the mixture strength is automatically controlled by means of an air bleed in the dip tube.

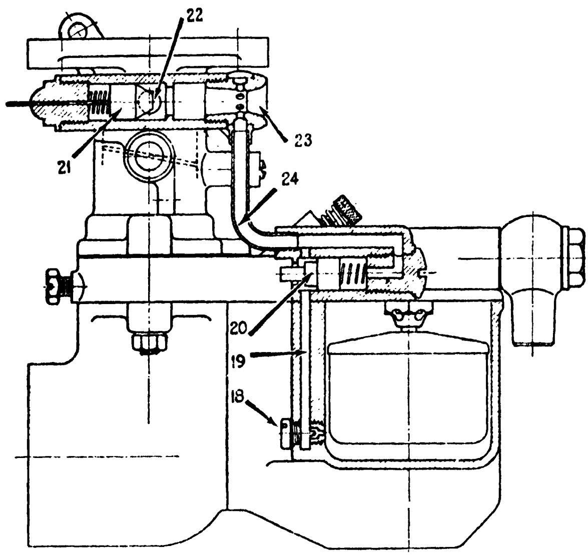


FIG. 19.

18. Starting jet.
19. Communication passage.
20. Air valve.
21. Manually operated main valve.
22. Inlet orifice—engine side of throttle.
23. Venturi of throttle.
24. Communication tube.

Fig. 20 illustrates the Stromberg downdraught carburettor type DBV-36, 42 which is compensated by means of radial holes in the main discharge jet (14), air being drawn through the high speed bleed (12). The acceleration pump is linked to the throttle lever and, the pump suction valve (18) being of the non-return type, when the throttle is opened suddenly the petrol contained under the pump piston (17) is discharged through the pump jet (24). The economy device is operated by the by-pass vacuum piston (5) which, when the depression in the suction pipe (7) is high, *i.e.* at small throttle openings, is inoperative. When the depression falls below a certain level, however, the piston spring (6) is no longer overcome and forces down the piston which in turn depresses the stem of the by-pass valve beneath it. This opens the valve and allows an extra supply of petrol to flow through the by-pass jet (10) to the base of the main discharge jet (14).

The S.U. controllable jet carburettor operates on the constant vacuum principle. In this the flow of air is controlled by a rising and falling piston to which is attached a taper needle which moves in and varies the area of the jet orifice. When the engine is idling, the piston is in its lowest position shutting off most of the air passage; at the same time the largest diameter of the jet needle is within the jet, almost shutting off the petrol flow.

The piston is connected to a suction disc operating in a cylinder which is in communication with the throat of the carburettor. When the throttle opens, the upper part of the suction disc chamber is subjected to increasing vacuum, and the suction disc is lifted together with the piston, allowing more air to pass through the choke way, whilst the taper needle recedes from the jet orifice and allows more petrol to pass. The operating factor is the partial vacuum in the throat of the carburettor, so that by pneumatic control alone the size of the airway and the delivery of fuel are made to adjust themselves to the rising engine speed.

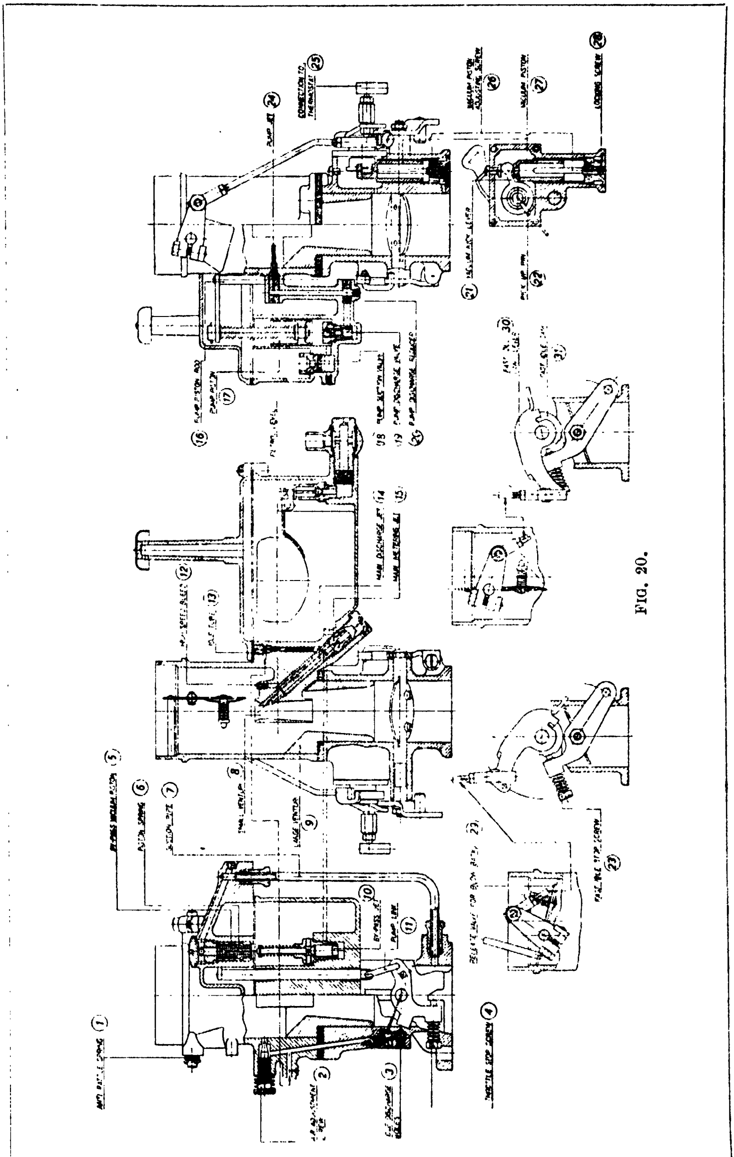


FIG. 20.

The air speed through the choke is actually constant throughout the whole range of engine speed, while the fuel is accurately metered by means of the taper jet needle. By raising or lowering the needle relatively to the piston, the mixture can be varied through wide limits, and any desired petrol-air ratio obtained.

The jet itself, by means of a bell-crank lever operated by hand from the dash, is capable of being raised or lowered relatively to the needle valve; when in its lowest position the area available for petrol is greatest, thus providing a rich mixture for starting, and the air orifice is not interfered with.

An auxiliary starting carburettor of the thermostatically controlled type is available for all S.U. carburettors.

Petrol Injection Systems.—A considerable amount of work on the development of petrol injection equipment has been undertaken of recent years, both on manifold and direct cylinder injection systems. The method of injection is similar to that employed on compression-ignition engines. Petrol injection solves many distribution problems and results in an increase in volumetric efficiency, the equipment is, however, costly, and cannot, as in the compression-ignition engine be offset by the elimination of the ignition system. It is still very much in the development stage.

General.—In view of the considerable increase in export trade the following table is of interest:—

TABLE VII.

HORSE-POWER AT VARIOUS ALTITUDES.

Altitude.	Percentage Horse-Power.
Sea-level	100·0
1,000 ft.	96·5
2,000 ,, ...	93·0
3,000 ,, ...	89·5
4,000 ,, ...	86·5
5,000 ,, ...	83·0
6,000 ,, ...	80·0
7,000 ,, ...	77·5
8,000 ,, ...	74·5
9,000 ,, ...	71·5
10,000 ,, ...	69·0

Clutches and Flywheels.

The clutch is a mechanical device for connecting and disconnecting the engine from the transmission gear at the will of the driver. It is invariably operated by a pedal at the extreme left of the driver; depressing the pedal disengages the clutch which automatically re-engages when it is released. Apart from the driver's control, there are two forms of clutch in which the re-engagement of the clutch is only completed when a certain engine speed is attained; these are (*a*) the so-called 'fluid flywheel,' which is really a turbo-pump, the kinetic connection being furnished by a fluid, set in motion by one set of blades, delivering its energy against another set oppositely inclined; (*b*) the centrifugal clutch which can be released by the pedal, but which is only completely closed at a certain speed by centrifugal force. These two devices make it impossible for the clutch to be engaged with a jerk. In the ordinary friction clutch there are two members, one attached to the engine shaft, the other to the first gear-box shaft; each is provided with suitable friction surfaces normally held in contact by springs but separable by means of a pedal when the clutch is to be disengaged.

The heaviest parts of the clutch rotate with the engine and act as (and largely in substitution for), a flywheel while the disengageable portion is made of very thin sheet steel as light as possible. The compressed asbestos fabric friction surface is riveted to the sheet steel so that the steel itself is not subject to friction with resulting heating and buckling.

Largely on account of their weight, cone clutches have become obsolete, and, for the same reason, the multidisc type is only used for very high powers.

The requirements of a friction clutch are:—

(1) High coefficient of friction under all conditions, which must not be affected by oil, water, or reasonably high temperatures.

(2) Gear-box member must be very light.

(3) Spring pressure must not be excessive, and adequate leverage for operating is necessary.

(4) Slipping must never take place even at the highest engine torques.

(5) Friction surface must have a long life, and be capable of withstanding a certain amount of intentional slipping without damage.

(6) The engagement must be smooth.

(7) **Heat must be capable of rapid dissipation.**
(8) **Lubrication of all working parts must be adequate and very simple to maintain.**

All parts of a clutch must be accurately in rotational balance.

When fitted with compressed asbestos fabric a clutch can absorb energy at the rate of 24,000 to 30,000 ft.-lbs. per sq. in. of facing per minute so long as the temperature is below 500° F.

Clutch calculations are very simple and the problem is to make the clutch torque equal to the engine torque plus a safety factor of 25 per cent.

Using the following symbols :—

H.P. = horse-power transmitted.
T_i = engine torque in lb.-ins.
n = r.p.m.
S = *total* spring pressure in lbs.
N = number of friction surfaces in contact.
R = effective radius of friction disc.
R_i = inner radius of friction disc.
R_o = outer radius of friction disc.
μ = coefficient of friction.

Then $T_i = \frac{63{,}024 \text{ H.P.}}{n}$ **and allowing for the necessary factor of safety the clutch torque will be** $1 \cdot 25\ T_i$

$$R = \frac{2\,(R_i^2 + R_i R_o + R_o^2)}{3\,(R_i + R_o)}$$

then $\quad 1 \cdot 25 T_i = s \times \mu \times N \times R.$

whence $\quad s = 1 \cdot 25\ T_i/\mu NR.$

Note.—**S is generally distributed over 3, 4, 6, or 8 separate helical springs, a single central spring being difficult to arrange, and heavy. N may of course be 1 or as many more as required.**

The dimensions of clutch friction discs is now standardised by the S.M.M. and T., ranging from 6 ins. × 4·5 ins. dia. up to 18 ins. × 13·5 ins. dia. The thickness ranging from ⅛ in. up to ⅜ in.

It is undesirable for the surface speed at the effective radius R to exceed about 7,000 ft. per min. and rivet heads must be sunk well below the rubbing surface.

Flywheels.—**A flywheel is necessary to smooth out variations in torque and to maintain an approximately constant angular velocity. Formerly flywheels of considerable mass were employed, but as engine speeds have increased they have become less necessary, and the present demand for rapid acceleration is incompatible with their use. All that is necessary is a substantial disc of mild steel to which the driving member of the clutch can be secured. The flywheel effect depends upon the polar moment of inertia. For a plain disc,** $I_o = \frac{\pi}{32} D^4$ **and for a hollow cylinder** $I_o = \frac{\pi}{32}(D^4 - d^4)$.

Again, for a plain disc the radius of gyration $k = (D/2) \times \sqrt{0 \cdot 5}$

and for a cylinder $\quad k = \sqrt{(D^2 + d^2)/8}.$

For high speed engines the rim tension is important and is approximately equal to $v^2/10$ **where** v = **velocity in ft. per sec.**

Ignition Equipment.

The ignition equipment on the modern car is invariably of the battery-coil type, whilst magnetos are more generally used on industrial, marine, tractor and motor-cycle engines.

The function of the ignition equipment is to provide a succession of accurately timed, high voltage sparks to ignite the explosive mixture in the engine cylinders. Since in a six-cylinder engine, three sparks are produced for each revolution of the engine, the ignition equipment is called upon to supply anything up to about 12,000 or more sparks per minute, each of which must be carefully timed to less than a thousandth part of a second.

Coil Ignition.

Coil ignition equipment, a typical diagram of which is shown in fig. 21, consists essentially of :—

(*a*) the battery which supplies the electrical energy ;
(*b*) the ignition coil, which transforms the battery voltage into a high voltage of the order of at least 6,000 volts to produce a spark at the plug gaps ;
(*c*) the contact-breaker, which opens and closes the primary circuit through the coil, and to which a condenser is connected in parallel ;
(*d*) the distributor which delivers the spark to the correct cylinder ; and
(*e*) the ignition switch for disconnecting the battery when it is desired to stop the engine.

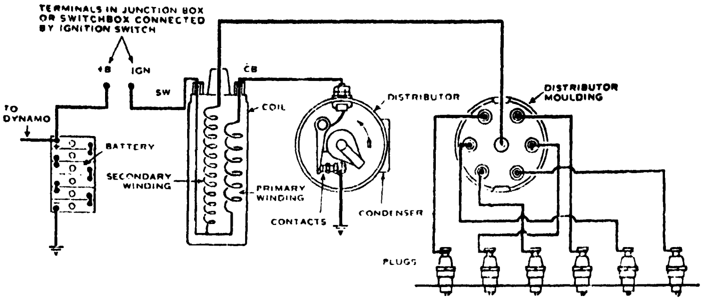


FIG. 21.

The ignition coil has a laminated iron core, around which are two windings ; a secondary winding of many thousands of turns of very fine wire, and on top of this, a primary winding of comparatively few turns of thick wire. The core and windings are housed in a cylindrical metal container filled with an insulating compound which effectively seals the coil winding and prevents the ingress of moisture.

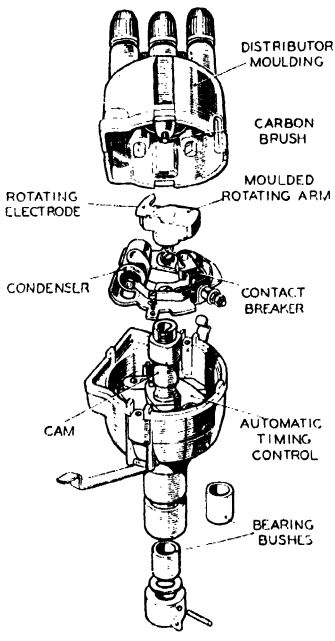


FIG. 22.

The contact-breaker and distributor form a combined unit which is gear-driven by the engine (fig. 22). The former consists of two contacts, one fixed and the other mounted on a pivoted lever provided with a spring which tends to bring the contacts together. The movement of this contact is controlled by a bakelised fabric heel which presses on the surface of a cam, having, either four

or six lobes according to the number of engine cylinders. When the distributor shaft is rotated, the contacts are made to separate at the appropriate instant by the cam lobes lifting the heel on the contact-breaker lever (fig. 2.).

The distributor comprises a rotating arm and a moulding containing on the inside a number of metal segments which are connected, via the terminals, to the sparking plugs by means of the high-tension cables.

When the ignition is switched on, current flows from the battery, throguh the primary winding of the coil and then on to the contact-breaker. On pressing the starter switch, the engine rotates, so driving the distributor shaft carrying the cam, and causing the contacts to make and break alternatively. Each time the contacts open, the collapse of the primary current in the coil gives rise to the high secondary voltage, which is passed from the coil to the rotating distributor arm. From here, it jumps the gap to one of the metal inserts in the distributor moulding, which in turn is connected by cable to one of the sparking plugs, with the result that a spark is produced at the plug points.

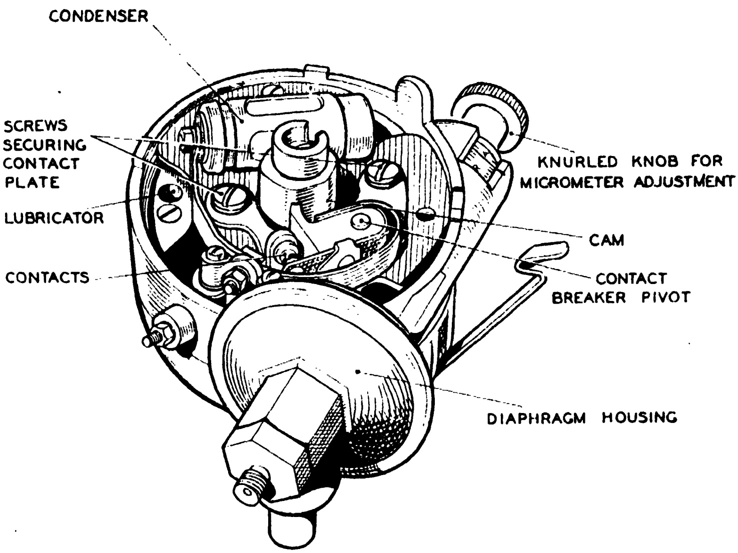


FIG. 23.

Immediately after the spark occurs, the contact-breaker closes and the cycle of operations will be repeated for the spark to occur in the cylinder next in the firing order.

A condenser is connected across the contact-breaker, the purpose of which is to obtain a quick break of the current and to control sparking at the contacts.

The equipment is provided with a centrifugally-operated automatic timing control. This mechanism is housed beneath the contact-breaker and takes the form of a governor which alters the position of the contact-breaker cam with relation to the driving shaft. The mechanism consists of two weights which tend to move outwards against the tension of a spring as the engine speed increases, causing the cam to move in the direction of the drive and thus advancing the timing.

In addition, a vacuum-operated timing control is incorporated on many types of distributor (fig. 23), the purpose of which is to alter the ignition timing according to the engine load. Advantage is taken of the fact that at any fixed engine speed the intake manifold suction is roughly inversely proportional to the engine load ; that is to say, with a light load the suction in the intake is of a comparatively high order, whilst on full load it is of low value. This variation of pressure is utilised to give the most suitable ignition timing at all loads. The control consists of a flexible diaphragm enclosed in a metal housing. The diaphragm is linked so that when acted upon by the vacuum in the induction pipe of the engine, its movement produces rotation of either the complete distributor head or contact-breaker base.

Magneto Ignition.

Two types of magneto are in general use (*a*) the rotating armature and (*b*) the rotating magnet; the operation of both types is based on the same general principle, namely, the rapid change of magnetic flux in a laminated iron core carrying a suitable winding.

In (*a*) the magnetic flux is provided by an inverted U-shaped permanent magnet and the reversal of the flux, relative to the core, is caused by the rotation of the armature between the laminated pole pieces of the magnet. The armature is provided with two windings, a primary winding consisting of relatively few turns of thick wire and a secondary winding consisting of a large number of turns of thin wire. In (*b*) the flux is provided by a permanent magnet, which is fitted with two laminated pole pieces and forms the rotor. This revolves between another pair of laminated pole pieces which are bridged by a laminated core on which are the primary and secondary windings. In both types the contact-breaker is arranged to open circuit the primary winding at the instant a spark is required.

In operation the rotation of the rotor produces an alternating magnetic field in the laminated iron core of the coil. This, in turn, generates an alternating low tension current in the primary winding. At about the instant when this current reaches its maximum value, one of the lobes of the cam is timed to strike the contact-breaker lever, so separating the contacts and causing a high voltage to be induced in the secondary winding. This high voltage is conducted to each sparking plug in the order of firing by means of the rotating distributor arm, and produces a spark at the plug points.

Magnetos are often fitted with an impulse starter in order to improve the low speed performance and retard the ignition for hand starting purposes. This unit consists essentially of two couplings which are flexibly connected by means of a coil spring. One coupling is fitted on the driving shaft, while the other is secured to the magneto spindle. As the engine is slowly rotated by hand, a pawl prevents the movement of the magneto rotor, whilst the driving member continues to rotate, thereby stressing the coupling spring until such time as the pawl is tripped by means of a cam in the impulse starter. When this occurs, the magneto rotor is accelerated rapidly through the sparking position and a powerful spark is produced, the timing of which is retarded for ease of starting. The sequence is repeated until the engine fires and continues to run, the pawls being held out of engagement at a comparatively low speed by centrifugal force.

A centrifugally-operated automatic timing device may also be fitted for certain applications.

The Dynamo.

The dynamo in general use on cars and commercial vehicles is cylindrical in form, consisting of a welded steel shell or yoke and with usually a two-pole magnet system and a conventional armature and commutator.

In most cases, the dynamo is of what is known as the swung mounted type and is driven by a belt which also forms the drive to the engine cooling fan. Many dynamos are of the ventilated pattern in which cooling air is drawn through the machine by a centrifugal fan integral with the driving pulley. For special conditions, however, such as on tractors where excessive dust is encountered, it is often the practice to employ totally enclosed dynamos.

The output from the dynamo varies in accordance with the strength of the magnetic field and with the speed at which the dynamo is being driven and as the machine is called upon to operate over a very wide speed range, a means of controlling the dynamo output must be provided.

Two methods generally employed have been the 'third-brush control' which has now been superseded entirely by 'compensated voltage control' of the dynamo output.

Third-brush Control.—The third-brush dynamo depends for its regulation on the armature reaction produced by the generated current. The fact that the field winding is connected between one of the main brushes and the auxiliary or third brush in such a manner that distortion of the main field as a result of armature reaction, causes a reduction of the voltage impressed on the field system. The chief merit of this method of control is its simplicity, but it has a serious disadvantage in that the current which the dynamo gives is a function of the voltage of the system. This means that as the voltage of the system becomes higher, the current which the dynamo will give before the regulating effect of the armature reaction comes into play, also becomes higher. Such a machine will, therefore, deliver a heavier current to a fully charged battery during a daylight run with the lamps switched off than it delivers to a partially discharged battery with the lamps in use. This would, of course, result in overcharging the battery during long daylight running periods, with a tendency to discharge the battery if the vehicle is used for long night runs without corresponding periods of daylight running.

To minimise this effect, the two or three charge rate system was adopted, in which a resistance is automatically inserted in the field system when the lights are switched off and short circuited when the lights are switched on. An added refinement usually provided is to have two alternative daytime positions of the charging switch, one giving maximum reduction in charge for summer running and the other a small reduction in charge for daylight running in winter. The operation of switching on the lights short circuits all the resistance and permits the generator to give its maximum output.

While this system of output control was reasonably successful in meeting the needs of the average motorist, it had limitations; one of the most serious being that the insertion of the resistance to reduce the dynamo output, also has the effect of raising the cutting-in speed of the dynamo

This is very undesirable for vehicles used for short runs only in heavy traffic or on vehicles used under conditions necessitating frequent starting and stopping.

Compensated Voltage Control.—This equipment comprises a specially designed dynamo and a regulator unit which varies the dynamo output according to the load on the battery and its state of charge. The dynamo is similar in appearance to the third-brush machine, but is wound differently and the third brush is omitted. The regulator unit is usually housed on the dash, and is integral with the cut-out, fuse and junction box. A wiring diagram is shown in fig. 24.

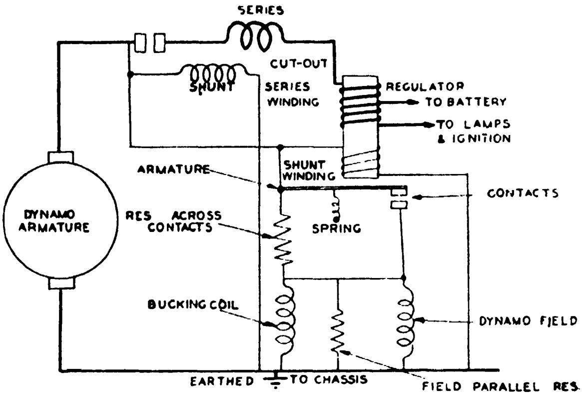


FIG. 24.

The operation of the regulator depends upon the fact that the voltage of a battery varies between certain fixed limits according to the state of charge of the battery, the voltage being, of course, a maximum when the battery is fully charged, and a minimum when the battery is fully discharged.

The regulator is combined structurally with the cut-out. The regulator and cut-out are, however, electrically separate, employing separate armatures, though they possess field systems which are common over a portion of the magnetic path.

The windings consist of a voltage winding connected directly across the dynamo and two current windings, one of which carries the full current from the dynamo to the battery, while the other winding carries the current of the lighting, ignition and accessory loads. These coils assist each other in energising the magnet system and thus in effecting movement of the armature.

When the dynamo voltage reaches a predetermined value, the magnetic field due to the voltage winding becomes sufficiently strong to attract the armature. This causes the contacts to open, thereby inserting the resistance in the field circuit. This reduction in field current lowers the dynamo voltage, and this, in turn, weakens the magnetic field due to the voltage coil. This allows the armature to return to its original position, thus closing the contacts, so that the voltage returns to the predetermined maximum. The cycle is then repeated, and the armature is set into vibration.

As the speed of the dynamo rises above that at which the regulator comes into operation—about 20 m.p.h.—the amplitude of vibration increases and the periods of interruption increase in length, with the result that the mean value of the dynamo output undergoes practically no increase once the operating speed has been attained.

The series windings provide a compensation on this system of control, for if the control were arranged entirely on the basis of dynamo voltage there would be a risk of very seriously overloading the dynamo when the battery was in a low state of charge, particularly if the lamps were simultaneously in use. Under these conditions the dynamo would be forced to give an output to bring the voltage of the system up to the same value as if the battery were in its normal fully charged condition, and this, with a battery of low internal resistance would necessitate an extremely heavy current far beyond the normal capacity of the machine. The series winding assists the voltage coil, so that when the dynamo is delivering a heavy current into a discharged battery the regulator comes into operation at a somewhat reduced voltage, thus limiting the dynamo output accordingly.

By means of the temperature compensation the voltage characteristic of the dynamo is caused to conform more closely to that of the battery under all climatic conditions. In cold weather the voltage required to charge the battery increases, whilst in warm weather the voltage required is lower.

The method of compensation takes the form of a bimetallic spring suspension for the armature of the regulator, which causes the operating voltage of the regulator to be increased in cold weather and reduced in hot weather, and thereby compensate for the variation in charging current which would otherwise occur due to the changed characteristics of the battery.

The Starting Motor.

The electric starting motor is a four pole machine having an extended shaft to carry the engagement gear. Smaller motors are series wound, while larger machines usually have series-parallel field connections. The starting motor is of similar construction to the dynamo except that heavier copper wire is used in the construction of armature and field windings, as it must be remembered that the current consumption of the motor is very high. The average 12-volt starter under normal conditions takes 450–500 amperes at about 7 volts. Series-wound machines have two brushes, and series-parallel machines four brushes, fitted 90° to each other.

The starter drives in general use consist of three main types—the Lucas ' S ' type, the ' SB ' type and the ' rubber ' type. The construction and operation of these drives are described briefly below to assist in differentiation.

' *S* ' *Drive.*—The pinion is mounted on a threaded sleeve which is carried on splines on the armature shaft, the sleeve being arranged so that it can move along the shaft against a compression spring so as to reduce the shock loading at the moment engagement takes place. When the starter switch is operated, the shaft and screwed sleeve rotate and, owing to the inertia of the pinion, the screwed sleeve turns inside the pinion, causing the latter to move along the sleeve into engagement with the flywheel ring. The starter will then turn the engine.

As soon as the engine commences to run under its own power, the flywheel will be driven faster by the engine than by the starter. This will cause the pinion to screw back along the sleeve and so draw out of mesh with the flywheel teeth. In this manner, the drive safeguards the starter against damage due to being driven at high speeds by the engine.

On many drives of this type, a pinion restraining spring is fitted over the starter shaft to prevent the pinion being vibrated into contact with the flywheel when the engine is running.

' *SB* ' *Drive.*—This drive is similar in principle to the ' S ' type, but the pinion rides directly on the starter shaft so that a smaller pinion can be used to provide a larger gear ratio between starter and engine.

' *Rubber* ' *Drive.*—There are several different designs of this type of drive, in which a rubber coupling is used to transmit the drive from the starter shaft to the engine, but the basic principle of operation is the same in each case.

The drive embodies a combination of rubber torsion member and friction clutch in order to control the torque transmitted from the starter to the engine flywheel and to dissipate the energy in the rotating armature of the starter at the moment when the pinion engages with the flywheel. It also embodies an overload release mechanism which functions in the event of extreme stress, such as may occur in the event of a very heavy backfire, or if the starter is inadvertently meshed into a flywheel rotating in the reverse direction due to rocking back off compression.

When the starter is energised, the torque is transmitted by two paths, one via the outer sleeve of the rubber coupling and through the friction washer to the screwed sleeve, while the other path is from the outer to the inner sleeve through the rubber coupling and then directly to the screwed sleeve.

The torque through the rubber limits the total torque which the drive transmits and since the rubber is bonded to the inner sleeve, under overload conditions slipping will occur between the rubber bush and the outer sleeve of the coupling. Slipping does not take place under normal engagement conditions, when the rubber acts merely as a spring with a limiting relative twist on the two members of approximately 30°. Under conditions of unduly severe overload which might cause damage to the drive or its mounting, however, the rubber slips in its housing so that a definite upper limit is set to the torque transmitted and to the stresses which may occur. The bonding of the rubber to the inner sleeve is a recent design alteration, as a result of which the drive is capable of absorbing a torque 50 per cent. in excess of the non-bonded unit.

Splined Shafts.

Tests carried out on actual shafts show that the elastic limit for a splined shaft is lower than for a plain shaft of a diameter equal to the base diameter of the splines. This varies with the form of the spline for the following reason. In fig. 25 a section through two splines is shown.

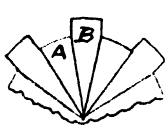


FIG. 25.

When the shaft is twisted, the sectors A and B become helices of which the lengths are different, both being functions of the diameters, respectively A and B, *i.e.* measured round the helix, B is longer than A, the difference being $\pi DB - \pi DA$, and this sets up a shearing stress along the radial lines. Constant stress reversal ultimately fractures the shaft into perfect sectors. It follows that the greater the difference between the respective diameters, the sooner will this result, hence a large number of small splines is considerably stronger than a small number of relatively large ones. In round figures the imposition of splines reduces the shearing strength of the shaft by from 5 to 7½ per cent., depending on their number. If the specific stress can be kept low (sometimes a difficult problem) the effect is minimised, but it is always present in some degree.

Where sliding is not to take place, continuous serrations are often substituted for square splines. When sliding is required the outer component is ground in its inner bore, and the corresponding surface of the shaft at A is also ground so that the top of the spline makes no contact. Splines are also applied to tapered shafts.

Splined shafts of all kinds are almost universally finished by grinding in special machines, and the corresponding parts broached to finished size.

The sections of splined shafts have been standardised by the British Standards Institution, and in the U.S.A. by the S.A.E., but the tables are too extensive for inclusion here.

Standardised broaches for forming these are also obtainable.

***Note.*—If the splines are completely supported between two adjacent components their shearing strength can be added to the shaft as there is no helical deformation. In practice however, this cannot often be arranged.**

Gears.

Spur Gearing.—For many years a pressure angle of 14½° was in use for involute gears and a stub form of tooth was general. The accepted practice for spur-gear teeth now, however, is to use a full depth involute tooth having a pressure angle of 20°.

Reference should be made to the B.S.I. Specification No. 436—1940 for straight spur and helical gear calculations and to Dr. H. E. Merritt's 'Gears.'

The following table gives the comparative pitch and module sizes in general use :—

TABLE VIII.

COMPARATIVE PITCH.

Diametral Pitch.	Circular Pitch.	Module British.	Module Metric.
1	3·1416	1·0000	25·3995
2	1·5708	0·5000	12·6998
3	1·0472	0·3333	8·4665
4	0·7854	0·2500	6·3499
5	0·6283	0·2000	5·0799
6	0·5236	0·1666	4·2333
7	0·4488	0·1428	3·6285
8	0·3927	0·1250	3·1749
9	0·3491	0·1111	2·8222
10	0·3142	0·1000	2·5400
P	*p*	*m*	Mm/m.

$$p = \frac{\pi}{P} \qquad P = \frac{\pi}{p} \qquad m = \frac{p}{\pi} = \frac{D}{T} = \frac{1}{P}$$

Tangential Load.—The tangential load per inch of face width for *wear* may be expressed as

$$L = \frac{X_c Z S_c}{K}$$

and for ***strength*** as —

$$L = \frac{X_b Y S_b}{P}$$

where

K = Pitch factor.
X_b = Speed factor for strength.
X_c = Speed factor for wear.
Z = Zone factor.
P = Diametral pitch.
S_b = Bending stress factor.
S_c = Surface stress factor.
Y = Strength factor.

The allowable tangential load for any pair of gears is the least of the four figures for wheel and pinion.

For automobile gear-boxes in which each individual gear is usually only subjected to occasional use, higher stresses may be allowed and the speed factor, in the above formulæ, may be expressed as follows, for both wear and strength :—

$$X = \frac{15 + G_f}{30} \text{ for forward gears,}$$

$$\text{and } X = \frac{15 + G_f}{25} \text{ for reverse gears.}$$

The value of the gradient factor G_f is obtained by the following formula :—

$$G_f = \frac{100\ MR_t}{W_g\ r}$$

where

M = Maximum engine torque, lb. in.

R_t = Overall reduction in the gear considered.

$= \frac{\text{Engine r.p.m.}}{\text{Roadwheel r.p.m.}}$

W_g = Gross laden weight of the vehicle, and trailer (if any), pounds.

r = Rolling radius of tyre, ins.

NOTE.—The values of the various factors used in the above formulæ may be obtained from B.S.I. Specification No. 436—1940.

Bevel Gearing.—For bevel gear calculations reference should be made to the B.S.I. Specification No. 545—1947 and to Dr. H. E. Merritt's 'Gears.'

Tangential Load.—The tangential load for *wear*, considered to be applied at the pitch radius of the gear may be expressed as :—

$$L\ (\text{pinion}) = \frac{X_{cp}S_{cp}ZF}{K\Omega_c}\left(\frac{C - F}{C}\right)\sqrt[3]{\frac{27,000}{1,000 + U_d}}$$

and

$$L\ (\text{wheel}) = \frac{X_{cw}S_{cw}ZF}{K\Omega_c}\left(\frac{C - F}{C}\right)\sqrt[3]{\frac{27,000}{1,000 + U_d}}.$$

The tangential load for *strength*, considered to be applied at the pitch radius of the gear may be expressed as :—

$$L\ (\text{pinion}) = \frac{X_{bp}S_{bp}Y_pF}{P\Omega_b}\left(\frac{C - F}{C}\right)\sqrt[7]{\frac{26,200}{200 + U_d}}$$

and

$$L\ (\text{wheel}) = \frac{X_{bw}S_{bw}Y_wF}{P\Omega_b}\left(\frac{C - F}{C}\right)\sqrt[7]{\frac{26.200}{200 + U_d}}.$$

where

C = Cone distance.	X_b = Speed factor for strength.
F = Face width.	X_c = Speed factor for wear.
K = Pitch factor.	Y = Strength factor.
P = Diametral pitch.	Z = Zone factor.
S_b = Bending stress factor.	Ω_b (omega) = Bevel gear helix angle factor.
S_c = Surface stress factor.	Ω_c (omega) = Overlap ratio factor.

U_d = Desired total running life of the gears in hours at full load, the usual value of which is 26,000.

For automobile bevel gears higher stresses may be allowed and the speed factor in both wear and strength formulæ expressed as follows :—

$$X = \frac{15 + G_f}{30}.$$

The value of the gradient factor G_f is obtained as described above.

NOTE.—The values of the various factors used in the above formulæ may be obtained from B.S.I. Specification No. 545—1947.

TABLE IX.—BASIC SURFACE AND BENDING STRESS FACTORS.

The following values for S_c and S_b are typical but reference should be made to the tables in the publications previously mentioned.

Material.	Min. Ult. Tensile Strength. (tons/sq. in.)	Min. Brinell Hardness No.	Basic Surface Stress Factor S_c.	Basic Bending Stress Factor S_b.
Cast iron	12	165	1,000	5,800
Malleable cast iron . .	20	140	850	11,000
Phosphor bronze : sand cast	12	69	700	7,000
0·30 per cent. carbon steel	35	140	1,600	19,000
0·40 per cent. carbon steel	40	163	2,000	24,500
0·55 per cent. carbon steel	45	192	2,300	26 500
3½ per cent. nickel steel .	55	241	3,100	33,500
4¼ per cent. nickel chromium steel	100	441	5,500	49,500

***Tooth Pressures.*—The actual tooth pressures, per inch of tooth width, obtained in ordinary practice vary over wide limits; thus, 1,500 lbs. per in. width may be taken as a suitable figure. In the case of racing cars much higher figures are common; thus in the 'Blue Bird' car of 1933 the gear box was designed for a tooth pressure of 2,700 lbs. per in., but the engine power having been increased this rose to 5,400 lbs. steady top gear load. This was with plain spur gears. In the case of the rear axle bevels, the steady load amounted to 6,000 lbs. per in. These pressures were only made possible by a very careful selection of steel, and the very short load period per tooth. The spur gears were ground after hardening and the lubricant was a castor base oil.**

Gear Boxes.

Very numerous efforts have been made, especially in the United States, to devise mechanical arrangements to simplify gear changing. In Great Britain practice has crystallised into two types, the epicyclic with pre-selector mechanism, or a simple type generally modelled on the old clash gear pattern, having all the gears in constant mesh or sliding, but in either case employing synchromesh clutches to effect engagement. In its simple form this consists of a pair of small cone clutches which are engaged before the gears are slid into mesh, or in cases where the gears are in constant mesh (as when helical gears are used), the dog clutches engaging them. The result is that the gears or clutches cannot be brought into contact until they are revolving at appropriate speeds and silent engagement is assured. This mechanism is usually applied to each speed except first.

***Number of Speeds.*—When there are more than four speeds one of them is generally an overdrive, which given sufficient engine power is very advantageous.**

The gear ratios should approximate to a geometric progression from high to low; but latitude is found in this.

Suspension.

Satisfactory and safe suspension has been rendered more difficult with increase of speed and rapid braking. The laminated spring is not so universally used as formerly, other forms having been called for with the tendency to use independent springing at each wheel, where it is extremely hard to find room for the laminated type; in conventional designs it will continue in use from its relative cheapness, simplicity, lateral stiffness, and its high internal friction, which acts as a damper. It has also the advantage that damage takes place slowly—usually one leaf at a time—so there is warning before total failure.

Some successful attempts have been made to use rubber cushion springs, which while functioning well are open to criticism on account of their weight.

There can be no doubt as to the improvement effected in suspension by doing away with the transverse axle common to two wheels; any obstruction met with by any wheel is surmounted without disturbing the equilibrium of the other corners of the chassis. It is, generally thought that the torsion-bar spring represents the direction of most probable development. In earlier models of independent springing, helical springs have been employed which have the advantage of being silent and not needing any lubrication. Both helical and torsion-bar springs, possessing no internal friction, demand more elaborate damping arrangements.

The usual device adopted for individual springing is a two-link parallel motion in which the links are of triangular construction in plan, the fixed ends being splayed apart to resist stresses occurring

longitudinally to the chassis, *i.e.*, as when braking; the construction has been given, from its form, the name of 'wish-bone.' Laminated, helical, or torsion bar springs can be incorporated in the design.

Figs. 26 and 27 illustrate typical independent front wheel suspension systems, the former employing torsion bar and the latter helical springs.

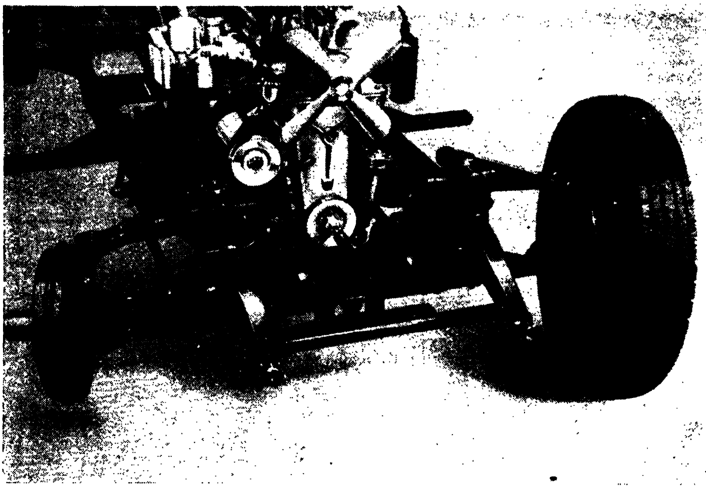


FIG. 26.

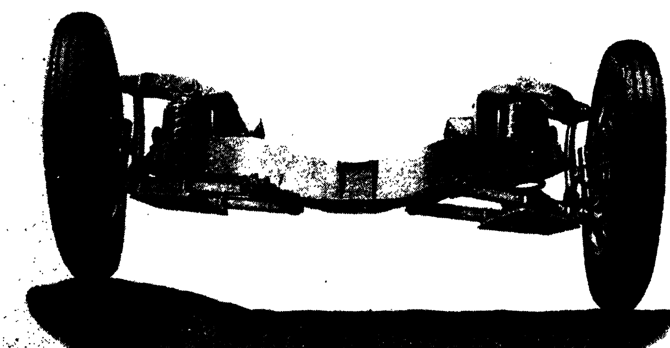


FIG. 27.

***Riding Comfort.*—Is dependent upon (*a*) the periodicity of the spring, and (*b*) the nature and extent of the damping applied to it. Periodicity depends on the amount of static deflection with a given load.**

Let t = seconds, l = deflection in ft.; then $t = \pi\sqrt{\frac{l}{32 \cdot 2}}$ from which Table X is calculated.

TABLE X.

PERIODICITY OF SPRINGS.

Deflection in Ins.	Complete Oscillations per Min.
1·5	153·0
2·0	132·7
2·5	119·5
3·0	108·3
3·5	100·0
4·0	94·1
4·5	88·8
5·0	84·0
5·5	80·0

The figures in this table are modified by the form of damping arrangement employed.

Front springs may have a range of 75–85 oscillations per min. and rear springs from 75–100, depending on the wheel base and track.

***Laminated Springs.*—Three types are in use: (A) the semi-elliptic; (B) the so-called cantilever; (C) the so-called quarter elliptic which is really the true cantilever, see fig 28.**

Let

W = load on spring in lbs.
L = length in ins.
δ = deflection due to W in ins.
R = W/δ, 'rate' in lbs per in. δ
f = stress in material lbs. per sq. in.
E = modulus of elasticity.
Z = modulus of section.
n = number of leaves.
t = thickness of leaves in ins.
b = breadth of leaves in ins.
M = bending moment.
I = moment of inertia of section.

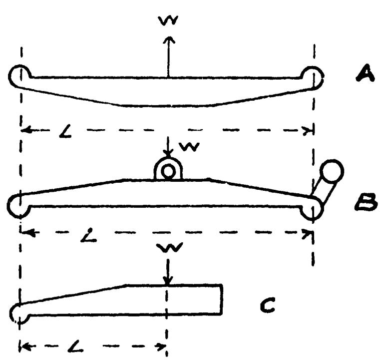


FIG. 28.

The modulus $E = 30 \times 10^6$, but owing to internal friction, irregularities of section, etc., a correction is needed; if 24×10^6 be taken for value of E, the results will conform to observed tests. Formulæ are given in the following table:—

TABLE XI.

LAMINATED SPRINGS. (See fig. 28.)

	Type.		
	A.	**B.**	**C.**
M	$WL/4$	$WL/2$	WL
I	$nbt^3/12$	$nbt^3/12$	$nbt^3/12$
Z	$nbt^2/6$	$nbt^2/6$	$nbt^2/6$
f	$3WL/2nbt^2$	$3WL/nbt^2$	$6WL/nbt^2$
δ	$WL^3/48EI$	$WL^3/24EI$	$WL^3/3EI$
δ	$\dfrac{WL^3}{96 \times 10^6 \times nbt^3}$	$\dfrac{WL^3}{48 \times 10^6 \times nbt^3}$	$\dfrac{WL^3}{6 \times 10^6 \times nbt^3}$

Semi-elliptic springs are frequently loaded unsymmetrically; in such a case BM = Wxy/L and $f = Wxy/Ls$; where x and y are the lengths of the two arms of the spring.

***Note.*—The stress f must never exceed 22 tons per sq. in. This is equivalent to limiting the energy stored in the spring relative to the weight of steel. The energy is the weight W, carried, multiplied by one-half the deflection in ft.; or**

$$E = \frac{W \times \delta}{2 \times 12}$$

A stress of 22 tons per sq. in. is equivalent to 3 ft.-lb. of energy per lb. of steel, and this must not be exceeded, irrespective of the quality of steel employed.

Spring calculations are complicated by the necessity of providing for the twisting of the axle due to braking, or the torque in driving, and these must be taken into consideration.

***Steel for Laminated Springs.*—The steel used is either a chrome-vanadium or a silico-manganese, for examples, see Table XII.**

TABLE XII.

PROPERTIES OF SPRING STEELS.

	Chrome Vanadium.	Silico Manganese.
Yield point .	184,800 lbs. per sq. in.	172,200
Ult. tensile .	194,900 lbs. per sq. in.	197,600
Red. area .	48 per cent.	36 per cent.
Elongation .	16 per cent. in 2 ins.	15 per cent. in 2 ins.
Brinell . .	450/455	4·55
C per cent. .	0·46	0·52
Mn ,, .	0·57	1·05
Si ,, .	0·17	1·95
Ni ,, .	0·15	—
Cr ,, .	1·40	0·05
Va ,, .	0·18	—

The foregoing is based on the assumption that the spring is strictly of rhomboidal proportions and that the initial curvature of each leaf is the arc of a concentric circle. In practice, mainly for appearance, increasing camber is given to the lower leaves to prevent separation of leaves on a rebound; the leaves are held together by a clip, rivet, or bolt, which puts initial stress into each leaf which increases as the shorter leaves are reached. This has the further bad effect of producing a localised stress in the leaf above where the lower leaf ends. To overcome this the end of the leaf is tapered both in breadth and thickness, but unless this is done with great intelligence it is ineffective.

The amount of extra curvature given to the lower leaves, known as 'nip,' should be a minimum. To check the rebound, additional leaves are frequently superposed above the top or master leaf, and the leaf next below it is preferably long enough at least partially to encircle the eye.

It is not necessary to weld the eye, and bushes are usually of rubber.

Good quality springs are frequently ground on the leaf surfaces after hardening, and sometimes interleaved with brass or zinc. With any laminated spring dust and mud should be excluded and continuous lubrication is imperative.

Helical Springs.—The stress conditions can be determined as follows:—

E_0 = transverse modulus of elasticity.
d = diameter of wire in ins.
R = mean radius of coil in ins.
n = number of free turns.
δ = deflection in ins.
W = load on spring in lbs.
W/δ = 'rate' per in. deflection.

then $W/\delta = E_0 d^4/64R^3 n$.

The value of E_0 is approx. 11,500,000; whence $W/\delta = 180{,}000 d^4/R^3 n$.

The maximum safe load = KD^3/R where K has a value of 10,000 to 12,000 lbs.

As in the case of laminated springs the periodicity is determined by the amount of deflection under static load.

Torsion-bar Spring.—Now coming into use as it lends itself to individual wheel springing. The bar is finished and ground to accurate dimensions the ends being left large and splined to engage the anchorage; one end being secured to the frame and the other to the wheel by means of links, the bar being thus subject to a twisting moment.

Let

T = twisting moment in lb. ins.;
R = radius of acting force in ins.;
W = tangential load;
d = dia. of bar in ins.;
f_0 = shearing stress lb./sq. in. at surface;
E_0 = transverse modulus of elasticity (11,500,000);
θ = degrees angle of twist at end of bar;
l = length of bar in ins.;

then $T = WR = \frac{\pi}{16} d^3 f_0$;

whence $f_0 = 16T/\pi d^3$; and $d = \sqrt[3]{16T/\pi f_0}$;

and $\theta = \frac{583 \cdot 61\, Tl}{E_0 d^4}$

The general question of suspension while mainly dependent on the springs, cannot be separated from the resilience of the tyres (a considerable matter since the introduction of large section low-pressure tyres) and the springs included in the term 'upholstery.' It is clear that the latter are of great importance, and may be of steel, sorbo rubber, or air cushions; hence the final oscillation period may be very different from that of the spring alone, and the resultant period is the one to consider. The vertical acceleration of passengers on the seats must not be great enough to lift them from the seats, hence they must at no time have an acceleration greater than that due to gravity. Generally speaking with a periodicity of 90 per min. at the springs the maximum vertical acceleration will probably not exceed this.

Frames.

The general principles of frame construction have undergone considerable modification in recent years. Formerly a considerable degree of general flexibility was deliberately aimed at, nowadays a maximum of rigidity is sought—particularly noticeable in passenger cars. With four independently sprung wheels a completely rigid frame would give best results. The universal employment of front-wheel brakes also profoundly effects the disposition of stresses in the frame. All types of frame are now heavily braced by structural diagonals; moreover, to secure the desired rigidity frames are of such a deep section that (as in the case of crankshafts) the frame structure is in the end subjected to very moderate specific stresses.

Car Weights.

Gross weight is the fully-laden weight of the complete car.

Tare weight is the unladen weight of the complete car including the body, but without the useful load of passengers or freight.

Chassis weight is the weight of the bare chassis without body or load, but includes oil, grease, water and fuel.

Front axle weight is the gross weight imposed on the ground by both front wheels and includes body and load.

Rear-axle weight is the gross weight imposed on the ground by both rear wheels including body and load.

Sprung weight is the total gross weight carried by the springs at either axle.

Unsprung weight is the weight of the axle, wheels, tyres, springs, brake drums and shoes, steering and brake rods., etc.

In calculating the frame stresses, the loads at each point of suspension of each unit must be taken, and the bending moment evaluated, taking account of the distributed load of the body and its fully laden contents. It is not necessary to calculate the shearing stresses as a rule, since these, for reasons already stated, are of very low specific value.

In the case of independent springing, and particularly when torsion-bar springs are used, the frame must be cross-membered at the spring anchorage to prevent buckling.

The following table gives the approximate weight of chassis parts for both passenger cars and commercial vehicles.

TABLE XIII.

APPROXIMATE PER CENT. WEIGHT OF CHASSIS COMPONENTS.

	Car.	Lorry.
Sprung weight:		
Radiator and water	4	5
Engine and gear box	37	27
Steering gear	1	2
Petrol tank and fuel	3	4
Frame	15	17
Total	60	55
Unsprung weight:		
Front axle, springs, wheels, tyres, brakes, steering levers, etc.	18	10
Rear axle, springs, wheels, tyres, brakes, etc.	22	35
Total	40	45
Total sprung and unsprung	100	100

For passenger cars the body weights are roughly as follows: body frame (steel), 14 per cent.; body framing complete with head fittings, 42 per cent.; steel panels, 10 per cent.; glass, mouldings, etc., 12 per cent.; trimming, 22 per cent. Total, 100 per cent.

For the whole car the chassis amounts to about 60 per cent.; body complete, 35 per cent.; and wings, valances, etc., 5 per cent. Total 100 per cent.

Dynamic Stressing.—The foregoing clearly applies to static loading only; dynamic stresses may often far exceed it, and concentration of load occurs as when striking obstructions, cornering, etc. When braking, there is always a transfer of load from rear to front axle. Thus if

w = weight transferred from rear to front axle;
W = weight supported by rear axle under static conditions;
μ = coefficient of friction between tyre and road;
h = height of centre of gravity in ft. above road;
L = wheel base in ft.

then $$w = \left(\frac{W \times \mu \times h}{L}\right)$$

This weight of course must be added to the static load supported normally by the front springs. It is thus seen that exact calculation of stresses is defeated by road conditions, hence a liberal margin of safety is required.

The stress in material should not exceed 20,000 lbs. per sq. in. where nickel alloy steel is used. A typical steel for hot pressed frames and cross-members would be 1 per cent. nickel-steel, yield point 68,000 lbs. per sq. in.; ultimate strength 94,000 lbs. per sq. in. Reduction of area 50 per cent. with 20 per cent. elongation.

It is to be noted that lack of rigidity at the front end of a frame is a fruitful contributory cause of 'wheel wobble,' due to the spring periodicity corresponding with that of the frame. By stiffening the latter and so raising the periodicity the trouble disappears.

Another change has taken place in the method of suspending the various units, which were formerly bolted solidly to the frame; great importance was then paid to three-point suspension to prevent distortion of *e.g.*, the engine, due to the flexion of the frame. All the components are now flexibly mounted on rubber cushions, the engine, for example, having a considerable elastic movement about its own principal axis.

This flexible mounting has not only prevented distortion in the components, but has, by insulating the noise producing elements, removed vibration and resonance from the frame, and also most of the remaining noise.

Modern technique in welding has enabled hollow box sections to be employed in frames which were formerly impracticable.

Very concentrated loading, such as occurs in a lorry fitted with a tipping body, requires special frame reinforcement at particular points. For these special calculations are necessary. The provision of a frame which will withstand modern speeds and the consequent road shocks, and will at the same time be relatively light is very largely a matter of good judgment by the designer.

Steering Gear.

The general lay-out of the modified Ackerman steering gear is well known, and the setting out of the linkage is most easily done by graphic methods; skill is required to effect the best all-round arrangement, which is a compromise between making the wheels track on wide or narrow lock. Satisfactory results can only be attained by experience.

The steering mechanism admits of wide variation in types, but takes one of five forms (1) The Nut and Screw, (2) the Worm and Sector, (3) the Cam and Lever, (4) Rack and Pinion, and (5) Re-circulating Ball.

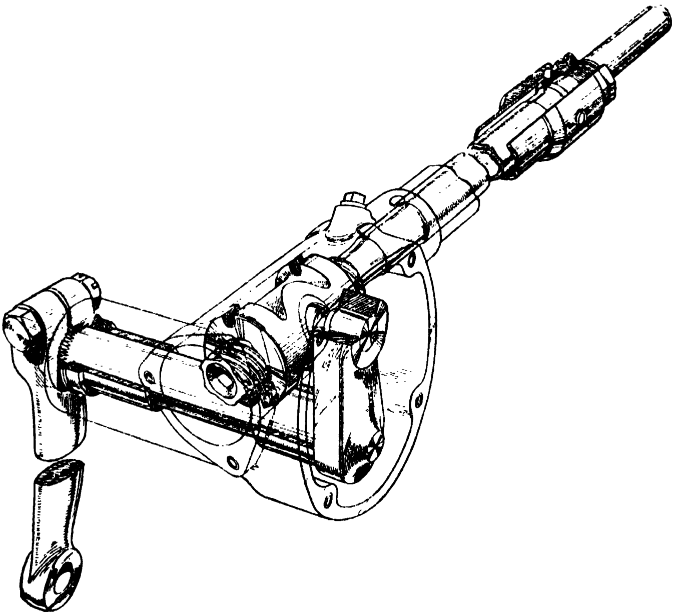


FIG. 29.

In all but the slowest running vehicles it is very desirable that the ratio of angular movement of the steering wheel to that of the first lever, should be differential, *i.e.*, as the lock increases so must the angular ratio or leverage, since on full lock great force is required to move the road wheels while a comparatively small force will suffice at the straight ahead position or zero lock. By making use of a toggle action it is possible to effect wide variation in the effective ratio for the two positions. With a plain worm and sector the angular ratio is constant at all points—largely to its disadvantage.

In certain designs of steering gear a variable ratio is provided, the ratio being at a minimum in the centre position and increasing as the wheel is turned in either direction.

An ordinary steering gear using the worm and nut is the Burman-Douglass, fig. 29. In this the sliding nut engages a spherical pin in a rocking arm; the differential motion referred to is present in a small degree (in one direction only) but the amount would seem to be negligible.

Fig. 30 illustrates the Marles double-roller type steering gear and fig. 31 a typical re-circulating ball type gear.

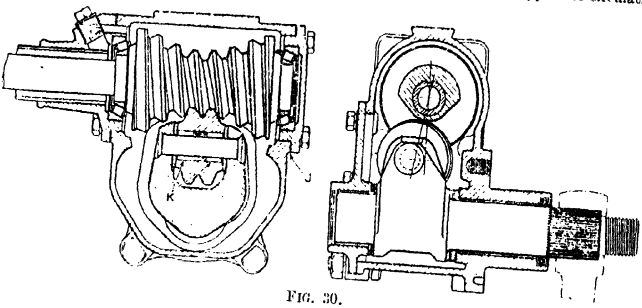


FIG. 30.

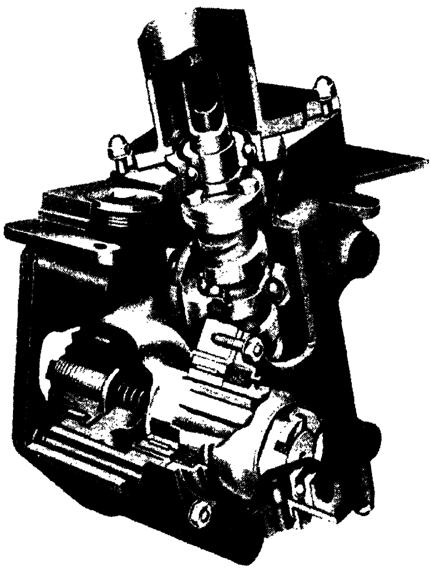


FIG. 31.

Brakes.

The function of the brake is to convert the kinetic energy of the vehicle into heat and dissipate this by radiation. The use of rubber tyres precludes the application of the brake to the wheel surface as is done in the case of railway wheels and those of horse-drawn vehicles, hence an independent brake drum is fitted to the side of the wheel. Owing to the large dimensions of the tyre, the brake drum cannot be much more than one-half the wheel diameter.

Brake drums are usually made of cast iron, alternatively they may be made from pressings or light alloy lined with cast iron; this material has the advantage of having a low coefficient of expansion, is less liable to score than alternative materials, and manufacture is relatively easy. Nickel and chromium are added with good results:—

Total carbon	**3·2**	**2·6**	**Manganese**	**0·8**	**0·5**
Combined carbon	**0·7**	**0·7**	**Nickel**	**—**	**1·0**
Silicon	**2·1**	**2·0**	**Chromium**	**0·35**	**0·45**
Phosphorus	**0·17**	**0·18**	**Sulphur**	**0·09**	**0·09**

the tensile strength of this is from 20 to 25 tons per sq. in.

Experience shows this to be by far the most satisfactory material.

The brake shoes are invariably faced with compressed asbestos fabric secured to the shoes with brass or aluminium rivets which has, when properly made, a coefficient of friction which lies between 0·3 and 0·4; the mean figure of 0·35 may be safely taken.

The wheel is kept revolving by a force equal to the weight on the ground × the coefficient of tyre friction applied at the effective radius of the tyre. Hence the braking force to overcome this can be expressed by the equation

$$P \times \mu_b \times d/2 = W \times \mu_r \times D/2.$$

where P = pressure on brake shoe; μ_b = coefficient of friction of brake lining; d = dia. of brake drum; W = the weight at the wheel tread; μ_r = road coefficient of friction; and D = the effective wheel diameter.

If $\mu_b = 0\cdot4$; $\mu_r = 0\cdot8$; and $D = 2d$, substituting,

$$P = \frac{W \times 0\cdot8 \times 2}{0\cdot4 \times 1} = 4W.$$

In this case the brake-shoe pressure is four times the vehicle weight.

If W = gross weight of vehicle the maximum braking effort or stopping force cannot exceed $W\mu_r$ when all the wheels are braked, and this is applied over the distance traversed before stopping = L ft.

The kinetic energy $E_k = Wv^2/2 \times 32\cdot2$ ft. lbs., and the potential energy $Ep = Wh$ ft. lbs., where v is the velocity in ft. per sec., and h is height fallen through during period of braking.

Hence $$L \times W\mu_r = E_k \pm Ep \text{ (on a gradient)};$$

so the stopping distance is

$$L = E_k(\pm Ep \text{ on a gradient})/W\mu_r.$$

this assumes a brake 100 per cent. efficient. It is thus seen that the shoe pressure may, and generally does, far exceed the gross weight of the vehicle.

If we take the last equation and substitute

$$L = E_k/W\mu_r$$

$$L = \frac{Wv^2}{64\cdot4 \times W\mu_r} \text{ and the W cancels out.}$$

Whence it follows that the weight of the vehicle has no effect upon its stopping distance, and a light car cannot be stopped in a shorter distance than a heavy one.

For any given deceleration the total braking force is given by the formula

$$P = W_f \times a/32\cdot2 \text{ lbs.}$$

where P = braking force; W_f = weight supported by braked wheels and a = desired deceleration. Here $P = p \times \mu_r$, where p = total shoe pressure and μ_b the coefficient of brake friction. The total pressure on the shoes for any desired deceleration is therefore

$$p = \frac{W_f \times a \times D}{32\cdot2 \times \mu_b \times d}$$

Where D and d are the respective diameters of wheel and brake drum.

This pressure is distributed over all the brake shoes, the combined area of which should be such that the unit pressure does not exceed 75 lbs. per sq. in. It can be shown that the total brake shoe area should be equal to W/15 sq. in.; where W = gross vehicle weight in lbs., and the brake drum is one-half the wheel diameter; this figure can seldom be attained in practice, but should be aimed at. It should be noted that the shoe should not cover much more than 100° of the drum surface or cooling is difficult.

***Stopping Distance.*—Assuming a tyre coefficient of friction of 0·6 the minimum stopping distances will be as follows:—**

Speed, m.p.h.	**10**	**20**	**30**	**40**	**50**	**60**
Distance, ft.	**5·55**	**22·25**	**50·0**	**88·8**	**138·5**	**200·0**

The brakes, however, are seldom in such condition that these minimum figures can be obtained in practice.

***Deceleration.*—This is expressed in the same terms as acceleration, and is directly proportional to the coefficient of tyre friction, thus**

$$\text{Maximum deceleration} = 32\cdot2 \times \mu_r \text{ ft./sec. per sec.}$$

(remembering that μ_r may be anything from 0·2 to 1·0) according to the state of the road, therefore the deceleration may vary 500 per cent., from 6·44 to 32·2 ft./sec. per sec.—a fact not always appreciated by drivers.

From the foregoing equations a car weighing 3,500 lbs., decelerating at 24 ft./sec. per sec. will require a total shoe pressure of 15,250 lbs., while an omnibus weighing 25,000 lbs. decelerating at 16 ft./sec. per sec. will require a total shoe pressure of 59,200 lbs. In both cases brakes on all wheels are assumed. Careful provision for forces of such magnitude is necessary. The pressures may be materially reduced by braking on surfaces running at higher speeds than the wheels, but both the area and heat dissipation of such brakes renders them of little use.

Legislation in Great Britain demands that there must be two independent means of operation, and one of them must be *direct*, and not assisted by any device, which, with modern weights and speeds, means that the *direct* system is of little use except for parking.

Servo Systems.—Practical considerations make it difficult to provide sufficient leverage to enable a driver to produce such forces as are needed, hence the use of various devices whereby mechanical assistance is given to the driver to produce the force required, and his muscular effort is so augmented that it only furnishes a small fraction of the power. Such systems are the Dewandre, in which the power is furnished by a large vacuum cylinder and piston, the vacuum being obtained from the suction of the engine when the throttle is closed; this system is mainly used for heavy commercial vehicles.

For cars, the systems in most general use are the hydraulic, of which the Lockheed is the best known; the Bendix, where the brake shoes are partially floating and the reaction of one shoe furnishes the force required to apply the other, and the Girling.

When an expanding brake is arranged with the two shoes resting on a fixed abutment and an expanding device diametrically opposite to the abutment, one shoe automatically increases its pressure as the result of wedging effect, when applied, and is called the leading shoe, whilst the other automatically decreases the pressure of its application and is called the trailing shoe. The leading shoe is the one in which the direction of rotation is from the expander towards the fixed abutment. Various devices have been produced with the object of causing the leading shoe to transmit some of its increment of pressure to the trailing shoe, and so increase the overall efficiency of the brake for the same pedal pressure.

Girling System.—Fig. 32 shows the arrangement of shoes and expanders assembled; in fig. 33, the hardened steel cone (1) actuated by the pull rod (2) causes the plungers (3) to move outwards through the interposed rollers (4). The plungers are in an oil-tight case which floats on the studs

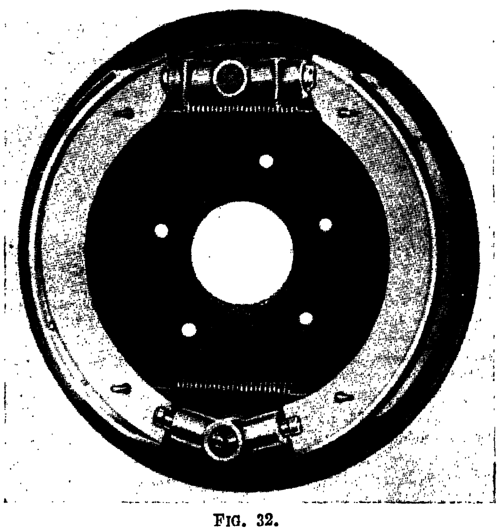


FIG. 32.

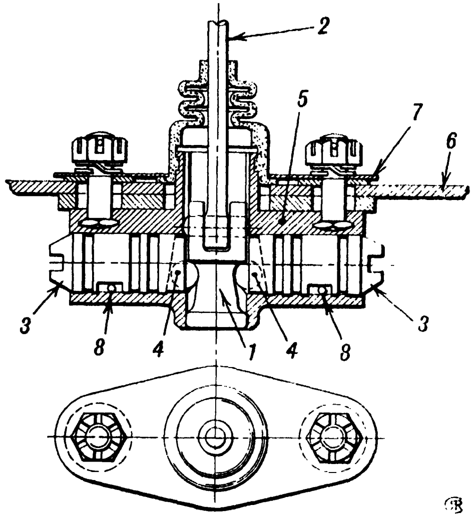


FIG. 33.

and spring washers (7), hence, the force is balanced between the shoes. Fig. 34 shows the adjustment at the heels of the shoes ; the cone (A) is forced by the screw between the plungers (C) which are thus extended carrying the shoes with them. From pull rod to shoe tip is a leverage of 6·33

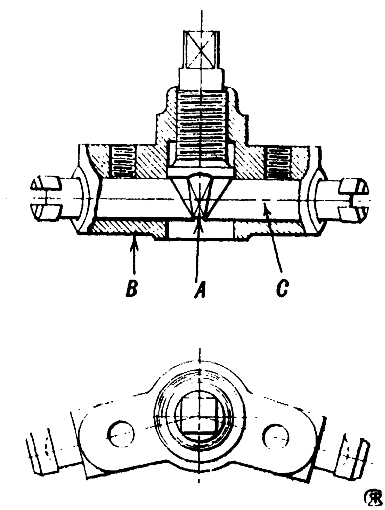


FIG. 34.

to 1. Fig. 35 shows the linkage arrangement. By means of floating levers the pedal movement is distributed equally between the eight shoes, all cross shafts are eliminated and the robust balance levers prevent any spring over which pedal movement has to be wasted.

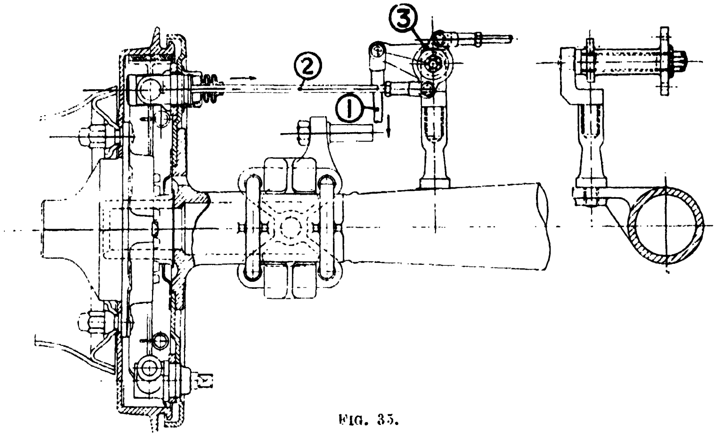


FIG. 35.

Fig. 36 illustrates the Girling two leading shoe arrangement in which, by means of bell-cranks and a connecting pushrod, the force applied by the expander is transmitted to the leading end of the second or trailing shoe, which then becomes a leading shoe being in full contact with the brake

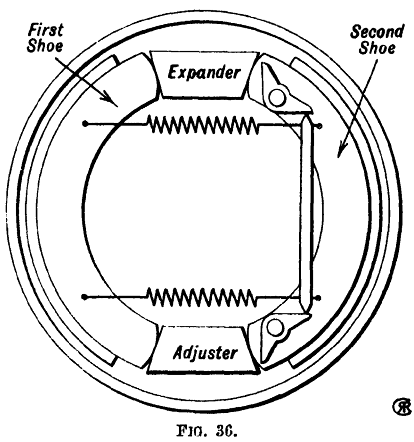


FIG. 36.

drum. Similar results are obtained in the Girling sliding shoe arrangement which, as the name implies, comprises a sliding second shoe, mounted on a steel carrier. The carrier and the standard fixed or leading shoe are operated in the normal Girling manner, when the brakes are applied the sliding shoe rotates until it comes into contact with the torque stop which is attached to the backing plate adjacent to the expander, thus becoming a leading shoe.

The Girling system is suitable for either mechanical or hydraulic operation ; the latest application, designed for use with cars having independent front suspension, being a combination of both, *i.e.* hydraulic operation for the front brakes and mechanical operation for the rear brakes.

Bendix Cowdrey System.—The operation of this sytem, as will be seen by reference to fig. 37, is similar to the Girling, balls being employed in the plunger or expander unit instead of rollers.

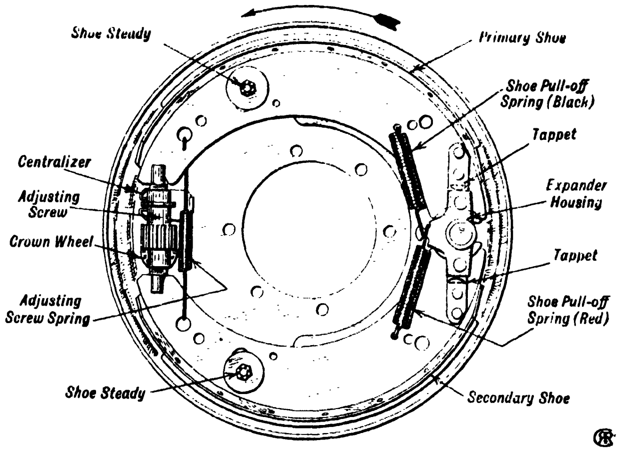


FIG. 37.

The adjuster unit, which is diametrically opposite to the expander, floats. When the operating plunger moves, the balls are pressed against the bevelled ends of the tappets, fig. 38. The bevels

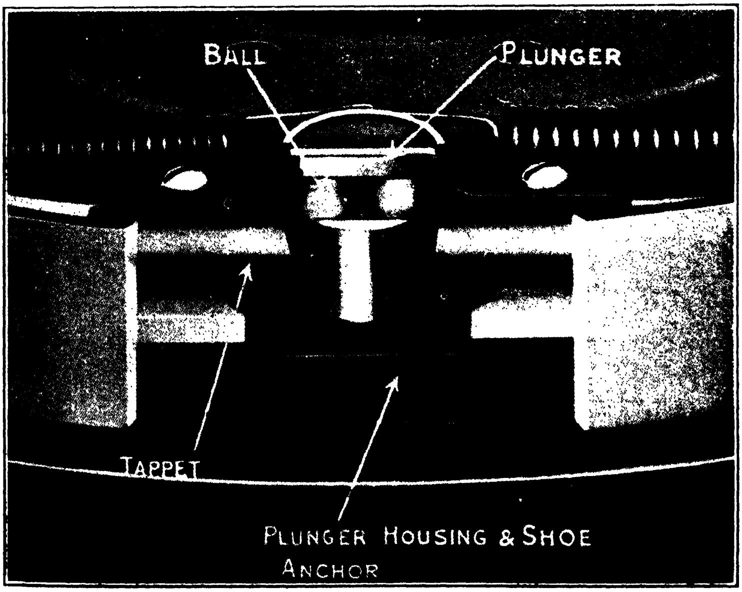
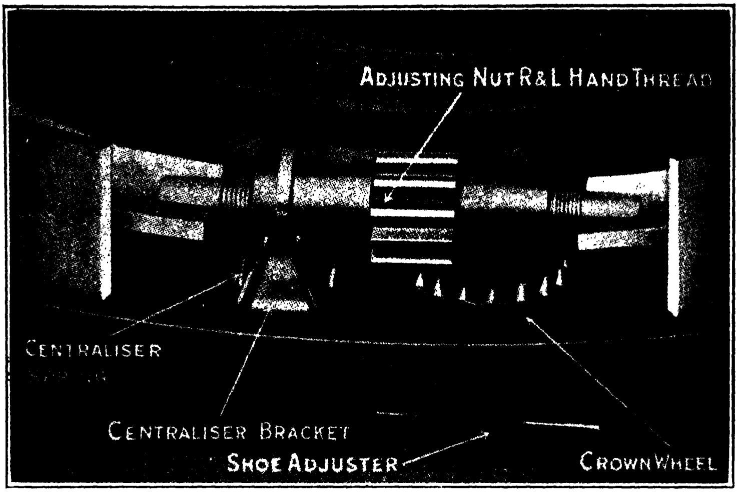
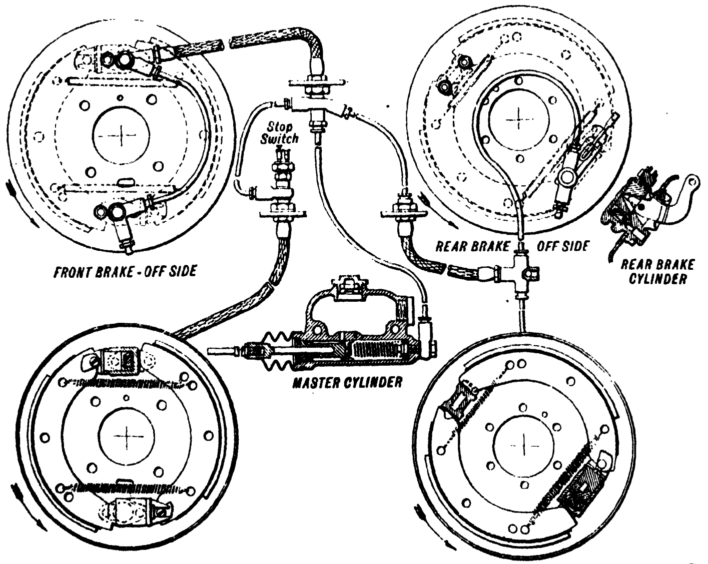


FIG. 38.



ADJUSTING NUT R&L HAND THREAD
CENTRALISER
CENTRALISER BRACKET
SHOE ADJUSTER
CROWN WHEEL

FIG 39.



Stop Switch
FRONT BRAKE - OFF SIDE
REAR BRAKE OFF SIDE
REAR BRAKE CYLINDER
MASTER CYLINDER
FRONT BRAKE - NEAR SIDE
REAR BRAKE - NEAR SIDE

FIG. 40.

are at slightly different angles so that the primary shoe moves out first; it is carried round slightly by the friction between the lining and the drum, and its torque is applied through the adjuster to the secondary shoe. This servo method of operating saves about *two-thirds* of the pedal pressure with corresponding reduction of stresses in mechanism.

Fig. 39 illustrates the adjusting means at the heel of the shoes. A compensated linkage is employed similar to that used on the Girling system. The Cowdrey system is suitable for either mechanical or hydraulic operation, and both these are employed according to the particular application.

Lockheed Hydraulic System.—In the Lockheed hydraulic system, the pressure exerted on the brake pedal is conveyed to the brake shoes by a column of special hydraulic fluid. The diagrammatic illustration, fig. 40, shows the Lockheed two leading shoe system, with two wheel cylinders placed between the front brake shoes and one wheel cylinder between the rear brake shoes. The master cylinder has a single piston, likewise the wheel cylinders, and all pistons are provided with rubber cups to maintain pressure and prevent loss of fluid.

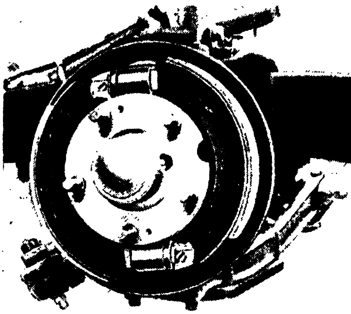


FIG. 41.

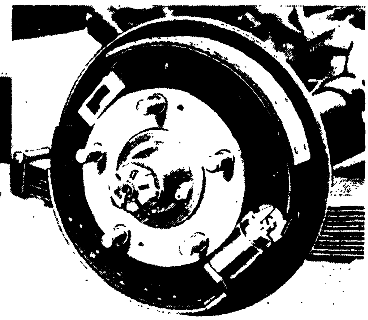


FIG. 42.

When the brake pedal is operated, the master cylinder piston applies a force to the fluid causing the single piston in each front wheel cylinder to come into contact with the leading tip of its respective brake shoe, while the trailing tip of the shoe finds a floating anchor by utilising the closed end of the actuating cylinder of the other shoe as its abutment. At the same time the rear wheel cylinder, which is free to slide in an elongated slot in the rear back plate between the tips of the leading and trailing shoes, operates on the tip of the leading shoe and this shoe abuts against a fixed anchor block at the bottom of the back plate, the web of the shoe being free to slide in a slot in the block. The trailing shoe is located in a similar manner between the anchor block and the closed end of the wheel cylinder, and is free to slide and therefore self-centering. The trailing shoe is operated by movement of the cylinder assembly as a result of the reaction of the leading shoe against the brake drum. Further effort on the pedal increases the force applied to the brake shoes.

Mechanical operation of the rear wheels is provided by means of a lever, which is connected to the handbrake lever through the necessary linkage, and the outer piston.

The front and rear brakes are shown in figs. 41 and 42 respectively.

Axles.

Both front and rear axles have essentially similar functions. Both are subjected to vertical and horizontal bending moments, and to torque. Front axles (generally) carry the steering mechanism, and rear axles (generally) the driving arrangements. Although very different in appearance, axles for independent wheel suspension have generally the same functions as conventional forms, and axles for front-wheel drive, or four-wheel steering are entirely special and **will not be considered. The axles of six-wheeled vehicles are special cases of suspension** rather **than of axle design.**

Normally front axles are of I section except the part between the spring seat and the wheel which alone is subject to torque. Generally the axle ends in a boss carrying the king pin to which the forked steering axle is fitted, this being known as the 'reversed Elliott' type.

There are at least four main types of rear axle housing in general use to-day, the built-up type, the pressed steel type, usually comprising two stampings welded together, the cast type, which is

really of composite construction, and the forged type. The pressed steel type is universal on the lighter cars and the forged steel type on the heavier commercial vehicles. The bevel or worm gears are supported, together with the differential gears, in a malleable-iron or light alloy casting which forms the cover for the axle.

In calculating stresses, the force causing a horizontal bending moment due to traction or braking cannot exceed μW which also limits the torsion. The vertical and horizontal bending moments can be combined into a common resultant.

If W = **the vertical bending moment;**
H = **the horizontal bending moment;**
T = **the twisting moment;**
M = **the combined moment;**

then $M = W + H + \sqrt{[(W + H)^2 + T^2]}$.

As, however, the static stresses (which alone can be evaluated) are much lower than stresses due to impact, very liberal allowance has to be made by a large factor of safety.

Front wheels are sometimes inclined outwards, but usually not more than 1°, and the king-pin is inclined in the opposite direction, pointing in fact to a point from $\frac{1}{2}$ in. to 1 in. inside the point of wheel contact. The king-pins are also given a rearward slope of 2 or 3° to ensure castor action of the wheel.

Both front and rear axles are very largely the product of proprietary manufacture, this being the case with all kinds of vehicles.

Lubrication.

The important consideration in selecting oil is viscosity at the normal running temperature, measured by a viscometer, the viscosity decreasing as the temperature is raised. The colour of the oil is no indication of its characteristics nor is its specific gravity.

If two surfaces are in contact and have relative movement there are three possible kinds of friction, all of which obey different laws.

(*a*) *Dry* friction, the surfaces being unlubricated and therefore in contact.

(*b*) *Greasy* friction, the surfaces being in contact, but lubricated.

(*c*) *Viscous* friction where the surfaces are completely separated by an oil film.

The latter is the normal condition and the only one to be considered. When a shaft rotates in a bearing it tends to climb up the side of the latter, and so assumes a position where the shaft axis is eccentric to that of the bearing and the oil is dragged round in a continuous stream at high velocity forming a wedging action which keeps the surfaces apart. There is a critical pressure-velocity figure where the oil film will break down, and since however smoothly the surfaces may be finished they consist microscopically of small projections which will penetrate the film, causing seizure. As this takes place the condition of *greasy* friction has occurred.

It is known that oils of animal or vegetable origin, *e.g.* sperm and castor oils, possess a condition of *oiliness* much more than mineral oils.

Hence, if the lubricant is defective in oiliness the friction at the metallic projections will be very high and heat is generated which decreases the general viscosity, ending in complete seizure. Oiliness will defer this condition. The heat is removed by, partly, conduction from the bearing surfaces, but mainly by oil running through the bearing and escaping at the sides. The advantage of forced lubrication being thus mainly due to the cooling effect of excess lubricant. An alternating load is less destructive than one unidirectional.

Again, Ricardo points out that a narrow bearing loses more oil through side leakage than a wide one, so is less efficient with natural lubrication, while with forced lubrication a narrow bearing can carry a heavier continuous load per sq. in. of projected area, because of the greater quantity of oil which can be passed through it ensuring better cooling. Again, at low speeds, the wedging action of the oil film is slow. It has been shown that even a bronze bearing may be crushed without destroying the oil film. Gudgeon pins are subject to the lowest rubbing velocity, but owing to the reciprocating motion, provided they are properly supported against deflection pressures up to 6,000 lbs. per sq. in., can be carried without forced lubrication.

***Types of Oils.*—These are either mineral oils, hydrocarbons of the paraffin series, or those of animal and vegetable origin. Since the latter possess greater oiliness it is usual to compound them with oil of mineral origin. They are seldom used pure on account of their cost.**

All oils tend to carbonise at high temperatures, and the resulting soot, with water vapour, causes thick sludge to form in the crankcase, which has little lubricating value, and tends to choke oil passages. Leaky pistons, allowing petrol to mix with the oil, reduce viscosity, hence the necessity for regular replenishment.

In engines of high performance the oil is circulated through a radiator for cooling it, and is also filtered to remove sludge and foreign matter.

Most if not all proprietary oils show consistent viscosity under working conditions, but those which lose an appreciable bulk under high temperatures are unsuitable, as are those whose constituents separate in use.

***Greases.*—For slow moving bearings greases are often used, *e.g.* spring eyes, etc., but there is some danger of grease solidifying unless very frequently renewed under high pressure, hence a thick oil applied from a grease gun has advantages.**

Chassis Lubrication.

ADVANTAGES OF OIL FOR LUBRICATION

(C. T. Myers.)

(1) Sustains as heavy a bearing as grease with less friction.

(2) Fed to one properly located point in a bearing will quickly spread over the entire surface.

(3) Will flow through a very small hole, and can be conducted a considerable distance by capillary attraction and surface tension.

(4) Carries much less dust than grease, can be filtered, and not so much exposed to dirt as grease, when in bulk or when being applied.

(5) Contains no inert matter to clog holes or channels.

(6) Engine oil can be used.

(7) Easier to apply automatically than grease.

(8) A flow of clean oil tends to clean a bearing, and bearings arranged for oil lubrication can be cleaned with petrol or paraffin.

(9) When oil is used on spring bolts, the seepage works down into the spring leaves, preventing rust.

LUBRICATION OF TRANSMISSION AND DIFFERENTIAL.

The following characteristics of the lubricant are necessary :—

(*a*) Must be of a character and consistency to coat, follow, cushion, and adequately lubricate the gear mesh.

(*b*) Must be capable of thorough distribution to any be[illegible] or plain bearing, and must have no corroding effect on the balls, rollers, [illegible]

(*c*) Should be of a body and character to minimise resistance and power loss due to the churning action of the gears. In the case of transmission, it should offer the minimum of resistance to the manual act of gear changing.

(*d*) Should be capable of carrying off the greatest amount of heat, and of exerting a washing action on the gear teeth and bearing surfaces.

(*e*) Should be a body to prevent leakage from the gear box or back axle under all normal operating conditions.

Electric Traction on Tramways.

The technical arrangements for electric tramways are practically standardised all over the world. Two 4-pole ventilated series or compound motors are used, usually with interpoles. The nominal rating of each motor is 30 to 60 h.p. with double-decked cars. The current transmission is effected by means of an overhead trolley-wire and rail return, collection being performed with a wheel trolley or sliding bow-collector. The voltage is most commonly 500 to 600, although a few American inter-urban roads have adopted a pressure of 1,200 volts. For control, a drum-controller or contactor is used, allowing of series-parallel and field-weakening methods for the regulation of the speed. Braking is by air or electric devices, the latter including rheostatic and magnetic track-brakes. Hand-controlled braking is used as a standby. For tramway sub-stations, rotary converters have replaced the older motor-generator sets, but mercury-vapour rectifiers are coming into use for modern installations (see Section XXVI (Vol. I), Electrical Engineering).

Tractive Resistance on Tramways.

The average pull required per ton of load to keep a car moving is 30 lbs. With exceptionally clean rails, as during a heavy rainfall, and with the rail grooves void of dirt, the pull may be as low as 20 lbs. per ton, or below this figure, approximating to the 10 or 12 lbs. of railway work. Again, a very dirty, clogged rail may demand more than the 30 lbs. Multiplying the pull by the speed in feet per minute gives the power in foot-pounds per minute, so that the horse-power is H.P. $= \frac{T \times W \times S}{33{,}000}$, where T is the pull per ton necessary, W is the load in tons, and S the speed in feet per minute. This rule gives the horse-power on the level. To find the power on a grade, the vertical rise of the car is found by dividing the speed of the car in feet per minute by the length of road in which there is a rise of 1 foot. The number of feet of vertical lift multiplied by the total weight in pounds gives the work done per minute against gravity, or $\frac{S \times W}{F}$, where W is the load, S is the speed, and F is the distance in which the road rises 1 foot. Thus on a 1 in 30 grade, F = 30. **The addition of the two results gives the total horse-power. Track resistance is increased at curves, also increased horse-power is required during acceleration. The latter depends not only on weight of car as a whole, but also on moment of inertia of rotating parts.**

RHEOSTATIC BRAKING.

Electric braking consisted first of simple rheostatic braking, the drag on the motor armatures, acting as generators, causing the retarding force necessary. Resistance is inserted at first to prevent excessive currents and retardation and too severe strains on the motor windings, etc. This resistance is gradually cut out as the controller handle moves over the braking notches.

Electro-magnetic braking was added later, the braking effect being then partly electrical and partly frictional.

One weak point of electric braking was that, should the car be stopped when going up an incline, there was a tendency to run back, and then the electric braking had no effect unless the reversing handle was moved into the backward position. Omission to do this caused a number of accidents. Many modern cars are equipped with air brakes.

Regenerative Braking Systems for Tramways.

An effective system of regenerative braking has the advantage of enabling an increase of schedule speed to be obtained economically. A number of methods have been developed. In one, the two motors of a car are connected permanently in series and compound wound, the shunt fields being permanently series-connected across line and earth. Speed regulation is obtained by control of the motor field currents. The series field turns are modified in strength by a diverter which is shunted across the series coils, the object being to utilise the series characteristics of the motors as much as possible when motoring and to reduce the reverse compound effect when regenerating. The system as applied to British conditions embodies the additonal complication of series-parallel control.

A SUMMARY OF THE BOARD OF TRADE REGULATIONS FOR ELECTRIC TRAMWAYS.

The following is a brief summary of the principal regulations affecting electric traction :—

When any rails on which cars run, or any conductors laid between or within 3 feet of such rails, form any part of a return, such part may be uninsulated. All other returns, or part of a return, shall be insulated, unless of such sectional area as will reduce the difference of potential between the ends of the uninsulated portion of the return below 7 volts.

Where any uninsulated conductor laid between or within 3 feet of the rails forms any part of a return, it shall be electrically connected to the rails, at distances apart not exceeding 100 feet, by means of copper strips having a sectional area of at least $\frac{1}{16}$ of a square inch, or by other means of equal conductivity.

All uninsulated returns shall be connected to the negative pole of the generator; the negative terminal of the generator shall be also connected through a current-indicator, mentioned below, to two separate earth connections, which shall be placed not less than 20 yards apart. This clause enacts that the two earth connections may be replaced by making one connection to a water-pipe of not less than 3 inches internal diameter, provided that permission be obtained from the owner of the water-pipe and the person supplying the water.

The earth connections referred to in this regulation shall be constructed, laid, and maintained so as to secure electrical contact with the general mass of earth, and so that an electro-motive force not exceeding 4 volts shall suffice to produce a current of at least 2 amperes from one earth connection to the other through the earth, and a test shall be made at least once in every month to ascertain whether this requirement is complied with.

When the return is partly or entirely uninsulated the company shall, in the construction and maintenance of the tramway, (a) so separate the uninsulated return from the general mass of earth and from any pipe in the vicinity, (b) so connect together the several lengths of the rails, (c) adopt such means for reducing the difference produced by the current between the potential

of the uninsulated return at any one point and the potential of the uninsulated return at any other point, and (*d*) so maintain the efficiency of the earth connections specified in the preceding regulations as to fulfil the following conditions, viz. :—

(i.) That the current passing from the earth connections through the indicator to the generator shall not at any time exceed either 2 *amperes per mile of single tramway line or 5 per cent. of the total current output of the station.*

(ii.) That if at any time and at any place a test be made by connecting a galvanometer or other current indicator to the uninsulated return and to any pipe in the vicinity, it shall always be possible to reverse the direction of any current indicated by interposing a battery of three Leclanché cells, connected in series, if the direction of the current is from the return to the pipe, or by interposing one Leclanché cell if the direction of the current is from the pipe to the return.

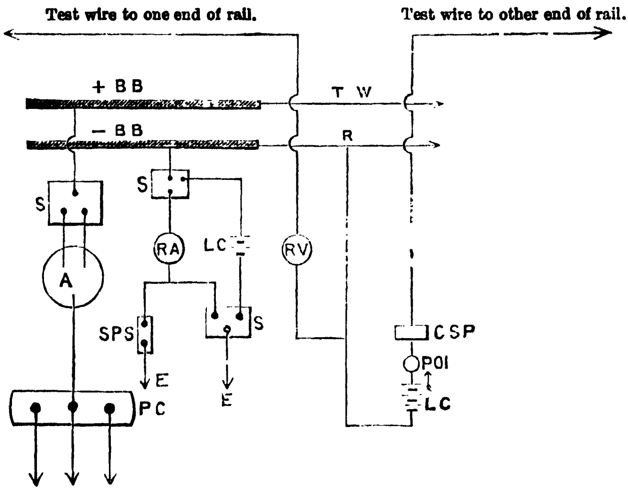


FIG. 43.—Diagram of Board of Trade Switchboard.

A, Ammeter reading $\frac{1}{10}$ – 3 and $\frac{1}{2}$ – 10 A.; B B, Bus bars; R, Return; T W, Trolley wire; S, Two-way switches; R A, Recording Ammeter 2 to 25 A; R V, Recording voltmeter 2 to 10 V.; L C, Leclanché cells; S P S, S.P. switch; E, Earth; P C, Three-way plug connections; P O I, P.O. indicator; C S P, Contact of sensitised paper.

In order to provide a continuous indication that the condition (i.) is complied with, the company shall place in a conspicuous position a suitable, properly connected, and correctly marked current indicator, and shall keep it connected during the whole time that the line is charged.

The owner of any such pipe may require the company to permit him at reasonable times and intervals to ascertain by test that the conditions specified (ii.) are complied with as regards his pipe.

When the return is partly or entirely uninsulated, *a continuous record shall be kept by the company of the difference of potential during the working of the tramway between the points of the uninsulated return farthest from and nearest to the generating station. If at any time such difference of potential exceeds the limit of 7 volts, the company shall take immediate steps to reduce it below that limit.*

The current density in the rails shall not exceed 9 amperes per square inch of the cross-sectional area.

The insulation of the line and of the return when insulated, and of all feeders and other conductors, shall be so maintained that the leakage current shall not exceed one-hundredth of an ampere per mile of tramway. The leakage current shall be ascertained daily before or after the hours of running when the line is fully charged. If at any time it should be found that the leakage current exceeds *one-half* of an ampere per mile of tramway, the leak shall be localised and removed as soon as practicable, and the running of the cars shall be stopped unless the leak is *localised and removed within 24 hours.*

The insulation resistance of all continuously insulated cables used for lines, for insulated returns, for feeders, or for other purposes, and laid below the surface of the ground, shall not be permitted to fall below the equivalent of 10 megohms for a length of one mile.

Trolley Buses.

Under certain traffic conditions where electric power is available, railless traction provides a valuable alternative to the motor omnibus and the electric tramway. Two overhead conductors are provided, as lead and return, and the vehicles carry trolley poles with under-running trolleys collecting the current from the conductors. These trolleys permit the vehicle to deviate at least 15 ft. from the centre-line on either side, so that they can pass other vehicles, and draw in to the kerb at stopping places. Railless tramcars compared with motor omnibuses have higher acceleration, lower working costs, longer life, and greater elasticity as regards the number of passengers carried, while they give a more reliable service and a greater annual mileage, and can climb hills with greater ease.

Present-day, single-deck trolley buses can accommodate up to 40 passengers. Double-deck, three-axle trolley buses can accommodate up to 70 passengers. The modern trend of design is to use all-metal bodywork instead of the composite construction body, and consequently, by reason of Ministry of Transport Regulations, low-voltage lighting has to be used. This may be provided by means of a small motor-generator set of approximately 1,600 watts output; or by a dynamo, belt-driven from the main motor. If it is required to manœuvre the bus when the overhead power supply is not available, special batteries are used. These are generally of the alkaline type, consisting of two 30-volt, 50-ampere-hour batteries; they are connected together in series for traction work and in parallel for lighting.

Where composite construction bodies are used, the lighting may be obtained from the overhead supply and the lamps are connected together in series. An arrangement is provided whereby if one lamp fails it short-circuits across its terminals, and consequently does not extinguish the other lights on the same circuit. A small auxiliary dynamo, either driven direct off the front end of the main motor, or belt-driven from the propeller shaft, provides the navigation and emergency lights.

The maximum speed of a trolley-bus depends upon the nature of the route, but is generally in the region of 35 m.p.h. The average speed of a single-deck trolley-bus is between 12 and 14 m.p.h., and the average speed of a double-deck, seventy-seater trolley-bus is 11½ m.p.h. The energy consumption of a two-axle, single-deck trolley-bus is 1·3 units per car-mile, and of a three-axle, seventy-seater trolley-bus 2·25–2·5 units per car-mile. The motor omnibus has an average speed of 10 m.p.h., and the fuel consumption of a Diesel-engined bus—fifty-six seater type—is 10–12 miles per gall. These figures, of course, depend upon the route and the nature of the service.

Trolley-Bus Motors and Controls.

Where single motors are preferred, the one-hour rating is of the order of 80 h.p., and these are now being built with a weight of 12 lb. per h.p. Double motors, each of half this rating, permit of series-parallel control and provide an economical half speed for running in traffic or fog. The double motor dispenses with the differential, by mounting one motor on each side of the chassis, and allows the bridge system to be used in the transition from series to parallel, thereby preventing change of torque. When series-wound motors are used, field control is resorted to in order to obtain improved starting conditions. Compound-wound motors are used where regeneration is required. The shunt field is employed to control the field strength, regeneration being obtained by increasing the excitation. This is sometimes done in conjunction with diversion of a portion of the series field.

Remote control is provided by a pedal-operated master controller. The contactor gear, which can be either electrically or pneumatically operated, is mounted either in the driver's cabin or on the chassis frame.

In order to save the mechanical brakes and also reduce the energy consumption, regenerative control in conjunction with rheostatic braking is extensively used. The vehicles regenerate down to 14–12 m.p.h., after which the rheostatic brake comes into operation, giving a maximum retardation in the region of 4 m.p.h. per second. Where buses are operating in very hilly districts, run back brakes can be fitted. These consist of a contactor which is normally held open by the current; should this fail or the bus start running backwards, the contactor closes and limits the bus speed to 2 m.p.h. Another safety device in the form of a coasting brake, which, by engaging a suitable control lever, limits the speed to 10–15 m.p.h. may be fitted. Deviation indicators, for driving in fog, indicate how far away from the overhead wires the bus may be. Leak alarms, showing when there is any chance of electric shock to the passengers, and dewirement indicators to warn the driver that the trolleys are off, may also be incorporated.

Electric-Battery Traction.

A secondary battery supplies d.c. driving motors. The scope of the vehicle is limited by the capacity of the battery and the need for frequent charging. Batteries are frequently used on trucks, on road vehicles and on shunting locomotives.

Industrial Electric Trucks.

For moving material about railway stations, factory premises, etc., electric battery trucks have been widely adopted, capable of carrying up to 5,000 lbs. and running at speeds up to 8 miles per hour; the net weight of the trucks is 1,600 to 2,500 lbs., and the mileage per charge 15 to 25 miles. In a railway warehouse 11 tractors and 600 trailers handled 130,000 tons per annum at an average cost of 15·1*d*. per ton. The loading platform can be fixed or made to lift, as required.

Charging Battery Vehicles.

The charging of vehicle batteries is nearly always controlled automatically, and does not involve the removal of the battery from the vehicle; a connection being taken from the charger to a socket on the vehicle. The charging rate is controlled automatically by the characteristic of the charging generator or rectifier and the battery, the result being that the current falls as the battery voltage rises on charge. The duration of the charge is usually controlled by the ampere-hour meter on the vehicle. This meter is set to run 10–15 per cent. slower on charge than on discharge, so as to allow for the overcharge required by the battery, and when the needle reaches full charge it trips out a circuit-breaker in the charging circuit.

Battery vehicles are well adapted for charging during the night hours, and specially low tariffs for this purpose are now offered in many districts. A standard Charging Plug and Socket is specified in B.S.I. Report, No. 74—1937.

Petrol-Electric Lorries.

In the petrol-electric vehicle the motive power is provided by a petrol engine, but the mechanical gear and clutch are replaced by a very flexible electric transmission. In a recent type of Stevens petrol-electric lorry of 3 tons capacity, the generator driven by the engine gives from 300 amperes at 80 volts to 230 amperes at 110 volts, and in addition to driving the vehicle it can be used to supply power for charging a battery or electric lighting, welding, etc. The electric motor is supplied through a special controller which gives forward and reverse connections with a wide range of speed variation, and when put in the neutral position enables the generator to supply electricity for external use. In the last case the generator can be given any desired characteristic—under, level, or over-compounded. A 3-ton waggon can supply three arc-welding equipments at 60 volts.

Oil-Electric Traction.

The Diesel engine operates most efficiently when developing a constant torque at a constant speed, *i.e.* a constant output. Thus, in order to utilise the Diesel engine to its full capacity, some device is required to convert the constant torque of the engine into the variable torque needed to haul the train. Instead of a mechanical gearbox, an electric transmission is frequently used for this purpose. This entails an electric generator and driving motors—the latter are usually direct-current traction motors.

Both constant-speed and variable-speed Diesel engines are used. Engines with critical speeds are included in the former class and various devices for controlling the power have been developed. In the Armstrong-Whitworth method of automatic constant-power control for constant-speed engines, it is claimed that, when desired by the driver, the engine output is kept within about $\pm$ 2½ per cent. of the correct value. Other well-known systems employ a generator with a characteristic designed to approach a hyperbolic curve, so that the product of terminal voltage and current is constant.

In the case of variable-speed engines, which can run at any speed between idling and full speed, the generator is designed to give a hyperbolic output curve. Here again several devices are in use to obtain the desired result.

In competition with the mechanical drive, the Diesel-electric drive is obtaining wide use in shunting locomotives, in motor-coaches, and in rail-cars. By fitting the 'dead man's' handle, one man only is required on the footplate. Diesel shunting is considered to be a necessary auxiliary to railway electrification, where there are objections to electrifying the sidings. Diesel locomotives have also received much publicity for driving high-speed trains.

Much of the earlier criticism regarding unreliability, noise and smell of Diesel locomotives and motor-coaches is no longer applicable, while improved facilities for maintenance and repair have added to the popularity of this form of traction.

See also Descriptive Section XXX.

SECTION XXXI

PART I

AIRCRAFT . . . . pp. 389–444

(Revised.)

PART II

AERO ENGINES . . . . pp. 445–466

(Revised.)

SECTION XXXI

PART I

TYPES OF AIRCRAFT—AERODYNAMICS—SCALE EFFECT AND DYNAMIC SIMILARITY—AIRSCREWS—PERFORMANCE—DESIGN.

The Aeroplane is commonly known as a 'Landplane' when used for land purposes, or a 'Seaplane' or flying-boat when used for maritime purposes. Machines of this type are initially able to leave the ground or water and subsequently maintain themselves in flight by virtue of their forward motion relative to the surrounding air.

An aeroplane consists essentially of five components:—

1. The main planes, by means of which the aeroplane is sustained in flight.

2. The fuselage or hull, in which the pilot, passengers, and/or other load, such as cargo, petrol, and oil, are carried.

3. The tail, which is rigidly attached to the rear of the fuselage or hull and comprises a tail plane and elevators in a horizontal plane, and fins and rudders in a vertical plane. This component is mainly responsible for the control of the aircraft.

4. The undercarriage, which, in the case of land planes, consists of a braced structure connected to the underside of the fuselage near its front end, supporting an axle carrying wheels, by the aid of which the machine leaves or alights on the ground.

In the case of twin- or multi-engined aeroplanes, this component may be in duplicate.

In getting off and landing, the tail is protected by a small tail wheel or skid attached to the rear end of the fuselage.

In Arctic regions, the wheels are replaced satisfactorily by long skids, while some aeroplanes are convertible into seaplanes by replacing the undercarriage structure by a similar one having two floats instead of wheels.

In the case of flying-boats, the hull serves the dual purpose of the fuselage and the undercarriage. Present-day hulls, when manufactured in duralumin, have a very important advantage over wooden hulls owing to the absence of water soakage, which in the latter hulls may amount to several cubic feet; this equivalent weight is thus available for useful load.

5. The engine, being the unit controlling the speed and height to which a machine may attain. In single-engined aeroplanes it is placed in the nose of the fuselage, so that the airscrew operates as a tractor; installations employing a pusher airscrew are rare.

When outboard engines are used, the mountings generally form part of the main plane structure.

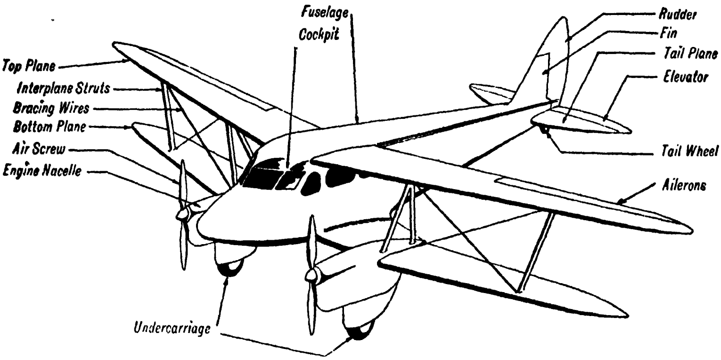


FIG. 1.—Main Components of a Small Biplane.

GEOMETRY OF PLANES.

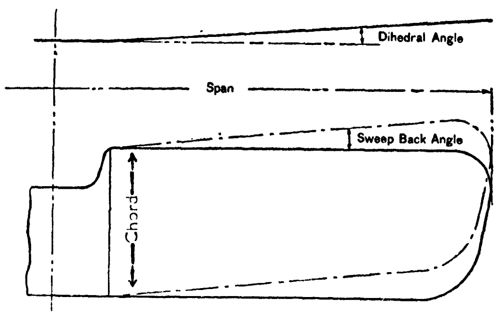


FIG. 2.

The several components are indicated in fig. 1, while the geometry of the planes is indicated in fig. 2.

AERODYNAMICS.

Wind Channels.

An essential instrument for aerodynamic research is the wind channel in which the forces and moments on a model suspended in an air stream are measured. But it is impossible to apply the data thus obtained to the full-scale machine or component (as distinct from a detail such as a wheel or strut) without certain corrections, known as 'scale effect.'

Two principal types of channel have been developed, the 'open' type and the 'closed circuit.' Most of those in use are of the latter type, in which the same air is caused by means of a fan to move along a closed circuit of rectangular plan form. The working portion occurs at one of the longer sides.

OPEN TYPE CHANNEL.

The open N.P.L. type channel is normally built of wood and supported on a braced steel structure, so that the horizontal centre line is well above the floor. The working portion of the channel is situated towards the end of the front portion of the channel which is fitted with a carefully shaped

intake. This part of the channel is generally square in section and about six diameters long. In order to further reduce any possible eddy motion which may have been set up at the intake, and to counteract the tendency of the fan to impart rotation to the air stream, a honeycomb is placed at each end of the channel.

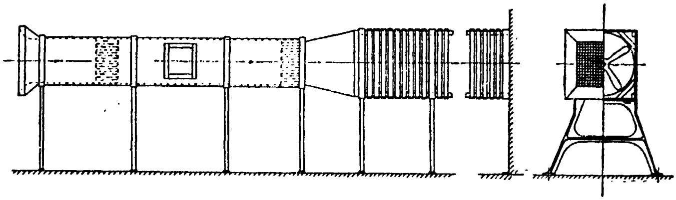


FIG. 3.—Modified N.P.L. Type, 2 ft. 6 ins. Wind Channel.

End View of Channel.

The end of the channel at the rear honeycomb is rigidly connected to a metal fan chamber, so constructed that in a distance of a few feet its section changes from that of the channel, viz. square or rectangular, to circular. The four-bladed fan is driven directly by a motor mounted on independent supports, so that no vibration from it shall reach and affect the air stream.

This type of wind channel, in which air enters at one end and is discharged through the diffuser at the other, must be installed in a large room free from obstacles which interfere with steady flow.

In the region of the working portion, glazed air-tight doors are fitted for convenient access and visibility.

CLOSED CIRCUIT CHANNEL.

This type of tunnel is one in which air is set in motion round a closed, or nearly closed, circuit. Diagrammatically, fig. 4 shows an arrangement in which a motor-driven fan is mounted near the intake and the air propelled round the tunnel to be discharged at the other end in the form of a jet. The latter is co-axial with the intake and the distance between the two ends of the tunnel forms the working portion.

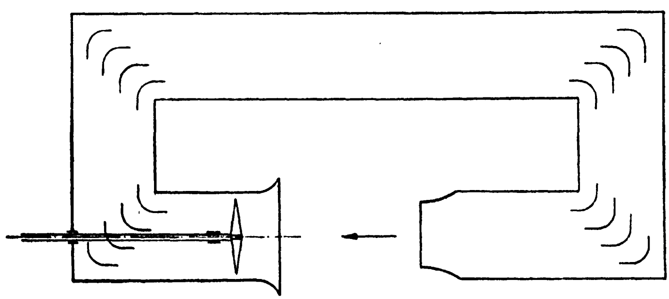


FIG. 4.

The air is helped round the four corners of the tunnel by a number of guide vanes. Power costs for this type is low since once the air is speeded up it is necessary only to maintain this *same* air in motion. Sometimes the working section is enclosed, but access to the models is not then as easy.

It has long been realised that with high-speed aircraft, special care must be taken in order to reduce the scale effect corrections which should be applied to results of model tests. The criterion being that Reynold's number $\frac{vl}{\nu}$ (where v is speed, l some linear dimension, usually the wing

chord, and ν the kinematic viscosity of the fluid medium) for the full-scale machine, and the model should be the same, suggests the possible courses :

(a) To increase the size of model and/or wind speed of test.

(b) To increase the wind speed and decrease the kinematic viscosity of the air by increasing its pressure, *i.e.* its density.

THE VARIABLE DENSITY CHANNEL.

Working on the lines indicated in (*b*), above, a wind channel was constructed in America in 1923 in which the model was tested in air at 20 atmospheres pressure. Some years later there was erected at the N.P.L. a similar channel to operate at 25 atmospheres pressure (fig. 5). Both channels were constructed in steel with hemispherical ends.

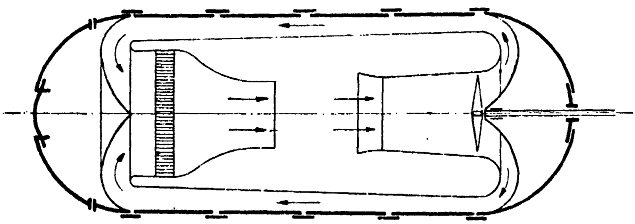


FIG. 5.

Comparative particulars are :—

	American.	N.P.L.
Jet diameter, ft. .	5	6
Pressure, atmospheres .	20	25
Wind velocity, ft./sec. .	70	90
Reynolds number	R	1.93 R
Normal h.p. of fan motor	250	400

DYNAMIC SIMILARITY AND SCALE EFFECT.

Flight tests and experiments on full-scale aircraft are an expensive matter ; in many instances the required information may be obtained from a test on a geometrically similar model in a wind channel.

It does not necessarily follow, however, that because the model and full-scale machine are geometrically similar, there will be similarity of flow of fluids past these bodies, nor similarity of motion of the latter. In other words, there may not be dynamic similarity. The whole of the utility and application of model test results to full-scale depends on the conditions of dynamic similarity being known and satisfied.

Generally the model test cannot be carried out so as to satisfy all the required conditions simultaneously, but as a rule one particular condition has a predominating effect on the motion concerned, and if this is fulfilled the model results may generally be applied to full-scale. Owing to the neglect of the secondary conditions there is frequently a discrepancy between the full-scale results as predicted from model experiment (depending, for instance, on the shape of the body concerned) and those obtained directly from full-scale flight tests. Due to this 'scale effect' a correction has to be applied to the model test figures to bring them into agreement with full-scale results.

The resistance to motion of that part of the hull of a ship or flying boat in contact with water is made up of two parts, viz. (*a*) skin friction (due to the viscosity of water), and (*b*) wave formation (due to the gravitational effect). At high speeds the resistance due to skin friction is relatively unimportant so that the wave-making resistance alone may be considered.

Froude's law of comparison demands that the corresponding speed on the water should be

$$v_m = v\sqrt{\frac{l_m}{l}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

both for model and full-scale hull.

COMPRESSIBILITY EFFECTS.

Whereas in the normal forms of wind tunnel the value of Reynold's Number (R) does not often exceed one million, it may in a variable density tunnel reach 5 or even 10 million. So that, although in actual flight at high speeds R may run as high as 20 million or more, it is commonly somewhat less: hence the variable density tunnel gives results which may in general be regarded as valid. The one serious difficulty to-day with tunnel work on models is that when the speeds become so high as to approach the velocity of sound the flow pattern changes substantially and this, with some uncertainty as to the effect of the tunnel walls, makes the interpretation of the results difficult. Wind tunnel tests are sometimes checked by wake measurements in free flight (in these tests a 'rake' of pressure tubes is used to measure the heads in the air wake behind the aircraft) and these have proved useful: but even such free flight tests, though avoiding the complication of tunnel constraints, are liable to fail when the velocity of sound is approached owing to the great spread of the air flow pattern. When the airspeed is 0·8 of the velocity of sound at any altitude, it is said to have a 'Mach Number' of 0·8, or M = 0·8. Hence in airflow at high speeds and high altitudes it is necessary to pay attention to the values of both R and M. High values of M are particularly liable to arise at altitude, because the velocity of sound falls with fall of temperature and its speed of 750 m.p.h. at sea level becomes only about 650 m.p.h. in the stratosphere (30,000 ft. and over). Nevertheless, despite the great increase in drag as values of M approach unity, modern fighter aircraft have reached such speeds.

At subsonic speeds (M less than unity), wind tunnel measurements can be relied upon to give results which, when the appropriate corrections are made, can be applied to full scale design. It is also possible to obtain useful design data from the little ultra-high speed tunnels used in artillery research in obtaining the ballistic coefficients for shells and rockets, even though the speeds are then well into the supersonic range. The difficult range of speed is the sonic (where M = 1) for it is then that the drag coefficient increases with great suddenness and the air flow pattern with it. A careful study is being made of this difficult region but far too little is at present known to enable the results of model tests to be interpreted as a reliable guide to aircraft constructors. The alternative procedure is to experiment with full-scale aircraft despite the risk to human life, or to employ pilotless aircraft with radio control. Although the full-scale experimentation has unfortunately led already to the loss of valuable lives, it has enabled the first flights into and through the 'sonic barrier,' as it is sometimes called, to be made by one or more intrepid pilots. The great engine thrust needed for the first penetration of the barrier by an aircraft was provided by boosting-rockets. Once through the barrier it is generally anticipated that the nature of the air forces to be encountered will be fairly readily predictable, with the result that the task of the aircraft designer will be eased.

It is likely that to meet military requirements, aircraft capable of flying at supersonic speeds will prove necessary. Hard as the sonic barrier may be to penetrate, it will be less difficult for small aircraft than for large ones. Had the barrier proved impenetrable, then once bombers and interceptor fighters had been pushed to the limit of their speed, there would have been no margin possessed by the interceptor to enable its task to be carried out, and the effectiveness of the defence against the bomber would have suffered gravely. As things are, however, there will not long hence be many interceptors capable of supersonic speed, whilst for some years, perhaps many years, bombers speeds will be limited to the subsonic. Hence fighters, with their margin of several hundreds of miles an hour of superior speed, will be at a great advantage, and the strength of the defence will gain greatly—much to the satisfaction of all peace-loving countries.

Aerofoils.

When the main planes of an aeroplane are considered apart from the rest of the machine they may be referred to as 'aerofoils,' and the shape of a main plane section given by a plane parallel to the plane of symmetry is spoken of as the 'aerofoil section.'

A typical section is shown in fig. 6.

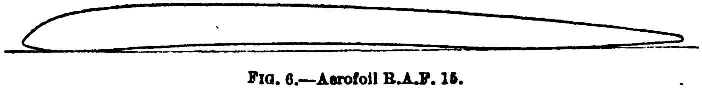


FIG. 6.—Aerofoil R.A.F. 15.

The upper and lower surfaces of an aerofoil are cambered, being arranged to meet in the front to form a nose or leading edge, while at the rear the aerofoil is finished off with a somewhat elongated trailing edge.

The chord of an aerofoil is generally defined as the length of the common tangent to the lower cambered surface cut off by projectors from the leading and trailing edges. In the case of a 'thick' aerofoil section in which the lower camber is convex like the upper, the chord may then be defined as the length of the line passing through the centre of curvature of the leading and trailing edges cut off by the extremities of the aerofoil.

Fig. 7 shows an aerofoil inclined at an angle of incidence α to the direction of the wind. The resultant force R acting on the aerofoil depends upon α, the wind speed V, and the area of the aerofoil S. Let the line of action of R cut the chord AB in C. The resultant force R may be resolved normal to and along the direction of flight giving a lift force L and a drag force D. The angle between the direction of the resultant force and the lift component is γ. The position of C is determined in the case of model experiments by the measurement of the moment M acting at the point A, the sign of M being reckoned positive when acting in a clockwise direction.

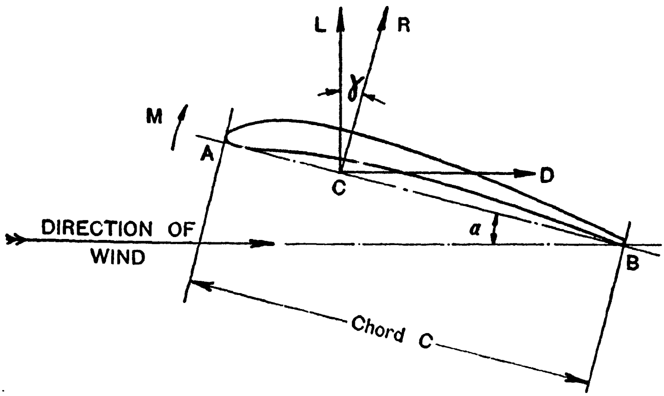


FIG. 7.—Forces Acting on an Aerofoil.

Provided that α is kept constant R, and therefore L and D, remain practically proportional to the area of the aerofoil, the square of the speed, and the density of the air. Under the same condition the ratio $\frac{AC}{AB}$ and γ remain practically constant. If ρ is the mass density * of air, the quantities $\frac{R}{\rho SV^2}$ and $\frac{AC}{AB} \cdot \gamma$ are non-dimensional and also practically independent of the linear dimensions of the model or wind speed during a given test.

The following definitions may now be made :—

$$\text{Lift coefficient } C_L = \frac{L}{\frac{1}{2}\rho SV^2} \qquad (2)$$

$$\text{Drag coefficient } C_D = \frac{D}{\frac{1}{2}\rho SV^2} \qquad (3)$$

$$\text{Moment coefficient } C_M = \frac{M}{\frac{1}{2}\rho ScV^2} \qquad (4)$$

Centre of pressure coefficient—

$$\text{or } \frac{AC}{AB} = -\frac{M}{c\,(L\cos\alpha + D\sin\alpha)} = -\frac{C_M}{C_L\cos\alpha + C_D\sin\alpha} = -\frac{C_M}{C_L} \text{ approx.} \qquad (5)$$

It is thus possible to express the aerodynamic characteristics of an aerofoil by plotting, as in figs. 8 and 9, the coefficients in equations (2) and (3) against the independent variable α, the angle of incidence. In experimental work α is given various values between $-5°$ and $+20°$, so that the whole flying range is more than covered.

* The mass density $= \frac{\text{weight density in lbs./ft.}^3}{g} = 0{\cdot}00238$ slug/ft.^{3} at ground level under standard conditions.

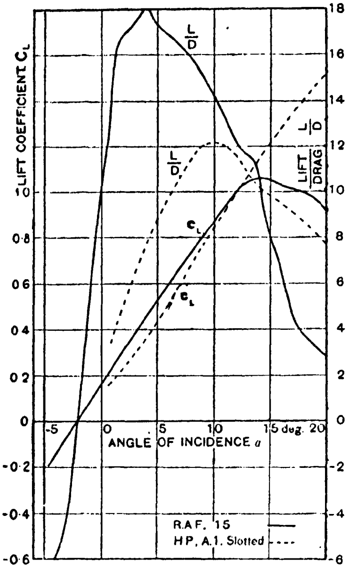


FIG. 8.

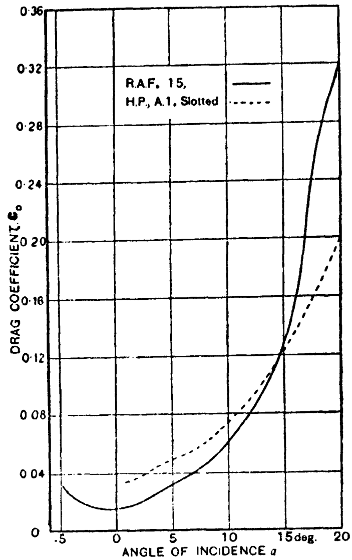


FIG. 9.

Flaps.

On aircraft with a high wing loading, it is necessary to fit some device that will limit the landing speed to a reasonable figure. The usual method is by the fitting of flaps to the aft underside surface of the wings. There are a large number of different designs of flap in use (some well-known types being shown in fig. 10), all of which depend for their action, on the large increase they give, when in the open position, to the maximum value of the lift coefficient.

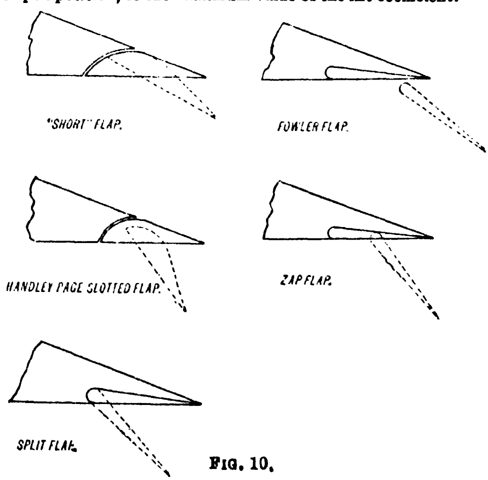


FIG. 10.

Slots may also be used to increase the maximum lift coefficient, but this involves a tall leggy undercarriage to enable the aircraft to land with the high angle of incidence required by the slotted wing.

Flaps have the further merit in increasing the drag, which for low drag machines, effects a very important reduction on the landing run.

By partially opening the flaps, the take-off may be improved by about 10–15 per cent.

EFFECT OF FLAPS.

Type of Flap.	Per cent. increase in C_D max.	Per cent. increase in C_D min.
Split flap . . .	35	14
Lap flap . . .	54	10
Fowler flap . . .	79	10

THE SLOTTED AEROFOIL.

The slotted aerofoil was produced simultaneously by Handley Page in this country, and Lachmann in Germany in 1919, both working independently. At that time the primary object was to produce an aerofoil having a high lift coefficient in order to reduce the landing speed of a given machine.

The main aerofoil, throughout its entire length, was made with an adjustable nose of streamline form called an auxiliary aerofoil, which could be moved to or away from the main aerofoil by a hand-operated link motion. With the slot closed the main aerofoil is of normal section. With the auxiliary aerofoil in its forward position the slot is said to be open.

Later it was found that good results were given if the auxiliary aerofoil were made of duralumin sheet, bent to suit the shape of the nose of the main aerofoil. Fig. 11 shows a leading edge slot of this type fully open. The main aerofoil section is the Handley-Page A.1, which is a slightly modified R.A.F. 15 shown in fig. 6, having increased camber to permit deeper spars being employed. It will be noticed that there is a rear slot provided between the back of the rear spar and the leading edge of the aileron. When the aileron is depressed the slot remains

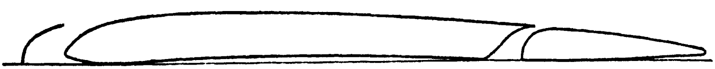


FIG. 11.—Aerofoil H.P., A.1 (Modified R.A.F. 15).

open and it is found that the effect of this is to reduce the angle of incidence required for take-off or landing from an excessive figure above 20°, to one in the region of 18°. With the front slot open and the aileron at 0°, the results of wind-channel tests (for which k_D and α have been corrected for channel restraint) are shown compared with those for a standard R.A.F. 15 section in figs. 8 and 9. It will be seen that an increase in lift coefficient of nearly 100 per cent. is obtained, but at the expense of an increased angle of incidence. The model used for these observations was 36-in. span by 6-in. chord (6·4 ins. with slot open), and the experiments were performed in a channel 4 ft. × 4 ft. at a wind speed of 44 ft./sec.* It is clear, therefore, that by the use of a leading edge slot extending the whole main plane span, a greatly increased speed range is possible.

Factors affecting the Aerodynamic Qualities of an Aerofoil.

Quite apart from its geometrical properties, the results of tests on an aerofoil lose much of their value unless certain data are specified. The model chord, aspect ratio, and wind speed should be quoted, as these quantities have been shown to have a considerable effect on the characteristics. Experiments at the N.P.L. have in the past been undertaken to ascertain the extent of the variations involved when an aerofoil of given section is tested as a monoplane.

These tests were carried out with models of the same chord, but different spans. Both C_L and $\frac{L}{D}$ were found to be affected. The maximum valve of C_L increases with the aspect ratio, as does the gradient of the curve for which C_L is plotted against α. This effect is still in evidence for aspect ratios well in excess of 10 or 12, but it is most serious for low values up to about 5, becoming less important as the aspect ratio is increased above this amount.

* *Flight*, January 28, 1926. 'Tests on Slotted Aerofoil.' F. Handley Page.

The $\frac{L}{D}$ variation is very large for values of α above 2° or 3°; below this region the effect is, however, small. For a particular model, by changing the aspect ratio from 4 to 6 it was possible to obtain a 20 per cent. increase in the value of $\frac{L}{D}$. Hence large aspect ratios appear to be desirable from an aerodynamic point of view, but as a rule structural considerations limit the aspect ratio to a figure between 5 and 6.

Distribution of Pressure over the Chord of an Aerofoil.

In order to determine the manner in which the pressure varies over the chord of an aerofoil on both upper and lower surfaces, a special model is prepared, and measurements made of the fluid pressure at various places. The model is preferably made of metal, and the surface drilled deep

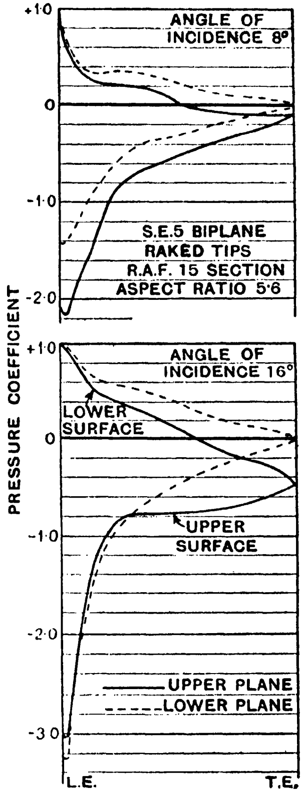


FIG. 12.—Distribution of Load over Chords of Biplane. (R. and M. 355.)

enough at the desired places, so that all the holes are connected to a common tube, which is finally connected to a Chattock gauge. The holes are filled with plasticene, and one opened at a time, so that the local pressure may be taken. It has been shown that the size of the hole is unimportant; generally it is $\frac{1}{32}$ in. diameter. In this way the distribution of pressure over any chord may be obtained. Except for a region near the tips, the pressure distribution across the chord for the whole length of plane (of constant chord) is indicated in fig. 12 for two angles of incidence. This refers

to experiments on R.A.F. 15 biplane of 6-in. chord, the results of which are given in R. and M. 355. Generally there is a suction on the upper surface and a smaller pressure on the lower surface. Both the positive and negative pressures reach their greatest values near the leading edge where the change in pressure is very rapid. At the point on the nose normal to the air stream, the pressure is known to be $\frac{1}{2}\rho V^2$ lbs. per sq. ft. In fig. 12 the coefficient of $\frac{\rho V^2}{2}$ is plotted for each point, and it is clearly shown that near the nose of the upper surface of the aerofoil the pressure rapidly falls to a local value of approximately $-3\cdot0\rho V^2$.

Distribution of Pressure over the Span of an Aerofoil.

The investigation described in the previous paragraph was extended in order to find out how the intensity of load per foot-run of span varied over the whole of the top and bottom planes. For angles of incidence of 8° and 16° the results are exhibited in fig. 13, in which, for convenience, the ordinates are coefficients which represent the normal force per foot-run of span per foot-chord divided by $\frac{\rho V^2}{2}$.

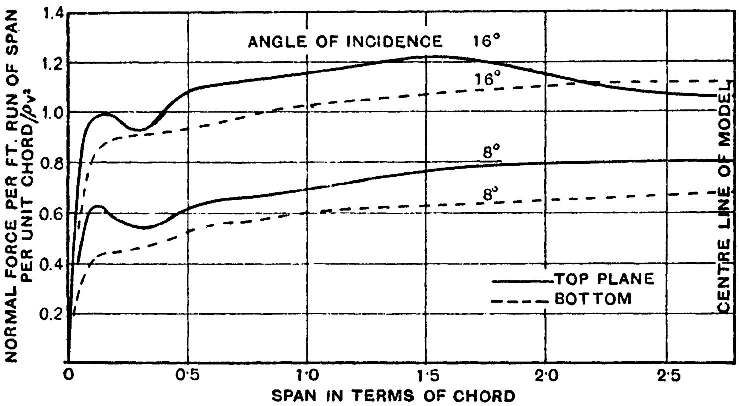


FIG. 13.—Variation of Normal Force along Span. (R. and M. 355.)

As would be expected, the normal force on the bottom plane is generally less than that for the top plane, while, with the exception of a temporary rise near the tips, there is a general falling off in loading as the tip is approached.

Low Drag Wings.

The airflow over a wing surface tends to be smooth and 'laminar' over and just behind the leading edge but to become eddying and 'turbulent' thereafter. The drag from laminar flow is much less than that from turbulent, so that any delay in the transition from the one to the other is a gain. It is hard, however, to attain in practice, and hard to preserve when attained, as the transition is affected by any waviness in form, or even by roughness due to the presence of sand or dirt on the wing surfaces. But since the drag would, in some cases, be reduced by 50 per cent. if the flow could be made laminar throughout, it will be seen that the inducement is great. Efforts have been made to preserve laminar flow by air suctions in the after part of the wing, but there is much work to be done before they can be shown to produce a good balance-sheet, in which the saving in the power required for propulsion is greater than the power used to produce the necessary suctions. It seems likely that jet-propelled aircraft will lend themselves more readily than their predecessors to this development, and if they do the speeds attained by such craft will be higher than even they are at present. Although drag coefficients rise greatly as M approaches unity, they fall again thereafter. It is only at the 'barrier' itself that they reach formidable heights. Gun projectiles and rockets have, of course, long since penetrated the barrier, but it is difficult to do so with aircraft.

The Aerofoil Theory.

The flow of air past an object is a three-dimensional one. Consequently the results in the case of an aerofoil of infinite span, for instance, would not be the same as for one of finite span.

In the case of a finite aerofoil the type of flow associated with the lift is known to be of the form shown in fig. 14. It will be seen that above the aerofoil, where the pressure is low, the

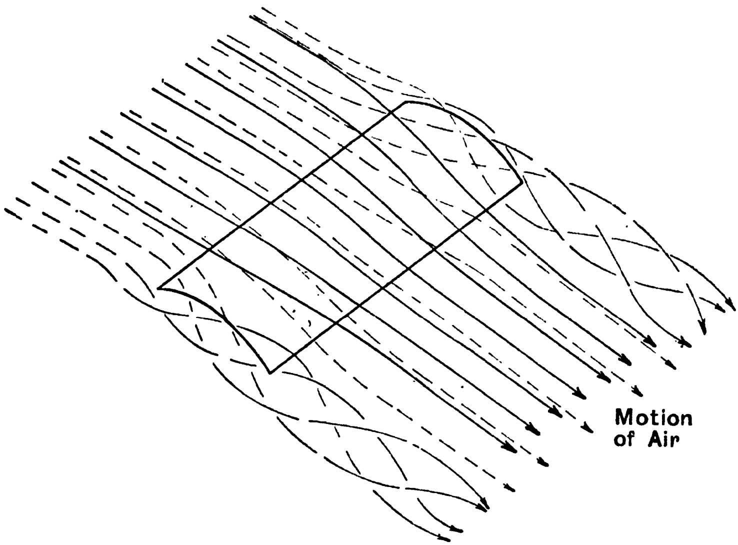


FIG. 14.

streamlines tend to converge towards the centre; while those underneath, in the high pressure region, tend to flow outwards. At places where these streamlines leave the aerofoil, trailing vortices are formed. In the conventional infinite aerofoil these trailing vortices of course do not exist. In general there will be a downward velocity or down wash of air over the whole span of the wing (fig. 15) and a corresponding upwash outside the limits of the wing tips. This

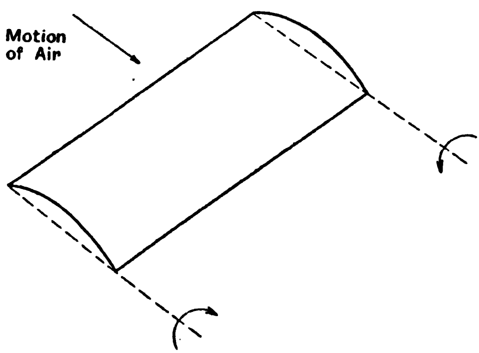


FIG 15.

downward velocity is known as the induced velocity, w, which is small compared with the air velocity V (fig. 16). The very small component induced velocity in a direction parallel to the

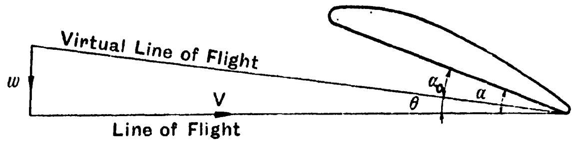


FIG. 16.

chord is neglected. The effect of the induced velocity is to reduce the angle of incidence α to an effective angle α_0 given by

$$\alpha_0 = \alpha - \theta \text{ where } \theta = \tan^{-1}\frac{w}{V}$$

Corresponding to motion along the virtual line of flight inclined at θ to the free stream (fig. 17) let C_{L_0} and C_{D_0} be the lift and drag coefficients. A component of the lift increases the drag, for

$$C_L = C_{L_0} \operatorname{Cos}\theta - C_{D_0} \operatorname{Sin}\theta$$

$$\text{and } C_D = C_{D_0} \operatorname{Cos}\theta + C_{L_0} \operatorname{Sin}\theta$$

but θ is small and $\operatorname{Sin}\theta = \theta$ radians. So that in the limit we may write

$$C_L = C_{L_0}$$

$$C_D = C_{D_0} + C_{L_0}\theta = C_{D_0} + C_L\theta$$

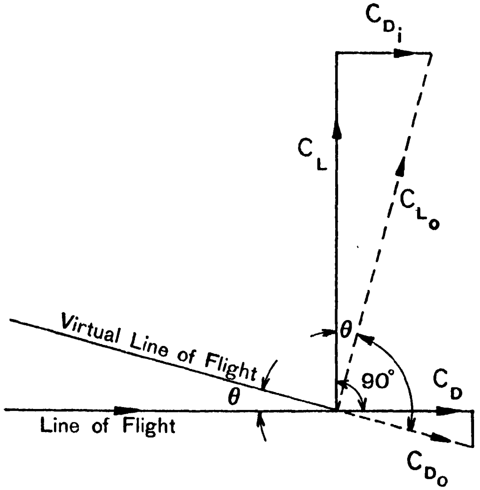


FIG. 17.

Induced Drag.—The term $C_L\,\theta$ is the induced drag coefficient C_{D_i}, and is so named since it is due to the induced velocity of the trailing vortices.

The Total Drag Coefficient **C_D for a monoplane aerofoil is the sum of the profile drag coefficient C_{D_0} and induced drag coefficient C_{D_i};**

thus: $$C_D = C_{D_0} + C_{D_i} \text{ where } C_{D_i} = C_L\,\theta_i$$

The Value of **C_{D_i}.—By making certain assumptions, including the type of aerodynamic loading, the value of the induced velocity may be calculated for a given monoplane or biplane.**

Assuming an elliptical distribution of lift over the span of a monoplane (*cf.* fig. 17) the mathematical treatment shows that w is constant at all positions along the span and that

$$\theta = \frac{C_L}{\pi A} \text{ radians}$$

where A = aspect ratio.

The relation between three dimensional and two dimensional flow for a given lift coefficient C_L may now be expressed as follows:—

$$a = a_0 + \frac{C_L}{\pi A}$$

$$C_L = C_{L_0}$$

$$C_D = C_{D_0} + \frac{C_L^2}{\pi A}$$

The latter equation shows that for a given lift coefficient C_L the induced drag depends upon the aspect ratio only. Further, the greater the aspect ratio, the smaller is the value of the induced drag, which becomes zero when A is infinite, corresponding to two dimensional flow.

By observing C_D and calculating C_{D_i} it is found that below the critical angle of incidence C_{D_0} is tolerably constant.

THE AIRSCREW.

A typical airscrew is shown in fig. 18. The shape of the blade changes continually along its length, but at all cross sections the resemblance to an aerofoil is maintained, except that near the boss it becomes more or less streamline.

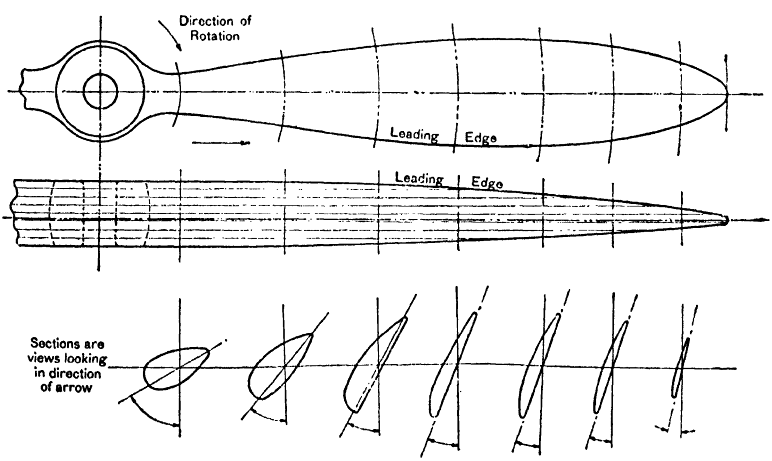


FIG. 18.—Typical Wooden Airscrew showing Laminations.

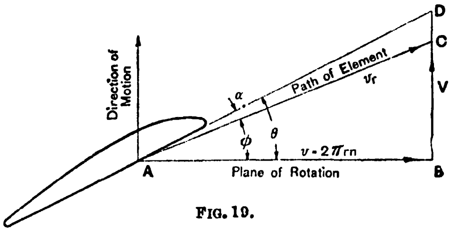


FIG. 19.

Consider an element of the blade as in fig. 19, and let

V = velocity of translation of element, *i.e.* forward velocity of aircraft;
n = rotational airscrew speed;
r = radial distance of element from axis of rotation;
a = angle of attack (or incidence) of blade element to helical path;
ϕ = angle between direction of relative wind at blade element, *i.e.*, the plane of rotation;
θ = blade angle of element, *i.e.*, the angle between chord of element and plane of rotation.

so that $\theta = \alpha + \phi$

The triangle ABC is development of helical path of element, in which AB represents linear velocity ωr or $2\pi rn$ in the plane of rotation; BC represents V; and AC, the path of the element, represents the relative wind velocity v_r.

Thus $$\tan \phi = \frac{V}{2\pi rn}, \quad \text{or,} \quad 2\pi rn \tan \phi = V.$$

Now an airscrew is designed for a definite value of V corresponding to race or climb conditions, hence in a given airscrew $2\pi rn \tan \phi$ is constant at all points along the blade, and hence $\tan \phi$ varies inversely as r, and each element describes a helix of the same pitch.

SLIP.

The analogy of a nut working on a bolt breaks down in the case of an airscrew for two reasons, viz.:

(*a*) For a given rotational engine speed the aeroplane, and therefore the airscrew, may have different translational speeds: and

(*b*) As the air passes the plane of the airscrew it receives an additional backward velocity and the airscrew is said to slip and the stream of air behind the airscrew is called the slip stream.

PITCH.

Considering the airscrew as a whole the term 'pitch' requires careful definition. Two pitches are in use.

(i) *The Geometrical Pitch.*—Suppose in a certain airscrew each blade element had no angle of attack as in fig. 20. In unit time each element moves forward V feet, but travels along its path

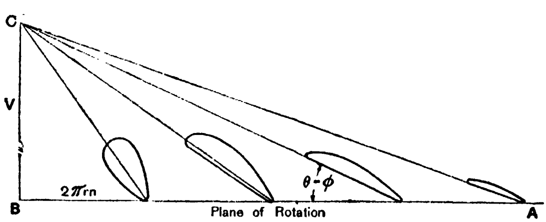


FIG. 20.

in the plane of rotation a distance $2\pi rn$. In these circumstances the advance BC of the airscrew per revolution is called the 'Geometrical Pitch' P_g and the airscrew is said to have a constant pitch.

Usually, however, airscrews are made with varying angles of attack α along the blade (fig. 19), thus the geometrical pitch of each element varies, and the airscrew cannot be said to have a geometrical pitch.

In such a case the geometrical pitch of an element situated at two-thirds radius from the axis is called the 'Geometrical Mean Pitch' for the airscrew, *i.e.*,

$$BD = P_g = 2\pi . \tfrac{1}{3}D \tan \theta = 2\cdot 1\ D \tan \theta,$$

where D = airscrew diameter.

(ii) *The Experimental Mean Pitch.*—Apart from other factors the thrust of an airscrew depends upon its rotational and translational speeds, n and V.

For any airscrew, at certain corresponding values of n and V the slip-stream entirely disappears, the air in the vicinity of the airscrew remaining practically undisturbed so that the thrust falls to zero. The value of $\frac{V}{n}$, or the advance per revolution when the airscrew thrust is zero, is called the 'Experimental Mean Pitch' P.

Under working conditions when the airscrew is developing thrust at speeds V_1 and n_1 the slip is given by

$$\frac{V_1}{n} + \text{slip} = P$$

Airscrew Characteristics.

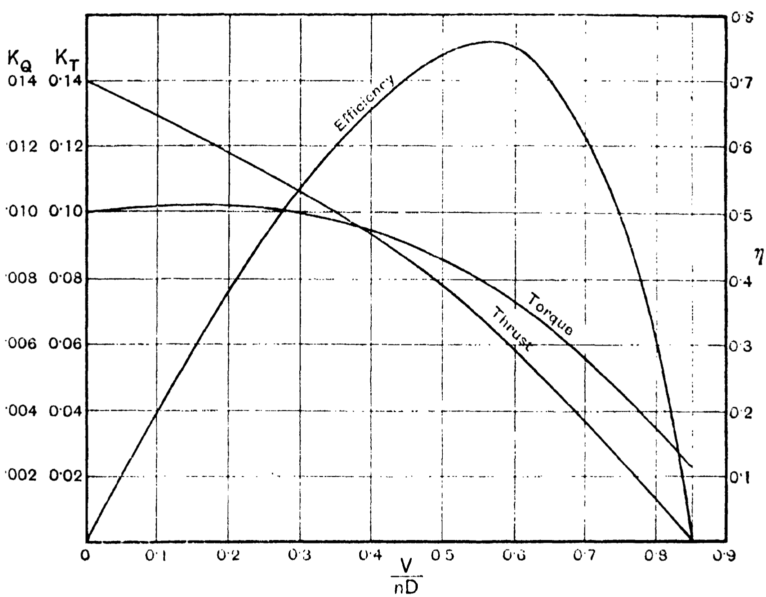


FIG. 21.—Airscrew Characteristic Curves.

The thrust T and torque Q of an airscrew are shown in Part D to depend upon ρ, V, n, and some linear dimension, generally taken to be the diameter D.

Provided that $\frac{V}{nD}$ (the advance per revolution as a fraction of the diameter) is constant, T is practically proportional to the mass density ρ, the square of the rotational speed n, and the fourth power of the airscrew diameter D.

∴ We may put

$$\text{Thrust coefficient } k_T = \frac{T}{\rho n^2 D^4} \quad . \quad . \quad . \quad . \quad . \quad (6)$$

similarly,

$$\text{Torque coefficient } k_Q = \frac{Q}{\rho n^2 D^5} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

where k_T and k_Q are both non-dimensional, varying only with the value of $\frac{V}{nD}$ provided that the tip speed does not approach the velocity of sound.

Hence, in the same way that C_L and C_D depend upon the angle of incidence α for an aerofoil, so k_T and k_Q depend upon $\frac{V}{nD}$ for an airscrew. The complete performance of an airscrew may therefore be expressed in terms of these quantities by plotting experimental results.

A typical set of characteristic curves is shown in fig. 21.

The efficiency curve is obtained from the known values of k_T and k_Q from the relation

$$\eta = \frac{1}{2\pi} \cdot \frac{V}{nD} \cdot \frac{k_T}{k_Q} \cdot = \frac{1}{2\pi} \cdot \frac{k_T}{k_Q} \cdot J. . \quad . \quad . \quad . \quad . \quad (8)$$

where

$$J = \frac{V}{nD}.$$

It is seen that the thrust is maximum when the airscrew has no translational speed V, falling off to zero at a certain value of J. This value corresponds to the experimental mean pitch P as a fraction of D.

The torque remains tolerably constant, first increasing slightly then falling off to a low value when the thrust is zero.

The efficiency is zero when $V = 0$ and also when $T = 0$.

The working range of this airscrew corresponds to values of J between 0·40 and 0·65, the maximum efficiency being reached when J is between these values. Except in special cases the value of J at top speed does not exceed unity.

Airscrew Theories.

A number of theories have been put forward by means of which expressions for thrust, torque and efficiency may be obtained. Among the most important are :—

(a) The Froude momentum theory.
(b) The application of Bernouilli's theorem
(c) The simple aerofoil theory.
(d) The inflow theory.

The Froude Momentum Theory.

In propounding this theory Froude supposed the actual airscrew replaced by an ideal one consisting of an imaginary disc in whose plane an instantaneous change of pressure occurred under working conditions.

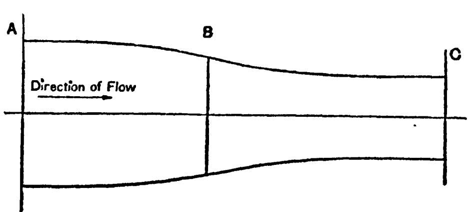


Fig. 22.

Regarding the airscrew as stationary, the ingoing and outgoing streams are indicated in fig. 22. On the upstream face of the disc there is a region of negative pressure which causes motion, and on the downstream face the pressure instantaneously changes to a positive value which increases the pressure of air in the outgoing stream or slip-stream.

Assuming frictionless streamline flow, and considering sections A and C sufficiently removed from the disc B such that the streamlines are parallel, tabulate the quantities concerned :—

Section.	Velocity.	Cross-sectional Area of Stream.	Pressure.
A	V_1	A_1	p_1
B	V_0	$A_0 = \frac{\pi}{4}D^2$	—
C	V_2	$A_2 = KA_0$	p_1

The outgoing stream contracts, making the slip stream velocity V_2 greater than V_1.

Thus

$$V_2 = V_1(1 + b), \quad \ldots \ldots \quad (9)$$

where b is the outflow factor.

Neglecting losses, it may be shown that,

$$V_0 = V_1\left(1 + \frac{b}{2}\right)$$

$$= V_1(1 + a_1) \quad \ldots \ldots \quad (10)$$

where a_1 is the inflow factor

Hence in the ideal airscrew, half the added or slip stream velocity is imparted to the air immediately in front of the airscrew.

THRUST.

Since the thrust is given by the rate of change of momentum of the air stream, it may be deduced that,

$$T = \rho KA_0(1 + b \quad bV_1^2 \quad \ldots \ldots \quad (11)$$

where T is the thrust.

EFFICIENCY.

The ideal efficiency may be given as,

$$\eta_f = \frac{\text{work got out per sec.}}{\text{work got out per sec.} + \text{added kinetic energy of slip stream}}$$

from which

$$\eta_f = \frac{1}{1 + \frac{b}{2}} = \frac{2}{2 + b} = \frac{1}{1 + a_1}$$

which is known as the ideal or Froude efficiency. Generally the actual efficiency is between 80 per cent. and 85 per cent. of this ideal value.

THE APPLICATION OF BERNOUILLI'S THEOREM.

A consideration of the instantaneous pressure difference at the plane of the airscrew leads to a result identical with equation 12, so that in the ideal airscrew the theoretical inflow factor a_1 is half the outflow factor b. In the inflow theory it is assumed that under all conditions $a_1 = \lambda b$.

Actually λ is less than ½ being more nearly ⅓.

The Bernouilli and momentum theories may be combined to give results which agree fairly well with practice, so far as thrust is concerned.

The Simple Aerofoil Theory.

Considering the airscrew as made up of independent aerofoil elements moving along helical paths forms the basis of this theory proposed by Froude and subsequently developed by Lanchester and Drzewiecki.

Axial inflow as indicated in the momentum theory is neglected, the velocity of air relative to the airscrew and immediately in front of its plane being taken as V, the translational speed of the aircraft.

FORCES ACTING ON BLADE ELEMENT.

Fig. 23 shows the forces acting on an element of blade at radius r from the axis and substituting V for U and ϕ for ϕ_1, the diagram is applicable to the present case.

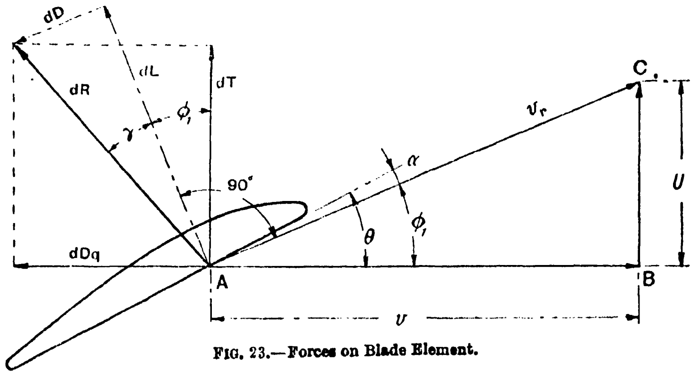


FIG. 23.—Forces on Blade Element.

We have, treating the element as an aerofoil,

dR = Resultant force on element
dL = Lift " " "
dD = Drag " " "

where,
dL and dD act normal to and along OA respectively.

The direction of dL is in advance of dR by angle γ such that

$$\gamma = \tan^{-1}\frac{C_D}{C_L}$$

dT = thrust on element, *i.e.*, the axial component of dR,
dD_q = transverse component acting at radius r from axis of rotation
dQ = torque reaction on element,

then

$$dT = dR\cos(\phi + \gamma) \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$
$$dD_q = dR\sin(\phi + \gamma)$$
$$dQ = r\,dR\sin(\phi + \gamma) \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

EFFICIENCY OF WHOLE AIRSCREW.

The efficiency of the whole airscrew is given by:

$$\eta = \frac{\text{work done by airscrew}}{\text{work done by engine}}$$

which reduces to

$$\eta = \frac{\tan\phi}{\tan(\phi + \gamma)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

On this basis, the efficiency of the element is maximum when $\phi = 45° - \frac{\gamma}{2}$.

Since $\frac{\gamma}{2}$ is small, this means that the maximum efficiency occurs when the rotational and translational speeds of the element are approximately equal.

The efficiency of the whole airscrew may be taken to be that of an element situated at a distance from the axis of about two-thirds tip radius. Thus, if an element so situated is working under the above condition, the airscrew would be working approximately at its maximum efficiency.

The Inflow Theory.

The simple theory, which was the first to be applied to design, generally gave efficiencies which were in excess of those obtained in practice. This led to the adoption of the inflow theory which allowed for the fact that the air immediately in front of the airscrew has a velocity greater than that of translation of the airscrew.

The axial inflow factor a_1 may be obtained experimentally for any given model or it may be calculated.

There is experimental evidence that in addition to axial inflow, a rotational inflow also exists, acting towards the boss. This is, however, small and is generally neglected.

Helix Angle of Blade Element.

Fig. 23 shows the forces acting on an element of blade, as already described, but in arriving at the helix angle ϕ_1, the velocity of the air passing the airscrew disc is now given by

$$U = V(1 + a_1)$$

and

$$\tan \phi_1 = \frac{U}{\omega r} = \frac{V(1 + a_1)}{2\pi r n}.$$

Efficiency of Blade Element.

The efficiency of the element now becomes

$$\eta_a = \frac{1}{1 + a_1} \cdot \frac{\tan \phi_1}{\tan(\phi_1 + \gamma)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

Thus the actual efficiency is the product of the Froude and Drzewiecki efficiencies except that U and not V is used in calculating ϕ_1.

The Total Thrust and Torque per Blade.

For any element of blade having chord c, treating the element as an aerofoil, the resultant force dR on it is $dR = c_L \rho c\, dr\, v_r^2 \sec \gamma$,

and

$$\text{Total thrust/blade } T_b = \int_0^R pc \sec \gamma \cos(\phi_1 + \gamma)\, dr \quad . \quad . \quad . \quad (16)$$

$$\text{Total torque/blade } Q_b = \int_0^R pc\, r \sec \gamma \sin(\phi_1 + \gamma)\, dr \quad . \quad . \quad (17)$$

where

p = pressure/ft.² or air loading of blade $= C_L \dfrac{\rho v_r^2}{2}$.

In an analysis, the integrals in (16) and (17) are evaluated graphically. For this purpose about six sections or elements of blade are considered, approximately corresponding in position with those shown in fig. 18 and the thrust and torque grading for each element, viz.,

$$pc \sec \gamma \cos(\phi_1 + \gamma) \text{ and } pc\, r \sec \gamma \sin(\phi_1 + \gamma),$$

calculated, assuming $\sec \gamma$ to be unity. These are plotted on a base representing the blade radius, so that the area under each curve represents T_b and Q_b respectively.

The efficiency of the whole blade is,

$$\eta_b = \frac{V}{\omega} \cdot \frac{T_b}{Q_b}.$$

The efficiency of a well-designed airscrew is in the region of 80 per cent.

Diameter of Airscrew Required.

It is generally necessary to fix the diameter of an airscrew and determine the necessary blade widths to absorb the available torque by the use of equations (16) and (17).

In order to avoid a trial and error process as far as possible, a preliminary determination of the required diameter may be obtained by formulæ proposed by Watts.* These are given below :—

Two-bladed airscrews

$$P = 1 \cdot 11 \times 10^{-10} D^4 n^3 V \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

Four-bladed airscrews

$$P = 2 \cdot 30 \times 10^{-10} D^4 n^3 V \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

where

P = h.p. absorbed ; n = r.p.m. ;
D = required diameter in ft. ; V = translational speed in m.p.h.

Note that here V is an estimated speed which will, among other things, depend on the efficiency of the airscrew when designed. In view of this there is every justification for looking upon the first airscrew for a particular case as experimental, making any necessary modifications after the initial trials have been completed.

In a particular design the diameter may be restricted to one smaller than that given by the equations above, for reasons of, say, ground clearance.

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* *Screw Propellers for Aircraft.* H. C. Watts.

Horse-Power Absorbed.

The engine power curve, supplied by the engine manufacturers, gives the power to be absorbed at stated engine revolutions. The corresponding airscrew revolutions are known from the gear ratio of the engine, and at this speed the airscrew torque must be such as to absorb the power developed.

Thus, if

b = number of blades; P = horse-power absorbed.

$$P = \frac{2\pi\, Q_b\, n.b.}{33{,}000} \qquad \qquad (20)$$

where Q_b is the torque absorbed per blade as given in equation (17).

In design, the unknown maximum blade width c is determined from equation (20) and the remaining chords at specified sections are immediately known from the plan form adopted.

Rotational Speed.

In order that it may develop maximum power, the engine, and therefore the airscrew, must be run at or near a certain rotational speed. For a given flight or translational speed, the airscrew must have its maximum efficiency when the engine is running at that rotational speed. It follows that every combination of engine and aircraft will require its own particular airscrew.

Neglecting inflow, the maximum efficiency of a blade element at radius r from the boss occurs when $\phi = 45°$ approximately.

From fig. 19, $\tan \phi = \dfrac{V}{2\pi\, rn} = 1$ when $\phi = 45°$.

This means that for the element, $\dfrac{V}{nd} = \pi$ for best results. In any case the whole blade cannot work under this condition of maximum efficiency, and in practice it is not possible to cause the airscrew to rotate slow enough to give a value of π for the pitch diameter ratio.

On the other hand, in order to obtain a light airscrew for a given diameter, the blade widths must be small and the rotational speed high.

For conditions of maximum efficiency, airscrews operate, as a rule, at a value of J between 0·6 and unity.

The question is one of compromise, but, other things being equal, a slow-running airscrew has a greater efficiency than a high-speed one. It is also much less noisy.

Number of Blades.

Airscrews are made with either two, three, or four blades.

The choice of the number of blades usually rests with the designer. The advantages of two- and four-bladed airscrews are given below :—

Two-bladed airscrew—
- Weighs very much less than the corresponding four-bladed airscrew.
- Easy to transport.
- Easier to manufacture than four-bladed airscrew.
- Stronger at boss " " "
- Cheaper " " "

Four-bladed airscrew—
- Smaller diameter than corresponding two-bladed airscrew, which may be an advantage when clearances are important. Better balancing effect so that engine tends to run better and without vibration. Generally somewhat more efficient than two-bladed airscrew.

Manufacture.

Airscrews are made in either wood or metal, the materials being American walnut, Honduras mahogany, duralumin, or steel.

A very great disadvantage of wooden airscrews is that in tropical countries the blade warps and consequently its efficiency seriously falls.

Metal airscrews do not give this trouble.

Metal Airscrews.

Metal airscrews are manufactured from light alloy (generally duralumin) or mild steel.

There are very definite advantages in the use of metal airscrews, among which are :—

(a) Damage to blades by rain and hail is very rare.

(b) They withstand large and rapid changes in temperature and variations in atmospheric conditions generally, whereas the blades of wooden airscrews twist in tropical countries, causing a falling off in efficiency.

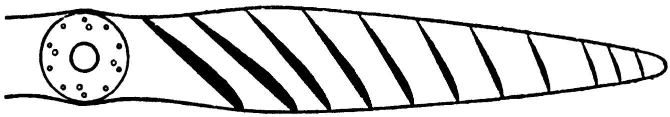


FIG. 21.

(c) As the blade tip sections can be made quite thin, as indicated in fig. 21, very high tip speeds may be employed, such as occur in high speed or racing aircraft.

ADVANTAGES.

Particular advantages of this type of airscrew are :—

(a) After preliminary trials the blades may be adjusted to suit the aircraft and engine speeds. The blades may also be altered to suit any subsequent change in these speeds.

(b) The single forging avoids joints in the blade just where a joint is least desired. The extra complication of blade attachment fittings are also avoided.

(c) In cases of accident these airscrews bend where wooden ones would break. Damaged duralumin airscrews are frequently corrected and put into service again.

(d) Bent blades may be straightened up to 5° without thermal treatment.

BLADE STRESSES.

Cases have been known of airscrews working satisfactorily at a stress of 27,500 lbs./in.², corresponding with a factor of safety of 2.

The Variable Pitch Airscrew.

The usual design condition for a fixed-pitch airscrew is for maximum efficiency at top speed and maximum horse-power. This condition, however, is not consistent with high efficiency during take-off and climb for the following reasons. The pitch of an airscrew designed for top speed is such that the angle between the incidence wind (the vector sum of the rotational velocity of the airscrew and the forward speed of the aircraft) and the nose-tail line of the airscrew blade section is that corresponding to maximum efficiency of the section. During climb and take-off, when the forward speed of the aircraft is considerably less than that at top speed, this angle will be greater than that required for maximum efficiency and may be so great as to stall the section. Moreover the torque coefficient for a given pitch increases with decreasing forward speed, thereby preventing the engine developing its full horse-power by reducing the engine r.p.m.

The Variable Pitch Airscrew surmounts this difficulty by enabling the pitch or inclination of the blades to be reduced for take-off and climb.

There are two main classes of Variable Pitch Airscrews :—

(1) Those whose pitch can be adjusted to fine pitch only on the ground (automatically taking up the coarse or top speed pitch in the air). These are called Adjustable Pitch Airscrews.

(2) The true Variable Pitch Airscrew, whose pitch can be adjusted in the air. The usual methods for adjustment are one or combinations of the following.

(a) Automatic control.—Usually brought about by the engine r.p.m. or the aerodynamic forces on the blades.

(b) Controlled pitch.—This may be achieved by some mechanism operated manually by the pilot, an electric motor, hydraulic pressure or by centrifugal forces.

Pitch variation is sometimes so considerable in range as to enable the pitch to be set at a negative angle, in which case the airscrew can be usefully employed as a brake when landing ; though as there is then no slip-stream effect on the tail surfaces much greater attention has to be paid by the pilot to directional control.

Contra-rotation airscrews are sometimes paired on a common axis; there are then no troublesome gyroscopic couples on turning to port or starboard, and this greatly assists the pilots control during manœuvre. They also remove any overturning tendency when opening the engine throttle on starting.

Thrust and Torque of Airscrews.

Expressions for the thrust and torque of an airscrew as given below show that its performance depends upon the value of J $\left(i.e. \frac{V}{nD}\right)$ at which it is working.

Consider first the case of airscrews whose tips speeds are less than 800 ft./sec. Then the effect of compressibility of the air may be neglected, although it is then on the point of becoming important.

STATIC AIRSCREW.

Consider a static airscrew, *i.e.*, one which rotates in a fixed plane and has no forward motion.

If the thrust T depends only upon airscrew diameter D, rotational speed n, mass density ρ, then

$$T = k_T \rho n^2 D^4 \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

where k_T is a constant called the thrust coefficient.

Similarly the torque,

$$Q = k_Q \rho n^2 D^5 \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

where k_Q is a constant called the torque coefficient.

MOVING AIRSCREW.

Let the airscrew be moving with translational velocity V. If the thrust now depends only upon V, D, n, and p,

$$T = \rho n^2 D^4 f_1\left(\frac{V}{nD}\right)$$

or

$$T = \rho n^2 D^4 f_1(J) \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

where J is non-dimensional.

Similarly

$$Q = \rho n^2 D^5 f_2(J) \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

Equations (23) and (24) show both T and Q to be dependent on J. In a given airscrew, D is fixed so that T and Q depend upon $\frac{V}{n}$. If then $\frac{V}{n}$ be kept constant $f_1(J)$ and $f_2(J)$ are also constant, so that we have

$$f_1(J) = k_T = \frac{\text{Thrust}}{\rho n^2 D^4} \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

$$f_2(J) = k_Q = \frac{\text{Torque}}{\rho n^2 D^5} \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

HIGH-SPEED AIRSCREWS.

From experiments on high-speed *static* airscrews, it appears that when the tip speed of such approaches the velocity of sound (in air) the airflow undergoes changes, until, if the tip speed be increased, the outflow ceases, and the slip-stream disappears when the velocity of sound is reached; the airscrew then appears to operate as a centrifugal fan. With high-speed airscrews this is serious for the phenomenon is associated with a falling off of the thrust.

It seems reasonable to suppose that at such high speeds the compressibility of the air plays an important part.

Considering compressibility effects only, if v_s is the velocity of sound in air, the resistance to motion takes the form

$$R = \rho l^2 v^2 f\left(\frac{v}{v_s}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

and the effect of compressibility depends upon the velocity v as a fraction of the velocity of sound v_s in the undisturbed fluid. The relative importance of this may be obtained by applying Bernouilli's equation to a compressible fluid and assuming adiabatic transformation.

Let δp = increase in pressure at a stagnation point in the body.
v = velocity of undisturbed stream.
ρ = mass density where speed is v.

Then, if $\gamma = \frac{7}{5}$,

$$\delta p = \tfrac{1}{2}\rho v^2 \left[1 + \tfrac{1}{4}\left(\frac{v}{v_s}\right)^2 + \tfrac{1}{40}\left(\frac{v}{v_s}\right)^4 + \ . \ . \ .\right] = \tfrac{1}{2}\lambda\rho v^2. \quad . \quad . \quad . \quad (28)$$

Since the series in equation (38) rapidly converges, the effect of compressibility may be neglected so long as $\tfrac{1}{4}\left(\frac{v}{v_s}\right)^2$ is small compared with unity.

The following table gives the approximate values of λ for the whole range of values of V from zero to v_s.

V *ft./sec.*	0	200	400	600	800	1,000
λ	1·000	1·008	1·033	1·077	1·149	1·224

When V = 220 ft./sec. (150 m.p.h.), λ = 1·01, so that the neglect of compressibility at this speed would introduce an error of 1 per cent. On the other hand, it is clear that for velocities of 800 ft./sec. and over the effect of compressibility can no longer be disregarded.

PERFORMANCE.

In the estimation of aeroplane performance, two important quantities are involved, viz. horse-power required, and horse-power available. Both of these are usually required for all attitudes of the aeroplane over the flying range for a stated series of altitudes.

On the one hand, the horse-power required is calculated from the known or estimated drag coefficients of the various components and the aerodynamic characteristics of the main planes; on the other hand the horse-power available is a quantity which is not obtained without considerable calculation. Apart from the characteristics of the airscrew itself, its performance also depends on the particular combination of engine and aeroplane.

Account has to be taken of (*a*) the variation of engine power consequent upon the decrease in density, pressure, and temperature, with altitude; (*b*) the changing value of J under which the airscrew operates with varying attitudes of flight; (*c*) the continual change of atmospheric conditions from day to day and place to place.

The Standard Atmosphere.

In order to afford a basis of comparison of full scale tests, an International Standard Atmosphere has been defined * which corresponds with the average conditions in Western Europe.

It is assumed that:

(i) The air is dry and its chemical composition the same at all altitudes, by volume being 78·03 per cent. nitrogen, 20·99 per cent. oxygen, 0·94 per cent. argon, and 0·04 per cent. carbon dioxide.

(ii) The temperature at mean sea-level is 15° C., and the barometric height reduced to 0° C. is 760 mm.

(iii) The weight of air under these conditions is 1·2257 kg./m.^{3} (0·07656 lbs./ft.^{3}). The value of 'g' is taken to be uniform at 980·62 cms./sec.^{2} (32·17 ft./sec.^{2}). The mass density ρ_0 of air at mean sea-level is therefore 0·00238 slug/ft.^{3}

(iv) For any altitude Z metres above mean sea-level up to a limit of 11,000 metres, *i.e.* to the bottom of the Stratosphere, the temperature (°C.) varies thus:—

$$\theta_z = 15 - 0\cdot0065Z \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

(v) Above 11,000 metres θ is constant and equal to $-$ 56·5° C.

(vi) It follows that for any altitude Z less than 11,000 metres, where the barometric pressure is p_z and specific weight a_z and specific mass ρ_z, these quantities are connected by the equations:—

$$\frac{p_z}{p_0} = \left(\frac{T_z}{T_0}\right)^{5\cdot256} = \left(\frac{288 - 0\cdot0065Z}{288}\right)^{5\cdot256} \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

and

$$\frac{\rho}{\rho_0} = \frac{a_z}{a_0} = \left(\frac{T_z}{T_0}\right)^{4\cdot256} = \left(\frac{288 - 0\cdot0065Z}{288}\right)^{4\cdot256} \quad . \quad . \quad . \quad . \quad . \quad (31)$$

(vii) For altitudes above 11,000 metres the relations are:—

$$\log_{10}\frac{p_1}{p_z} = \log_{10}\frac{\rho_1}{\rho_z} = \log_{10}\frac{a_1}{a_z} = \frac{Z - 11,000}{14,600} \quad . \quad . \quad . \quad . \quad (32)$$

where suffix 1 refers to conditions at Z = 11,000 metres.

Table I. shows the approximate values of relative density and pressure and temperature for various heights as given by equations (30) and (31). For convenience, altitudes in feet have been tabulated.

* Air Publication 1173.

TABLE I.

International Standard Atmosphere.

Height, feet.	Relative Density $\frac{\rho}{\rho_0} = (\sigma)$	Relative Pressure $\frac{p}{p_0}$	Temperature. °C.
0	1·00	1·00	15·0
1,000	0·98	0·97	12·5
2,000	0·94	0·93	11·0
3,000	0·92	0·90	8·5
4,000	0·88	0·86	7·0
5,000	0·86	0·83	5·0
10,000	0·74	0·69	− 5·0
20,000	0·53	0·46	25·0
30,000	0·38	0·30	− 11·5
40,000	0·25	0·18	− 55·0
50,000	0·15	0·11	− 55·0
60,000	0·09	0·07	− 55·0
70,000	0·05	0·04	− 55·0
80,000	0·03	0·025	− 55·0

The Variation of Piston Engine Power with Altitude.

It has been found by H. L. Stevens (R. and M. 960), that the variation in power is approximately proportional to $\left(\frac{p}{p_0}\right)^{1 \cdot 05}$.

For a given r.p.m., therefore, the b.h.p. in a standard atmosphere at mean sea-level (generally referred to as ground-level) must be multiplied by a horse-power factor $f(h)$ in order to obtain the corresponding b.h.p. at altitude h.

In symbolic form

$$(\text{b.h.p.})_h = \text{Std. b.h.p.} \times f(h) \quad . \quad . \quad . \quad . \quad (33)$$

the r.p.m. being the same in each case;
where

$$f(h) = \left(\frac{p}{p_0}\right)^{1 \cdot 05}. \quad . \quad . \quad . \quad . \quad (34)$$

Horse-Power Required.

The amount of labour involved in the calculation of the horse-power required depends, as in the case of the horse-power available, upon the data available and the degree of accuracy required. The value of the result depends upon a fairly accurate knowledge of the resistance of the several parts or components as deduced from model tests. Fortunately, for most components the resistance is proportional to the square of the velocity over a very large range, but in the case of elongated aerodynamic bodies such as a fuselage or flying-boat hull, where skin friction effects are important, the index of V is less than two. Unless due care is taken serious errors in the estimated resistance may arise. It is found, however, that if the resistance of such bodies is known for a speed approximately mid-way over the flying range of speeds, the error at the extreme speeds in applying the V_1^2 law is small.

RESISTANCES OF COMPONENTS.

The following list gives the resistances of various details or components at 100 m.p.h. at ground-level which may be used in the absence of other data. In all cases the resistance may be assumed to vary as the square of the speed.

Fuselage (good streamline) when $\alpha = 0°$	**2·6 lbs./sq. ft., projected area.**
„ (two cockpits and nose radiator) wheren $\alpha = 0°$	**8·0 „ „ „ „**
Tail plane and elevators	**0·75 lbs./sq. ft. of plan area.**
Fins and rudders	**0·60 lbs./sq. ft. of side area.**
Wheel, 800 × 150 mm.	**13·5 lbs. with shield to tyre.**
Struts, fineness ratios 2·5 to 5·0	**2·3 lbs./sq. ft. projected area.**

Add 0·4 lbs. for each end.

SIMPLE METHOD OF CALCULATION.

In an approximate method of calculating the horse-power required ($HP._R$) the following assumptions are usually made :—

(i) The tail load may be ignored.

(ii) The variation in resistance of components due to changes of α may be neglected.

(iii) The line of thrust passes through the C.G.

(iv) The obliquity of the line of thrust may be neglected.

(v) The airscrew slip-stream diameter is $0 \cdot 8$ D.

(vi) The resistance of parts in the slip-stream is increased by 25 per cent. for any speed of flight (i.e., the slip-stream velocity is always $\sqrt{1 \cdot 25}\, V$).

In view of condition (iii) the method outlined below cannot be applied to flying boats without modification.

The total resistance is composed of two parts (*a*) that due to main planes; (*b*) that due to remainder.

For the former,
in steady flight,

$$L = W \quad \text{. . . . .} \quad (35)$$

$$D = \frac{W}{L/D} = \frac{W}{\delta} \quad \text{. . . . .} \quad (36)$$

Let V_1 = flight speed in m.p.h.

then $V_{m.p.h.} = 1 \cdot 465\, V_{ft./sec.}$ and $(V_{m.p.h.})^2 = 2 \cdot 15\, (V_{ft./sec.})^2$

and
$$V_1^2 = \frac{W}{2 \cdot 15\, C_L \frac{\rho S}{2}} \quad \text{. . . . .} \quad (37)$$

For the latter,
if R_1 = resistance of parts outside slip-stream at 100 m.p.h.
R_2 = „ „ inside „ „ „ „
R = total resistance at 100 m.p.h.

then
$$R = R_1 + 1 \cdot 25\, R_2 \quad \text{. . . . .} \quad (38)$$

For a single-engined tractor, R_2 includes the fuselage, part of tail unit and undercarriage. Hence, using equations (36) and (38).

At any speed V_2, the total drag of the aeroplane is given by

$$D_W + D_B = \frac{W}{\delta} + R\left(\frac{V_1}{100}\right)^2 \quad \text{. . . . . .} \quad (39)$$

But for steady flight, using equation (3),

$$T = \frac{W}{\delta} + \frac{R}{10^4}\left(\frac{W}{2 \cdot 15 C_L \frac{\rho S}{2}}\right) = \frac{W}{\delta} + D^1 \quad \text{. . . . . .} \quad (40)$$

Finally,
$$H.P._R = \frac{V_1 T}{375} \quad \text{. . . . . . .} \quad (41)$$

Draw up Table II. for equal increments of C_L up to $C_{L\,max.}$ and plot the results as in fig. 25.

TABLE II.—ESTIMATION OF H.P. REQUIRED.

C_L	$\delta = \frac{L}{D}$	$\frac{W}{\delta}$	D^1	T	V_1	$H.P._R$

Deductions from Performance Curves.

Flight at Altitude.

In order to obtain the horse-power required $H.P._R$ for altitudes other than ground-level it is not necessary to go over a similar set of calculations.

If $H.P._R$ and V refer to ground conditions, then at altitude h where the relative density is σ the corresponding powers $H.P^1._R$ and speeds V^1 are given by

$$H.P.^1_R = \frac{H.P._R}{\sqrt{\sigma}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (42)$$

$$V^1 = \frac{V}{\sqrt{\sigma}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

Thus all the powers and speeds in Table II. are divided by the appropriate $\sqrt{\sigma}$ to obtain the corresponding figures at any desired altitude.

Range of Flight Speeds.

In fig. 25, the curves of $H.P._A$ and $H.P._R$ both relate to the same height, viz., ground-level. The extreme flight speeds are given at the points of intersection of these curves. At A the stalling speed is obtained whilst at B the maximum flight speed is revealed.

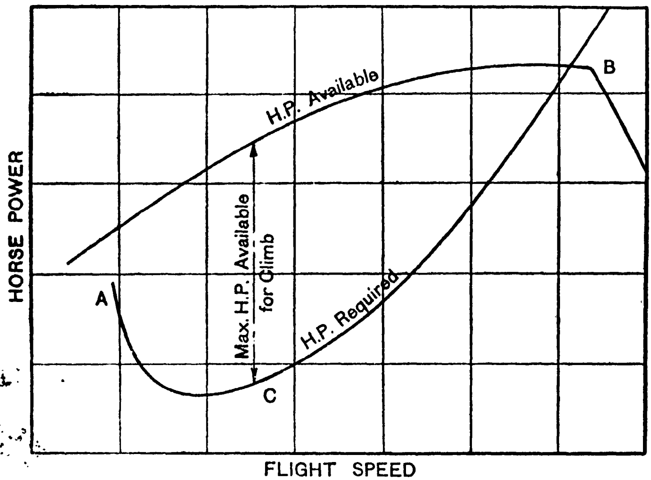


Fig. 25.—Aeroplane Performance Curves.

Rate of Climb.

The vertical intercept between the two curves corresponding with a certain flight speed V, represents the H.P. available for climb. The best climbing speed occurs when the forward speed is adjusted (by operating the elevator) to that at C where the h.p. available for climb is maximum.

It will be seen that the rate of climb r (or vertical velocity of ascent) will vary with the forward speed, in fact

$$r = \frac{(H.P._A - H.P._R)\,33{,}000}{W} \text{ ft./min.} \quad . \quad . \quad (44)$$

At the extreme speeds r is zero.

DESIGN.

Materials.

In Table III. is a list giving the mechanical properties of most of the materials used on aircraft construction.

TABLE III.

	Spec.	Description.	Tensile.	Comp.	Shear.	Bearing.	Ex. 10^{-6}	Gx. 10^{-6}
Bar.	B.6	Brass Naval	45,000	40,000	30,000	50,000	15·0	5·1
	L.1	Dural (up to 2½″ dia.)	45,000	34,000	30,000	67,000	10·5	4·2
	”	” (2½″ to 4″ dia.)	36,000	27,000	27,000	60,000	10·5	4·2
	”	” (over 4″ dia.)	30,000	22,000	22,000	54,000	10·5	4·2
	L.32	Alum	22,000	18,000	16,000	50,000	9·6	4·0
	D.T.D. 110	Dural for Rivets	45,000	34,000	34,000	67,000	10·5	4·2
	S.1	Med. Carbon Steel	67,000	51,000	51,000	100,000	30·0	13·0
	S.2	H.T. ” ”	123,000	95,000	77,000	150,000	30·0	13·0
	S.21	Mild Steel	50,000	40,000	40,000	100,000	30·0	13·0
	S.21	” ” Welded	38,000	38,000	30,000	100,000	30·0	13·0
	S.62	” ” Stainless	102,000	76,000	60,000	150,000	—	—
	—	Graphited Bronze Oilless Bushes	—	—	—	500 A.M. 1,600 H.P. 10,000 test	—	—
	S.6	40 Carbon Steel normalised	—	—	—	—	—	—
	S.11	H.T.S.	123,000	—	—	—	—	—
Forging.	S.6	Med. Carbon Steel	67,000	51,000	51,000	100,000	30·0	13·0
	S.11	H.T. ” ”	123,000	95,000	77,000	150,000	30·0	13·0
	D.T.D. 18A	Dural	45,000 36,000	34,000 27,000	27,000 27,000	67,000	10·5	4·2
	D.T.D. 88	Magnesium	25,000	25,000	18,000	50,000	6·5	—
	D.T.D. 130	Alum Alloy	—	—	—	—	—	—
	D.T.D. 280	—	—	—	—	—	—	—
Casting.	B.2	Gun Metal	30,000	30,000	24,000	50,000	—	—
	B.B.	PH. Bronze Brg.	40,000	40,000	30,000	50,000	—	—
	D.T.D. 25	Al Silicon	22,000	18,000	16,000	34,000	—	—
	D.T.D. 59	Magnesium	13,500	18,000	16,000	34,000	6·2	—
	L.5	Alum	22,000	18,000	16,000	34,000	9·6	4·0
Sheet.	B.4	Copper (Annealed)	30,000	30,000	24,000	50,000	18·0	6·4
	B.15	” (Half Hard)	30,000	30,000	24,000	50,000	18·0	6·4
	B.16	Brass (” ”)	40,000	40,000	30,000	50,000	15·0	5·1
	L.3	Dural	45 000	34,000	30,000	67,000	10·5	4·2
	L.4 L.16	Alum (Hard) ” (Half Hard)	20,000	17,000	12 000	34,000	9·6	3·87
	D.T.D. 50	Alum Silicon	25,000	20,000	16,000	34,000	—	—
	D.T.D. 111	Alclad	45,000	34,000	27,000	67,000	10·5	4·2
	D.T.D. 118	Magnesium (Weldable soft)	20,000	25,000	18.000	34,000	6·0	—
	S.3	Mild Steel	58,000	45,000	40,000	100,000	30·0	13·0
	—	” ” (where welded or heat treated)	38.000	38,000	32,000	100,000	30·0	13·0
	S.20	Tinned Steel	45,000	40,000	30.000	100,000	30·0	13·0
	D.T.D. 12A	Low Carbon Steel	45,000	40,000	30,000	100,000	30·0	13·0
	D.T.D. 57	Stainless Steel	120,000	95,000	77,000	150,000	30·0	13·0
	S.4	Nickel Steel	108,000	80,000	65,000	150,000	30·0	13·0
	D.T.D. 54	” ”	145,000	—	—	—	—	—
	D.T.D. 275	Alclad	—	—	—	—	—	—

TABLE III. (*continued*).

	Spec.	Description.	Tensile.	Comp.	Shear.	Bearing.	Ex. 10⁻⁶	Gx. 10⁻⁶
Tube.	D.T.D. 108	Copper	30,000	30,000	24,000	50,000	18·0	6·4
	T.2	Axle	190,000	145,000	104,000	200,000	30·0	13·0
	T.4	Dural	45,000	34,000	30,000	67,000	10·5	4·2
	T.50	H. Tensile . . .	102,000	87,000	60,000	150,000	30·0	13·0
	T.7	Copper. See D.T.D. 108 .	30,000	30,000	24,000	50,000	18·0	6·4
	T.8	Brass	52,000	40,000	30,000	50,000	18·0	5·1
	T.9	Alum	20,000	17,000	12,000	34,000	9·6	4·0
	T.26	Mild Steel . . .	58,000	45,000	40,000	100,000	30·0	13·0
	D.T.D. 89	Steel	102,000	87,000	60,000	150,000	30·0	13·0
	D.T.D. 89	,, Welded . . .	67,000	55,000	50,000	100,000	30·0	13·0
	D.T.D. 113	,, ...	78,000	67,000	60,000	150,000	30·0	13·0
	D.T.D. 113	,, Welded . .	67,000	65,000	50,000	100,000	30·0	13·0

STEEL.

This may be used in the form of bar, forging, casting, seamless tubing or sheet. The number of alloy steels in use on aircraft construction is very extensive and in the choice of the correct specification for any particular duty considerable experience is necessary. For instance, where the steel is to be welded one of the special welding steels may be used with advantage and where parts are liable to be exposed to corrosion stainless steel is advisable.

A fitting designed to transmit large loads should always be made from high tensile steel to keep the weight of the fitting down. Whether steel or duralumin is used in the construction of the primary structure of an aeroplane is usually a matter of preference, though it is usual for a tubular frame to be made from high tensile steel and a stressed skin structure or monocoque from duralumin.

DURALUMIN.

This is an aluminium alloy and in strength is the equal of mild steel though more susceptible to corrosion.

To minimise this, machined fittings in duralumin should be anodically treated. Fittings made from flat sheet should employ Alclad which is duralumin sheet protected on either face by a thin layer of aluminium.

In the use of duralumin, continuous care has to be taken to ensure that its remarkable properties are not impaired by bad handling and treatment.

MAGNESIUM ALLOY CASTINGS.

This metal has already established itself as a very economical material (from the weight point of view) for non-structural parts such as instrument mountings, subsidiary control mountings, etc.

WELDING.

The advantages of welding over riveting are briefly :—

1. Quickness.
2. Cheapness.
3. Ease of repairs.
4. Simpler design.
5 Structure more homogeneous.

The two main methods of welding are :—

(*a*) The oxy-acetylene welding process.
(*b*) The electrical resistance process of spot, seam and arc welding.

Oxy-Acetylene Welding.

Where heat-treatment after welding is impossible (local reheat-treatment being prohibited) it is advisable to use the low carbon, the carbon manganese and the chromium molybdenum steels, using a welding wire to DTD82A. Stainless steels are difficult to weld satisfactorily. Aluminium sheet (Spec. L. 4) as used for cowling, fairing and tanks can be easily welded using a flux to Spec. DTD119 and a welding wire of the same material.

In the gas welding of anodically treated parts (as in repair work) the protective coating should be thoroughly removed before welding.

In welding aluminium casting a welding wire of aluminium with 8–9 per cent. silicon is advised.

The welding of duralumin has not yet become a satisfactory process.

Magnesium alloys can be welded with a flux and wire similar to those used for aluminium.

Electric Seam Welding.

In this process a continuous but fluctuating current of low voltage forms a series of spot welds. It has been successfully applied to the welding together of high tensile steel strip (Spec. 54A) and proved itself at least equal to a riveted joint.

Spot Welding.

This process consists in passing through the sheets to be welded together a low voltage current, The water-cooled electrodes apply mechanical pressure during the passage of the current.

Strength of Spot Welds.

Material: M.S. Strength per spot weld in lbs.

Gauge.	Single Shear.	Double Shear.
22	1,100	1,600
20	1,600	2,300
18	1,900	2,750
16	2,400	3,400
14	3,400	4,200
12	3,900	5,600
10	5,800	8,400

Material: Light Alloy. Strength per spot weld in lbs.

Gauge.	Single Shear.	Double Shear.
26	170	310
24	240	390
22	340	530
20	420	950
18	460	950

BENDING OF ALUMINIUM ALLOY SHEET.

The following bend radii should be adhered to:—

Condition.	Inside Bend Radii.	
	22 G. and Less.	More than 22 G.
Fully annealed	$\frac{1}{2}t$	$1t$
Finally heat-treated within one hour after quenching . .	$1\frac{1}{2}t$	$2t$
Finally heat-treated and aged .	$2\frac{1}{2}t$	$3t$

CASTINGS.

Should not be used where failure of any one casting could cause the structure to fail or the control of the aircraft to be lost.

Castings used in primary parts of the structure should be radiologically examined or, if that is not practicable, proof-loaded.

RIVETS.

Some of the many types of rivets used in aircraft construction are shown in fig. 26.

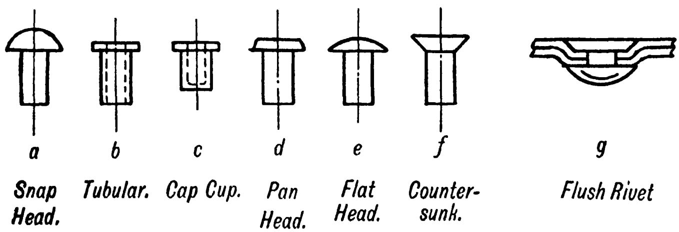


FIG. 26.

A riveted joint may fail in any of the following ways :—

(1) Failure in shear of rivet.
(2) ,, bearing of rivet.
(3) ,, tension of plate.
(4) ,, bearing of plate.

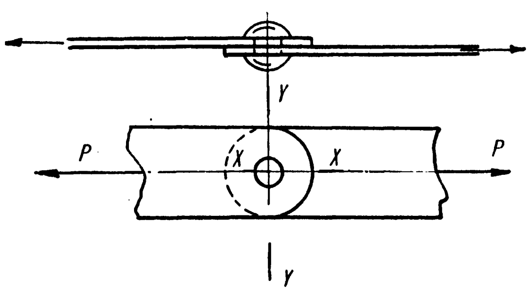


FIG. 27.

The strength of the riveted joint shown in fig. 27 is the least of the values given by the following formula :—

f_t = tensile strength of plate in lbs./in.
f_B = bearing ,, ,, ,, ,,
f_s = shear ,, rivet ,, ,,
f_B' = bearing ,, ,, ,, ,,

Bursting across XX	$T = \frac{5}{4} at f_t$ lbs. (empirical).
Tearing „ YY	$T = 2bt f_t$ „
Bearing in plate	$T = dt f_B$ „
Shear of rivet	$T = \left(\frac{\pi d^2}{4}\right) f_s$ lbs.
Bearing in rivet	$T = dt f_B'$ lbs.

GROUPS OF RIVETS.

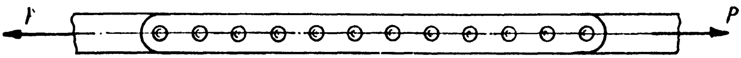


FIG. 28.

When two strips are connected together as shown in fig. 28 by a long row of rivets, it is important to remember that the loads on the end rivets are greater than the loads on the centre rivets due to the stretch in the plates.

GROUPS OF RIVETS ECCENTRICALLY LOADED.

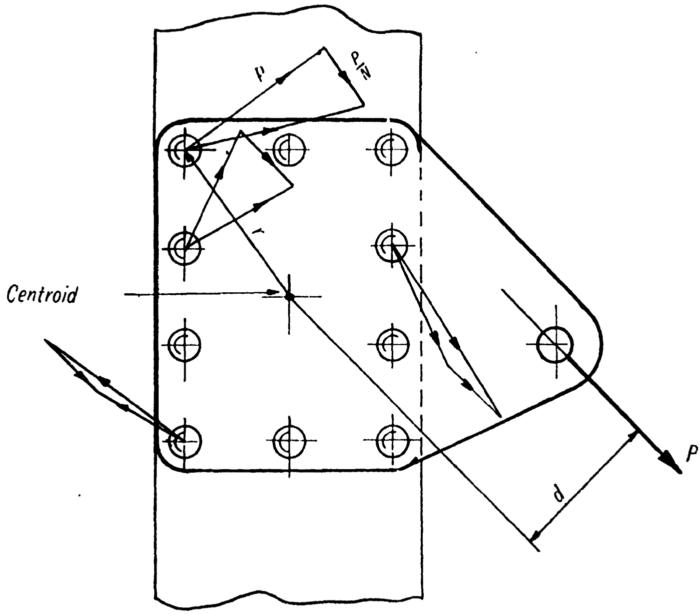


FIG. 29.

When a group of rivets (fig. 29) has to resist an offset load, the load on each rivet can be considered as being the vector sum of the direct load $\frac{P}{N}$ and the moment Pd. To find the loads on the rivets proceed as follows.

(1) Find the centroid of the rivets (work on rivet areas if all of the same material as is usually the case—if materials are different the areas should be adjusted in proportion to the bearing or shearing strengths of the various rivets).

(2) For each rivet calculate (area of rivet $\times$ r^2) and find total for all the rivets ($= \Sigma Ar^2$).

Then the load p on any one rivet X due to the offset load and acting in a direction normal to the radius r,

$$p = \frac{Pd \times \text{area of rivet } A \times r}{\Sigma Ar^2}$$

Then total load on rivet X = vector sum of $\frac{F}{N}$ and p.

Where $\frac{P}{N}$ acts in the direction of P.

Notes on Riveted Joints for Aircraft.

Never use rivets in tension.

Width of material round a rivet should not be less than $1\frac{1}{2}$ $\times$ diameter of rivet.

Rivet centres should not be less than 3 $\times$ diameter of rivet.

Solid rivets in double shear can only develop about $1\frac{1}{2}$ $\times$ single shear strength of rivet.

The strength of a group of rivets can be accurately determined by experiment only.

For riveting thin plates hollow rivets are easier to rivet and have bearing strengths more nearly equal to their shearing strengths than solid rivets.

The riveting of a long row of rivets causes a considerable stretch in the material.

Wiring Lugs and Sockets.

Let f_t = tensile strength of lug in lbs./in.^{2}

f_B = bearing strength of lug or pin (whichever is the less) in lbs./in.^{2}

f_s = shear strength of pin in lbs./in.^{2}

Then referring to fig. 30—

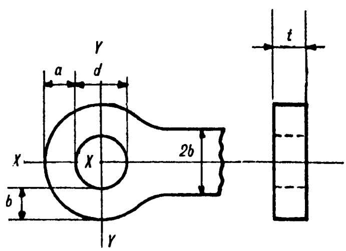


FIG. 30.

$$\text{Strength across XX} = \frac{5}{4} at f_t \text{ lbs.}$$

$$\text{,, ,, YY} = 2bt f_t \text{ lbs.}$$

$$\text{,, in bearing} = dt f_B \text{ lbs.}$$

$$\text{,, in shear} = 2\left(\frac{\pi d^2}{4}\right) f_s \text{ lbs.}$$

For a correctly designed lug these values should be approximately equal.

In order that the full bearing and shear strengths of the joint may be developed it is important that the thread finishes outside the lug.

It has been found that a large bearing stress will cause a considerable reduction in the shear strength of the pin. (See Air Publication 970, chapter VIII, section IV.) The maximum permissible bearing stress f_B that may be used depends upon the type of load to be taken. If the load is an alternating one, such as may occur on an engine mounting, it should be kept low. It is frequently advisable to bush lugs with H.T.S. bushes to prevent the development of slackness and of acilitate its remedy should it occur under working conditions.

STRUTS.

P = Load applied to strut in lbs.
A = Cross-sectional area in ins.^{2}
l = Length of strut in ins.
p = Ultimate strength of material in lbs./in.^{2}
E = Modulus of elasticity in lbs./in.^{2}
I = 2nd moment of area.
k = Radius of gyration of section in ins.
h = Distance of farthest fibre from the N.A.
a = Constant determined by experiment.

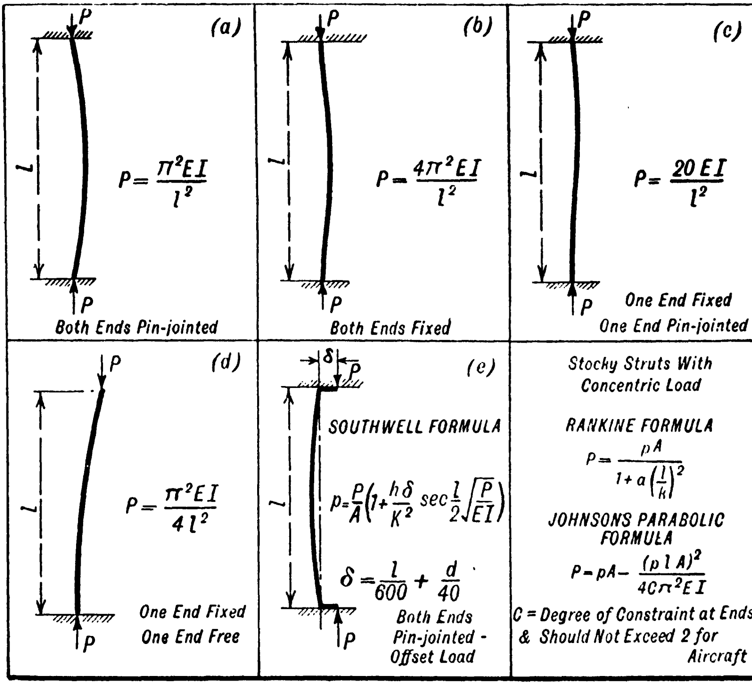


FIG. 31.

For long slender struts with concentric loading the well-known Euler formula may be used (fig. 31) to determine the maximum load that may be applied to the strut without failure. The value of the load thus found is such, that if one, but very slightly in excess of it be applied to the strut and the strut then deflected a small amount, then the deflection will continue to increase until the strut breaks. Though the strut finally breaks through the stress exceeding the maximum tensile or compressive stress that the material will stand, it is important to understand that the collapse in its initial stages is due to instability. The stress at which the strut fails in buckling $\left(\frac{\text{Euler load}}{\text{area of cross section}}\right)$ may be considerably less than the maximum stress the material will stand as a small block, but for very short, stocky struts it will approach the latter value. It is therefore important to check the Euler stress to find if it exceeds the ultimate strength of the material.

The Euler formula applies only to closed sections.

The strength of open sections, as struts (such as channels and angles), can be accurately found only by test.

In using the Euler formula for aeroplane struts it is unwise, owing to the deflection of the adjacent structure, to count on having fixed ends.

Short stocky struts with concentric loading can be designed either to the Johnson Parabolic formula or the Rankine formula.

Struts with Offset Loads.—In practice it is inevitable that there should be an eccentricity between the applied load and the neutral axis of the strut. The eccentricity will in general be the sum of the following :—

(*a*) Initial bow in tube.
(*b*) Offset end fittings.
(*c*) Bore of tube eccentric with outside.
(*d*) Offset load.

Various formula have been evolved to make allowance for this, the most well known, perhaps, being the Southwell formula. This equation is best solved by trial and error or by means of method described in Air Publication 970.

Many curves, however, are in existence giving the strength of struts of various diameters and thickness and materials that obviate the necessity for laborious calculation.

BEAM-COLUMNS.

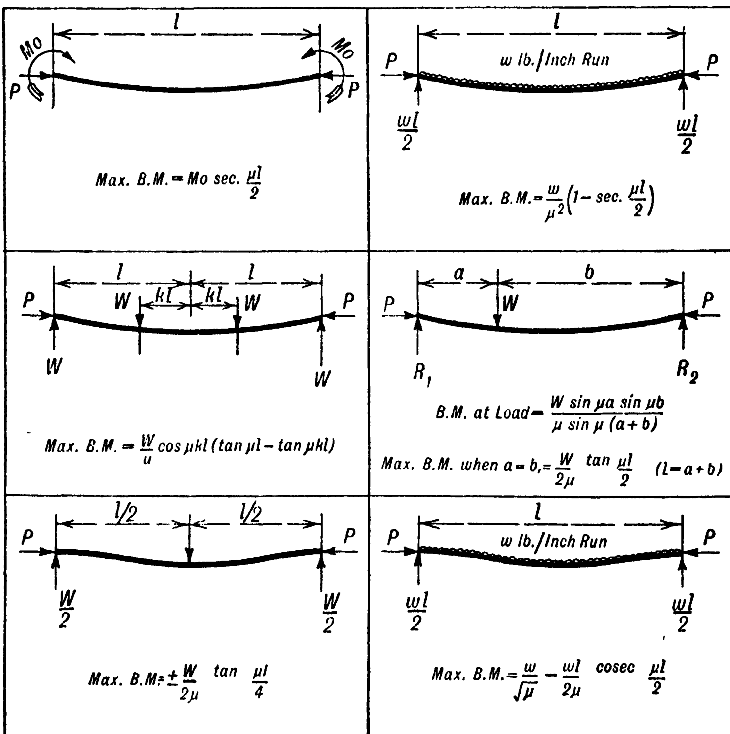
P = End load in lbs.

$\mu = \sqrt{\dfrac{P}{EI}}$.

I = 2nd moment at area in in.^{4}

E = Modulus of elasticity in lbs./in.^{2}

Max. B.M. in lbs./ins. when lengths are in inches and loads are in pounds



The Max. Stress is given by $p = \dfrac{P}{A} + \dfrac{\text{Max. B.M.}}{Z}$ where Z is the modulus of the section.

FIG. 32.

A beam-column can be defined either as a strut with a side load or a beam with an end load. In fig. 32 are given the formula for the more simple conditions of loading. For the stressing of continuous beams on three or more supports, with end loads, the reader is advised to consult Air Publication 970 or 'Aeroplane Structures' by Pippard and Pritchard.

Beams.

The elementary formulæ for beams on two supports are given on p. 424.

When a beam is continuous and carried on more than two supports, the theorem of three moments has to be used for determining the reactions at the supports and the bending moment diagram.

Theorem of three moments for a uniform load and rigid supports.—For a beam as in fig. 33 we have

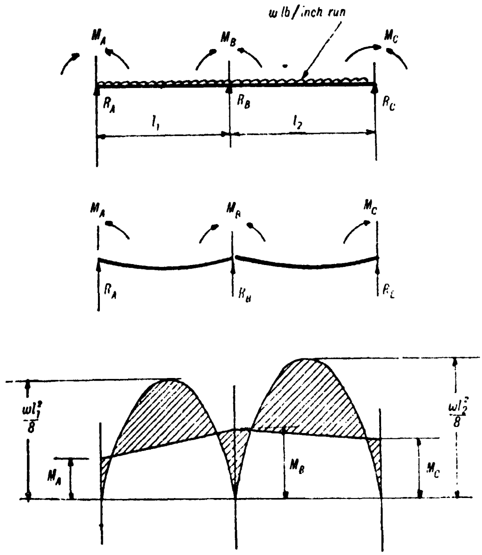


FIG. 33.

$$M_A l_1 + 2M_B(l_1 + l_2) + M_C l_2 = \frac{w}{4}(l_1^3 + l_2^3).$$

If there are more than three supports the above equation is applied successively to the adjacent pairs of spans and solved together.

For three supports there are three unknowns and one equation.

For N supports there are N unknowns and (N−2) equations.

The value of the bending moments at each end of the beam are known by the end conditions. For example, in fig. 36 as there is no overhang and the joints A and C are pinjoints $M_A = M_C = 0$.

The reactions at the supports (fig. 33) :—

$$R_B = \frac{wl_1}{2} + \frac{wl_2}{2} + \frac{(M_B - M_A)}{l_1} + \frac{(M_B - M_C)}{l_2}$$

$$R_A = \frac{wl_1}{2} + \frac{(M_A - M_B)}{l_1} \text{ and } R_C = \frac{wl_2}{2} + \frac{(M_C - M_B)}{l_2}$$

To draw the bending moment diagram, draw the diagram for each span of the beam as though it were pin-jointed at each support. Then superimpose the bending moment diagram for the fixing moments. The shaded portion of the diagram is the resultant bending moment diagram.

The above equations apply only to a beam with a uniform load. For any other system of loading the following should be used.

$$\frac{6A_1x_1}{l_1} + \frac{6A_2x_2}{l_2} + M_A l_1 + 2M_B(l_1 + l_2) + M_C l_2 + 6EI\left(\frac{y_a}{l_1} + \frac{y_c}{l_2}\right) = 0$$

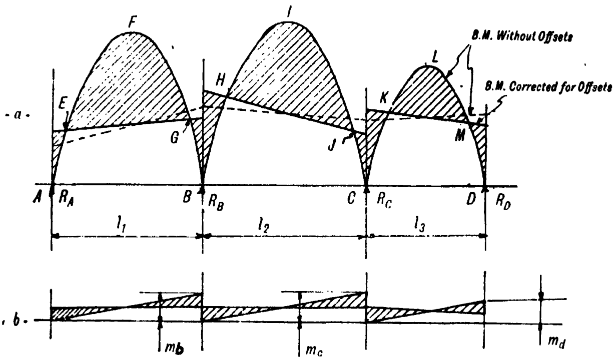


FIG. 34.

where in fig. 34 A_1 = area AEFGB (bending moment diag. of a simply supported beam between A and B).

A_2 = area BHIJC.

x_1 = Distance of centroid of AEFGB from A.

x_2 = " " " BHIJC from C.

y_a = vertical distance of B below A after loading.

y_b = " " " C " "

assuming A, B and C to be collinear before loading.

The reactions at the supports are the sum of the reactions for simply supported beams and reactions due to the end moments.

OFFSET MOMENTS.

Offset bend moments are introduced in the manner indicated in fig. 35.

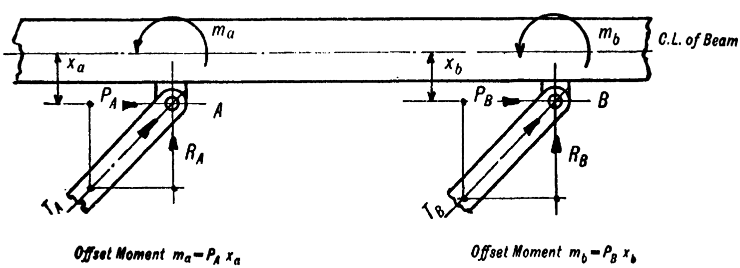


FIG. 35.

To find the bending moment diagram for a continuous beam with offset moments, first find the bending moment diagram, neglecting the offsets. Calculate the offset moments m_a, m_b,

m_0, etc., and apply the theorem of three moments to determine the bending moment diagram due to them acting alone.
This gives

$$M'_A l_1 + 2M'_B(l_1 + l_2) + M'_C l_2 = m_a l_1 + 2m_b l_2$$

where M_A', M_B', etc., are the bending moments at the supports due to the offsets.

From the end conditions and the repeated application of this formula the bending moment diagram for the offsets can be drawn (fig. 36b). This diagram should then be superimposed on the bending moment diagram drawn for no offsets (fig. 36a).

Generalised Theorem of Three Moments.

The form of the simple theorem of three moments as applied to continuous beams has been extended by Berry to allow for effects of compressive or tensile end loads in the various bays, such as occur in aeroplane spars.

Spar with Compressive End Loads.

If the supports remain collinear, we have, for two adjacent bays :—

$$\frac{M_{AR}l_1}{E_1I_1} f(\theta_1) + \frac{2M_{BL}l_1}{E_1I_1} \phi(\theta_1) + \frac{2M_{BR}l_2}{E_2I_2} \phi(\theta_2) + \frac{M_{CL}l_2}{E_2I_2} f(\theta_2) + \frac{w_1l_1^3}{4E_1I_1} \psi(\theta_1) + \frac{w_2l_2^3}{4E_2I_2} \psi(\theta_2) = 0;$$

where

$$f(\theta) = \frac{6(2\theta \operatorname{Cosec} 2\theta - 1)}{(2\theta)^2}; \qquad \phi(\theta) = \frac{3(1 - 2\theta \operatorname{Cot} 2\theta)}{(2\theta)^2}$$

$$\psi(\theta) = \frac{3(\tan \theta - \theta)}{\theta^3}; \qquad (2\theta)^2 = \pi^2 \frac{P}{P_e}.$$

P_e = Euler load for span in plane of bending.

Spar with Tensile End Loads.

For two adjacent bays we have :—

$$\frac{M_{AR}l_1}{E_1I_1} F(\theta_1) + \frac{2M_{BL}l_1}{E_1I_1} \Phi(\theta_1) + \frac{2M_{BR}l_2}{E_2I_2} \Phi(\theta_2) + \frac{M_{CL}l_2}{E_2I_2} F(\theta_2) + \frac{w_1l_1^3}{4E_1I_1} \Psi(\theta_1) + \frac{w_2l_2^3}{4E_2I_2} \Psi(\theta_2) = 0.$$

where

$$F(\theta) = \frac{6(1 - 2\theta \operatorname{Cosech} 2\theta)}{(2\theta)^2} \qquad \Phi(\theta) = \frac{3(2\theta \operatorname{Coth} 2\theta - 1)}{(2\theta)^2}$$

$$\Psi(\theta) = \frac{3(\theta - \tanh \theta)}{\theta^3}.$$

Spar with Compressive and Tensile End Loads in Adjacent Bays.

In this case the relevant terms for each bay are selected from the two general equations given above, so that the equation for two such bays will contain both trigonometrical and hyperbolic terms.

Values of the Functions.

The above functions have been calculated by Berry for various values of θ and are to be found in Air Publication 970 (H.M.S.O.).

Stressed Skin or Monocoque Construction.

A stressed skin structure consists of a thin sheet metal shell stiffened and strengthened by transverse and longitudinal members attached (usually by rivets but sometimes by welding) to the inside surface of the skin. The transverse members are called hoops, frames or bulkheads and the longitudinal members stringers (or longerons). The functions of the bulkheads are to prevent distortion of the monocoque, stabilise the longitudinals and to distribute over the monocoque skin any concentrated loads (such as those imposed by the landing gear, wing attachments, tail unit, etc.).

Stringers are necessary to increase the longitudinal compressive strength of the monocoque.

For the stressing of thin sheet metal structures there is no method approaching in simplicity and accuracy the well-known methods for determining the loads in the members of a tubular

structure. This is mainly due to the difficulty in calculating the loads at which local buckling or elastic instability of the thin sheet will occur.

However, a considerable amount of research has been carried out on problems of a simple but fundamental nature, and below is given a very brief summary of some of the results. It cannot be too strongly emphasised that in a last resort actual tests are the only satisfactory source of correct and reliable data. Even in the laboratory experimental results may differ by as much as 100 per cent., so in practice where the conditions of loading are still more uncertain, it is possible that greater differences than this may occur.

Thin-Walled Cylindrical Tube in Compression.

If the tube has a large l/k it will fail by buckling (fig. 36) (*i.e.* as a strut). If its l/k is small it will fail by local buckling or wrinkling. The tube should be stressed for both cases.

$$p \text{ (wrinkling stress in lbs./in.}^2) = 0 \cdot 18 \frac{Et}{r};$$

$$p \text{ (Euler or buckling stress in lbs./in.}^2) = \left(\frac{\pi k}{l}\right)^2 E;$$

where

E = Mod. of elasticity in lbs./in.²

k = Rad. of gyration of tube in ins. = $0 \cdot 7r$.

Flat Corrugated Sheet in Compression.

Where the compressive loads in a flat sheet are considerable it is sometimes expedient to replace a large number of closely spaced stringers by a corrugated sheet (fig. 37). Box spars are usually built this way.

A conservative estimate of the maximum wrinkling stress is given by the formula

$$p = 0 \cdot 12 \frac{Et}{r} \text{ lbs./in.}^2$$

The corrugations should also be stressed by the strut formula

$$p = C\left(\frac{\pi k}{l}\right)^2 E \text{ lbs./in.}^2$$

where C = fixing moment coefficient which should normally be taken as $1 \cdot 0$ but may be increased to a maximum of $1 \cdot 5$ under very favourable conditions.

k = the rad. of gyration must be calculated for the particular corrugation used.

Corrugated Thin Walled Tube in Compression.

The strength of a thin metal tube (fig. 38) may be greatly increased by corrugations, when the critical stress is due to wrinkling or elastic instability.

$$p \text{ (wrinkling stress in lbs./in.}^2) = 0 \cdot 12 \frac{Et}{r}.$$

$$p \text{ (Euler or buckling stress)} = \left(\frac{\pi k}{l}\right)^2 E$$

k = Rad. of gyration of tube = $0 \cdot 7R$ (not $\cdot 7r$).

r = Rad. of corrugation.

Thin Walled Circular Tube in Torsion.

The maximum shear stress that a thin metal tube (fig. 39) will withstand when in torsion without the development of local wrinkling is given by

$$f = \frac{CEt}{R} \text{ lbs./in.}^2$$

where $C = 0 \cdot 06$ to $0 \cdot 07$

Owing to the fact that shear wrinkles can develop in relatively small areas of unsupported surface, the presence of stringers and bulkheads does not affect to any marked extent the above formula.

For normal shear failure $p = \dfrac{T}{2\pi r^2 t}$ lbs./in.²

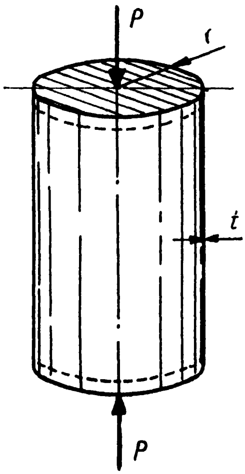


FIG. 36.

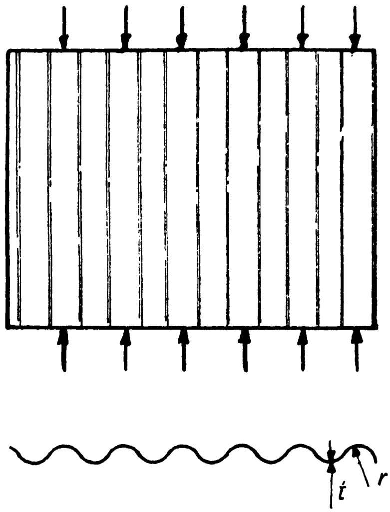


FIG. 37.

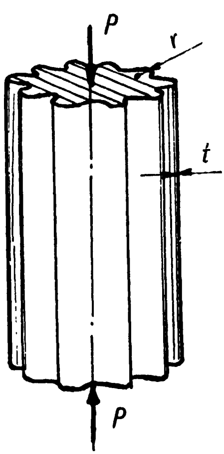


FIG. 38.

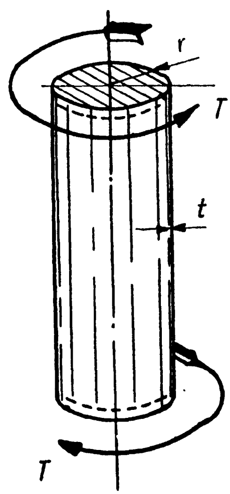


FIG. 39.

THIN WALLED CIRCULAR TUBE IN BEND (WITHOUT SHEAR).

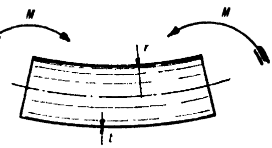


FIG. 40.

The compressive stress at which local wrinkling occurs for the case in fig. 40 is given by the formula

$$p = 0\cdot20\,\frac{Et}{r}\ \text{lbs./in.}^2$$

If tube is corrugated on the compression side this stress may be increased to

$$p = 0\cdot12\,\frac{Et}{r}\ \text{where now } r = \text{rad. of corrugations.}$$

BULKHEADS.

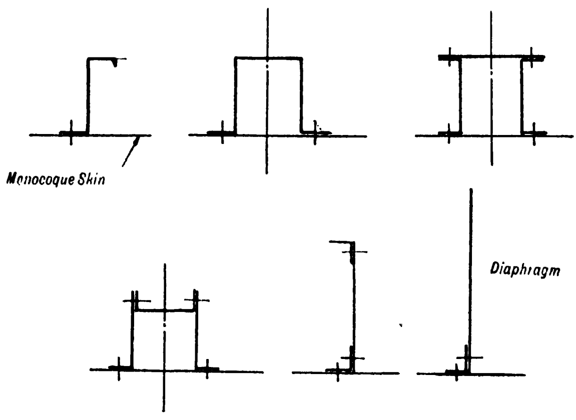


FIG. 41.

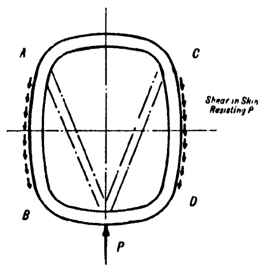


FIG. 42.

Fig. 41 shows a few typical bulkhead sections. The functions of a bulkhead have already been briefly referred to. When a concentrated load P is applied to a bulkhead (see fig. 42) it

is resisted by the shearing force in the skin between the points A and B and C and D, where the rivets connecting the bulkhead to the skin must be adequate to transmit the load from the bulkhead to the skin. If there are no reinforcing struts (shown chain-dotted) the bulkhead between B and D must be capable of transmitting the bending moment due to P.

STRINGERS.

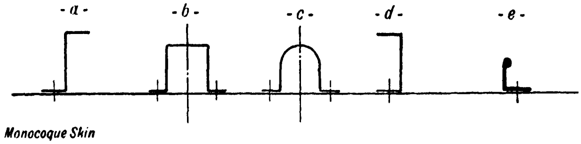


FIG. 43.

Fig. 43 shows a few typical stringer sections. The sections (*b*) and (*c*) are more efficient as struts than the other sections, whose free edges are liable to be unstable. In the choice of a stringer, of which a considerable length may be required per machine, the problems of manufacture and erection should be considered as well as those of strength and stiffness.

The failure of a stringer-skin combination may arise from any of the following causes:—

(1) Failure as a strut.
(2) Failure due to wrinkling of stringer.
(3) Failure due to wrinkling of skin.
(4) Failure due to twisting of stringer.

It should be noted that as the stringer is more likely to collapse as a strut towards the centre of the machine than outwardly (owing to the support given by the skin in the latter direction), it is perhaps desirable to offset the load on the stringer towards the centre of the machine away from the neutral axis of the stringer (see fig. 44).

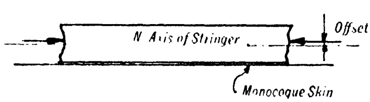


FIG. 44.

It is important to remember that the wrinkling of the skin may cause the premature failure of the stringer and *vice versa*.

The strength of the closed sections in compression, such as (*b*) and (*c*), can be calculated by the Euler formula, but the strength of the other open sections can be accurately determined only by experiment.

One method in estimating the strength of a stringer skin combination is to estimate the maximum stress permissible in either acting alone, and then multiply the lesser of these by the sum of their area.

HOLES IN A MONOCOQUE SKIN.

Holes in a monocoque should be so stiffened that the distortion with the hole is no greater than without it. Lack of rigidity may cause the development of high stresses in other parts of the structure.

The Torsion of Thin Sections.

WRINKLING STRESS.

When a thin-walled section is in torsion (fig. 45) the shear per unit length of section is constant. (This is implied in the formula $f = \frac{T}{2tA}$ where f = shear stress/in.^{2}, T = torsion, t = thickness of section, and A = area enclosed by outside contour of section.)

The value of the shear stress at which local wrinkling occurs is given by

$$f = 0{\cdot}06\,\frac{Et}{R}\ \text{lbs./in.}^2$$

where R is the argest radius of curvature of the section.

ANGULAR TWIST.

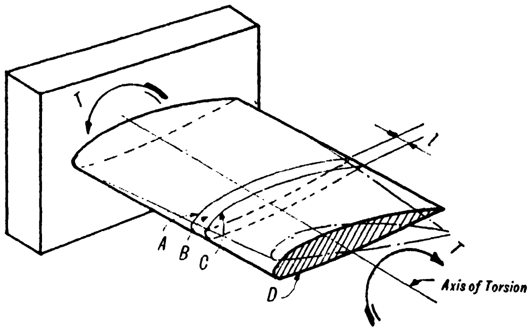


FIG. 45.

To determine the angular deflection about the flexural axis of the section shown in fig. 45 under the action of a continuously applied torque (continuous but varying along the length of the section), first divide the section into a number of comparatively short sections such as AC.

Then the angular twist of AC due to the torque applied between B and C is given by the formula:

$$\Delta\theta = \frac{TCl}{4A^2tN} \text{ radians}$$

where T = torque applied between B and C in lbs. ins.;
C = periphery of section in ins.
= AC in ins.;
A = area enclosed by contour of section in ins.^{2};
t = thickness of section in ins.;
N = shear modulus in lbs./in^2.

The total angular twist of the whole section is given by $\theta = \Delta\theta_1 + \Delta\theta_2 + \Delta\theta_3 +$ etc.
$= \Sigma\Delta\theta$ radians.

TORSION COMBINED WITH AXIAL COMPRESSION.

Let f_t = maximum permissible shear stress } From wrinkling formula above.
p_a = ,, ,, comp. ,,
f = stress due to torsion
p = ,, ,, ,, compression.

Then $\frac{p}{f_t - f} = \frac{p_a}{f_t}$ (conservative values).

Having determined f for a given torsion we can calculate the maximum compressive load that may be applied with it, or *vice versa*.

DEEP BEAMS WITH THIN WEBS (TENSION FIELD THEORY).

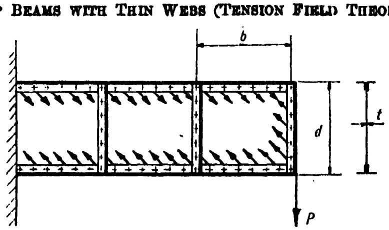


FIG. 46.

In this theory the web is assumed to be incapable of withstanding compression (fig. 46).

Wager, to whom the theory is due, gives the following formula :—

Let f = shear stress in lbs./in.^{2}

F = total vertical shear at section considered in lbs.

Then maximum tensile stress in web $= 2f = \dfrac{2F}{dt}$ lbs./in.^{2}

and is inclined at slightly less than 45° to the flanges.

VERTICAL LOAD ON STIFFENER.

$$P = \frac{Fb}{d} \text{ lbs.}$$

Note that the web being in tension prevents the stiffener failing through buckling as a strut.

FLANGE MEMBERS.

Due to the tension field in the web there is a tendency for the flanges to sag between the stiffeners.

$$\text{Load in flanges} = \frac{BM}{d} + \frac{F}{2} \text{ lbs.}$$

RIVETS CONNECTING FLANGES TO WEBS.

$$\text{Load per rivet} = 1{\cdot}35\frac{Fp}{d}$$

where p = pitch of rivets in ins.

F = vertical shear at section in lbs.

RIVETS THROUGH JOINT IN WEB.

$$\text{Load per rivet} = 1{\cdot}5\frac{Fp}{d}$$

where p = pitch of rivets.

CRITERION FOR A TENSION FIELD WEB.

To determine which of the two types of web are more economical the index figure $\dfrac{\sqrt{F}}{d}$ can be used.

If $\dfrac{\sqrt{F}}{d} < 8{\cdot}32$ a tension field web is demanded, but if $> 12{\cdot}5$ a shear-resistant web is in order. At values intermediate between these the more economical type of design can be determined only by more actual comparison of the two types of beam.

DEEP BEAMS WITH NON-BUCKLING WEBS.

The shear stress which produces wrinkling of the web is—

$$f = \frac{f^2}{3{\cdot}6\,CE}\left(\frac{b}{t}\right)^2 \text{ lbs./in.}^2$$

= ultimate shear stress in lbs./in.^{2}

If b/t is $> 1{\cdot}35\left(\dfrac{CE}{f}\right)^{\frac{1}{2}}$

the shear stress which produces wrinkling of the web is—

$$\frac{0{\cdot}9\,CE}{(b/t)^2} \text{ lbs./in.}^2$$

STIFFENERS FOR NON-BUCKLING WEB.

Intermediate stiffeners must be used when $b > 60t$. The size of these stiffeners can be fixed by the Wagner Equation.

$$\text{Second moment of area of stiffener} = \frac{2{\cdot}3}{t}\,C\left(\frac{Fd}{33E}\right)^{\frac{4}{3}} \text{ ins.}^4$$

where C = distance between centre of stiffener rivets in ins.

FRAMES.

Let L = number of links in structure.

N = number of joints (or nodes).

Then if the structure is to be rigid we have as the minimum number of links possible, the following equations.

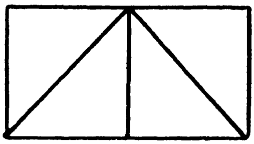


FIG. 47.

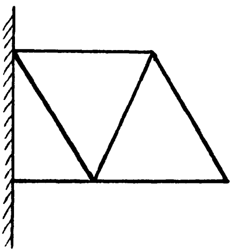


FIG. 48.

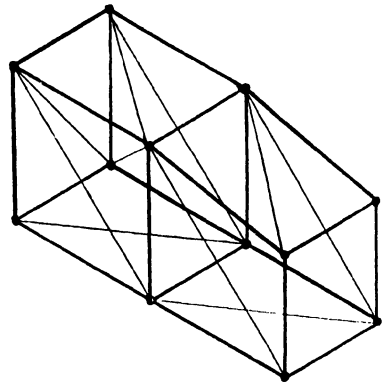


FIG. 49.

Plane Frame (fig. 47), L = 3N − 3.

Plane Frame attached to a solid support (fig. 48), L = 2N.

Space Frame (fig. 49), L = 3N − 6.

Space Frame attached to a solid support, L = 3N.

If a structure has fewer links than required by the above equations the structure is definitely incomplete; but, on the other hand, if the number of links satisfies or even more than satisfies the same equations, it does not necessarily follow that the structure is complete (since one part of a structure may have more links than necessary and another part less).

Structures that have fewer links than is required are called incomplete or deficient.

Structures that are just stiff with the minimum number of links are called determinate.

Structures that have more links than necessary for rigidity are called redundant or indeterminate.

Plane frames should be stressed by the usual method of drawing the force diagram.

Space frames can be stressed by Southwell's method of tension coefficients (see 'Aeroplane Structures' by Pippard and Pritchard).

Redundant frames can be stressed by the application of the Principle of Least Work or Castigliano's Theorems.

Particulars of Various Aeroplanes.

Particulars of various types of aeroplanes are given in Table IV., pp. 433-439.

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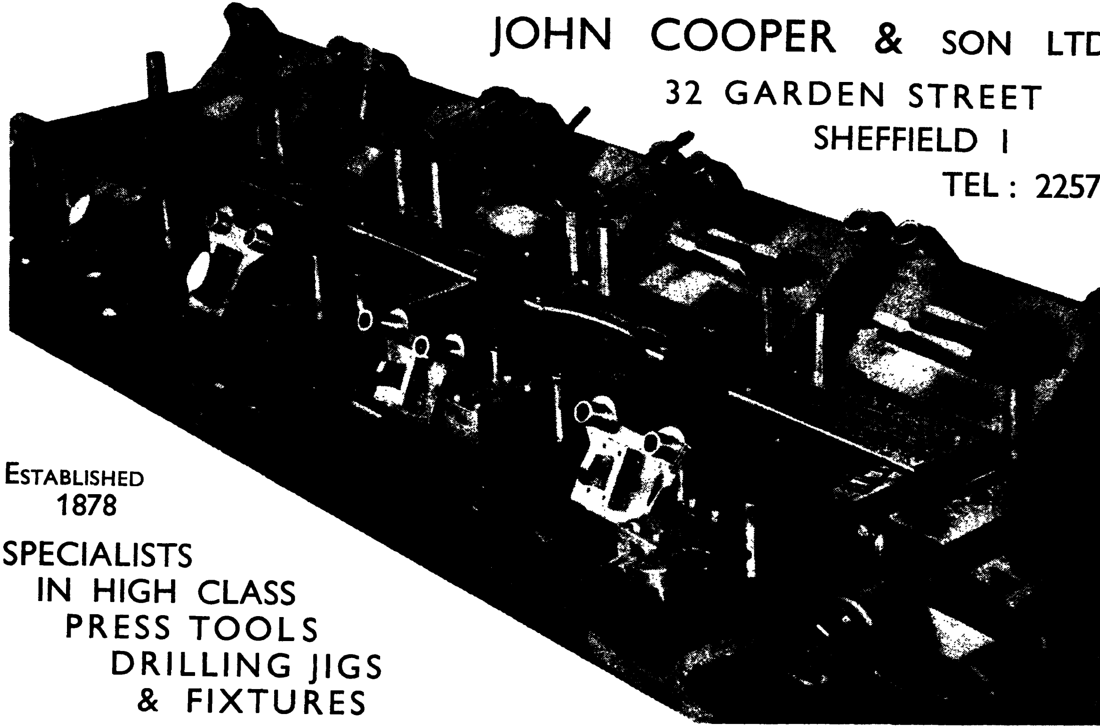

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SPECIALISTS
IN HIGH CLASS
PRESS TOOLS
DRILLING JIGS
& FIXTURES

F. 433

TABLE IV.—PRINCIPAL BRITISH MILITARY AND NAVAL AIRCRAFT: ABRIDGED DETAILS

Name.	Makers.	Engine(s). No.	Engine(s). Name.	Engine(s). Power per Engine.	Armament: Bombs: Cameras.	Speed. m.p.h.	Range. Miles.	Weight. lb.	Span. ft. in.	Remarks.
ROYAL AIR FORCE										
					Fighters and Fighter-Bombers.					
VAMPIRE F.3	de Havilland	1	Goblin 2	3,000 lb. s.t.	4 20 mm. guns	531	1,145	12,170	40 0	
METEOR F.4	Gloster	2	Derwent 5	3,500 lb. s.t.	4 20 mm. guns	585	950	15,175	37 2	Also equipped with racks for 2 1,000 lb. bombs, and/or 8 90 lb. rocket projectiles.
METEOR F.8	Gloster	2	Derwent 5	3,500 lb. s.t.	4 20 mm. guns	—	—	—	37 2	This version has a slightly longer fuselage than the Mk. 4.
HORNET F.3	de Havilland	2	Merlin 130 series	1,660 h.p	4 20 mm. guns	467	2,100	19,468	45 0	Equipped with racks for 2 1,000 lb. bombs and rocket projectiles.
MOSQUITO N.F.38	de Havilland	2	Merlin 113	1,535 b.h.p.	4 0·303 in. guns 4 20 mm. guns	404	2,050	21,400	54 2	
VAMPIRE F.B.5	de Havilland	1	Goblin 2	3,000 lb. s.t.	4 20 mm. guns; bombs and/or rocket projectiles	—	—	—	40 0	On part publication list.
SPITFIRE F.B.24	Vickers-Supermarine	1	Griffon	2,375 h.p.	4 20 mm. guns 3 500 lb. bombs or rocket projectiles	450	965	11,290	36 11	
BRIGAND F.B.1	Bristol	2	Centaurus 57	2,500 h.p.	4 20 mm. guns. Bombs or mines carried under fuselage. One bomb or 4 r.p. carried under each wing	358	2,810	39,987	72 4	

TABLE IV. (*contd.*)—PRINCIPAL BRITISH MILITARY AND NAVAL AIRCRAFT: ABRIDGED DETAILS.

Name.	Makers.	Engine(s). No.	Engine(s). Name.	Power per Engine.	Armament: Bombs: Cameras.	Speed. m.p.h.	Range. Miles.	Weight. lb.	Span. ft. in.	Remarks.
ROYAL AIR FORCE (*contd.*)										
					Bombers					
MOSQUITO B.35	de Havilland	2	Merlin 113	1,535 h.p.	1 4,000 lb. bomb, or 4 500 lb. bombs when fitted with long-range tanks	422	2,050	25,200	54 2	
LINCOLN	A. V. Roe	4	Merlin 85	1,680 h.p.	Twin 0·5 in. guns in nose-turret; 2 20 mm. Hispano guns in dorsal turret; twin 0·5 in. guns in rear turret. Max. bomb load, 22,000 lb.	295	Max. 3,380 1,150*	82,000	120 0	* With max. bomb load.
					Reconnaissance Aircraft					
MOSQUITO P.R.34	de Havilland	2	Merlin 76, 77 or Merlin 113, 114	1,250 h.p.	Electrically - operated vertical and oblique cameras: no armament	425	3,500	22,587	54 2	
SHACKLETON G.R. Mk.1	A. V. Roe	4	Griffon 57	2,650 h.p.	—	—	—	94,000	120 0	No further details yet available.
BRIGAND	Bristol	2	Centaurus 57	2,500 h.p.	—	358	2,810	39,987	72 4	Another version is used for meteorological surveys.
					Artillery Observation Post					
AUSTER 6	Auster	1	Gipsy Major 7	145 h.p.	Unarmed but carries radio.	124	345	2,147	36 0	
					Air/Sea Rescue Aircraft					
SEAGULL	Vickers Supermarine	1	Griffon 57	2,450 h.p.	—	260	1,230	14,500	52 6	

TABLE IV. (*contd.*)—PRINCIPAL BRITISH MILITARY AND NAVAL AIRCRAFT: ABRIDGED DETAILS.

Name.	Makers.	Engine(s). No.	Engine(s). Name.	Power per Engine.	Armament: Bombs: Cameras.	Speed. m.p.h.	Range. Miles.	Weight. lb.	Span. ft. in.	Remarks.
ROYAL AIR FORCE (*contd.*)										
					Flying-Boats					
SUNDERLAND 5	Short Brothers	4	R-1830-90B. Twin Wasp	1,200 b.h.p.	10 0·303 in. machine-guns in turrets at nose, amidships and extreme tail. 2 0·5 in. manually-operated beam guns. Bombs, depth charges, etc., carried on railed tracks.	213	2,044	65,000	112 9	
SEAFORD	Short Brothers	4	Hercules 100	837 h.p.	2 0·5 in. machine-guns in turret 4 0·303 in. guns in nose of hull 2 20 mm. cannon 2 0·5 in. beam guns 2 0·5 in. tail guns	242	3,100	75,000	112 9	Also carries bombs, depth-charges, etc., on railed tracks.
					Transports					
HASTINGS C. MK.1.	Handley Page	4	Hercules 101	1,675 h.p.	—	354	2,900	78,000	113 0	Also fulfils rôle of glider tug, freighter, paratroop transport, ambulance, troop-carrier, supply dropper, jeep carrier.
VALETTA C. MK.1	Vickers-Armstrongs Weybridge	2	Hercules 230	1,975 h.p.	—	294	300–* 1,680	36,500	89 3	* Depending on load. Also acts as glider tug.

TABLE IV. *(contd.)*—**PRINCIPAL BRITISH MILITARY AND NAVAL AIRCRAFT: ABRIDGED DETAILS.**

Name.	Makers.	Engine(s). No.	Engine(s). Name.	Engine(s). Power per Engine.	Armament: Bombs: Cameras.	Speed. m.p.h.	Range. Miles.	Weight. lb.	Span. ft. in.	Remarks.
ROYAL AIR FORCE *(contd.)*										
				Transports (contd.)						
HALIFAX C.8 .	Handley Page	4	Hercules 100	837 h.p.	—	270	3,000	65,000	104 0	
LANCASTRIAN .	A.V. Roe	4	Merlin 24	1,280 h.p.	—	315	3,200	65,000	102 0	
DEVON . .	de Havilland	2	Gipsy Queen 71	330 h.p.	—	210	—	8,500	57 0	
				Transport Gliders						
HORSA 2 . .	Airspeed	—	—	—	—	—	—	15,500	88 0	Main compartment seats 15 fully-armed airborne troops.
HAMILCAR . .	General Aircraft	—	—	—	—	—	—	36,000	110 0	A variety of military equipment can be carried up to a max. of 17,500 lb.
				Communication Aircraft						
AUSTER 6 . .	Auster	1	Gipsy Major 7	145 h.p.	—	124	345	2,147	36 0	
ANSON C.12 .	A. V. Roe	2	Cheetah 10	420 h.p.	—	190	820	9,500	56 6	
DEVON . .	de Havilland	2	Gipsy Queen 70	345 h.p.	—	210	500	8,500	57 0	
PROCTOR 5 . .	Percival	1	Gipsy Queen 2	208 h.p.	—	146	500	3,500	39 6	

TABLE IV. (*contd.*)—PRINCIPAL BRITISH MILITARY AND NAVAL AIRCRAFT: ABRIDGED DETAILS.

Name.	Makers.	Engine(s). No.	Engine(s). Name.	Power per Engine.	Armament: Bombs: Cameras.	Speed. m.p.h.	Range. Miles.	Weight. lb.	Span. ft. in.	Remarks.
ROYAL AIR FORCE (*contd.*)										
				Trainers						
AUSTER 7	Auster	1	Gipsy Major 7	145 h.p.	—	122	315	2,122	36 0	Designed for training A.O.P. pilots.
ATHENA 1	A. V. Roe	1	Mamba	1,010 s.h.p. — 220 lb. s.t.	—	287	—	7,609	40 0	Endurance 2½ hrs.
ATHENA 2	A. V. Roe	1	Merlin 35	1,280 h.p.	1 0·303 in. gun G.45B camera-gun in starboard wing	287	—	8,273	40 0	
ANSON 20	A. V. Roe	2	Cheetah 15	425 h.p.	—	171	645	10,400	57 6	Bombing and gunnery trainer, navigator trainer, radio trainer, each approp. equipped.
BALLIOL 1	Boulton Paul	1	Mamba	1,010 s.h.p. — 220 lb. s.t.	—	307	—	7,845	39 4	
BALLIOL 2	Boulton Paul	1	Merlin 35	1,280 b.h.p.	—	305	525	8,175	39 4	
OPERATIONAL TRAINER	Fairey	1	Griffon 12	2,000 h.p.	—	310	735	14,582	44 6	Developed from the Firefly ship-borne fighter.
METEOR 7	Gloster	2	Derwent 5	3,500 s.t.	—	585	520	14,175	37 2	Developed from Meteor twin-jet fighter.

TABLE IV. (*contd.*)—PRINCIPAL BRITISH MILITARY AND NAVAL AIRCRAFT: ABRIDGED DETAILS.

Name.	Makers.	Engine(s).			Armament: Bombs: Cameras.	Speed. m.p.h.	Range. Miles.	Weight. lb.	Span. ft. in.	Remarks.
		No.	Name.	Power per Engine.						
ROYAL AIR FORCE (*contd.*)										
					Trainers (*contd.*)					
FURY TRAINER .	Hawker	1	Centaurus 18	2,560 h.p.	—	445	1,650	13,800	38 5	Developed from Sea Fury ship-borne fighter.
PRENTICE 1 .	Percival	1	Gipsy Queen 32	250 h.p.	—	150	485	4,200	46 0	
PRENTICE 2 . .	Percival	1	Gipsy Queen 51	295 h.p.	—	159	—	4,200	46 0	
SPITFIRE TRAINER	Vickers Supermarine	1	Merlin 66	1,325 h.p.	4 0·303 in. guns can be installed for gunnery training	393	418	7,400	36 10	Developed from the Spitfire fighter.
ROYAL NAVY										
					Fighters (including Torpedo Fighters)					
SEA VAMPIRE .	de Havilland	1	Goblin 2	3,000 lb. s.t.	4 20 mm. guns	526	1,145	12,660	38 0 (clip wing)	Developed from Vampire land-based fighter.
SEA MOSQUITO F.37	de Havilland	2	Merlin 25	1,620 b.h.p.	4 20 mm. guns	383	1,100	23,000	54 2	
SEA HORNET F.R.	de Havilland	2	Merlin 130/131	2,030 b.h.p.	4 20 mm. guns	472	3,000	18,530	45 0	
SEA FURY . .	Hawker	1	Centaurus 18	2,480 b.h.p.	4 20 mm. guns. Racks for bombs and rocket projectiles	450	1,800	12,350	38 5	Max. emergency weight 13,500 lb.
SEAFIRE 47 . .	Vickers Supermarine	1	Griffon 87	2,350 h.p.	4 20 mm. guns. Racks for bombs and rocket projectile launchers	452	940	11,615	36 11	

Name.	Makers.	Engine(s). No.	Engine(s). Name.	Power per Engine.	Armament : Bombs : Cameras.	Speed. m.p.h.	Range. Miles.	Weight. lb.	Span. ft. in.	Remarks.
ROYAL NAVY (*contd.*)										
				Fighters (including Torpedo Fighters) (contd.)						
FIREFLY 5	Fairey	1	Griffon 74	2,020 h.p.	4 20 mm. guns. Provision for rocket projectile gear and bombs	384	1,526	15,615	41 2	
ATTACKER	Vickers Supermarine	1	Nene	5,000 lb. s.t.	4 20 mm. guns. Provision for rocket projectile gear or bombs	590	1,240	11,500	36 11	
N.7/46	Hawker	1	Nene	5,000 lb. s.t.	—	—	—	—	36 6	Speed probably exceeds 600 m.p.h.
				Strike Aircraft						
FIREBRAND 5	Blackburn	1	Centaurus 9	2,500 h.p.	4 20 mm. guns 1 1,000 lb. bomb under each wing or 1 1,850 lb. torpedo. Racks for rocket projectiles	341	740*	17,500	51 3	* With torpedo.
WYVERN	Westland	1	Eagle	3,500 h.p.	4 20 mm. guns 1 20 in. torpedo or 1 2,000 lb. bomb or 8 60 lb. rocket projectiles	455	—	22,000	44 0	Endurance 3¾ hrs.
				Target Tugs						
MOSQUITO T.T.39	de Havilland	2	Merlin 72	—	—	292	—	23,000	54 2	Endurance 2 hrs. cruising and 1 hr. towing.
STURGEON 1	Short Bros.	2	Merlin 140	1,660 h.p.	—	430	1,600	21,700	59 9	Endurance, including 1 hr. towing, 3¼ hrs.

NOTE.—The following types are not included in this table :—
Gloster E.1/44 experimental jet fighter (prototype only built).
Auster A2/45, Heston A2/45 } Artillery Observation Post (prototypes only).
Saunders-Roe SR/A.1 experimental jet flying-boat fighter (three prototypes only built).
Scottish Aviation A4/45 light communications aircraft (prototype only).

The figures quoted in the table are all maxima, and are not necessarily related.

TABLE V.—BRITISH CIVIL AIRCRAFT—IN WEIGHT ORDER.

Name.	Makers.	Engine(s). No.	Engine(s). Name.	Engine(s). Power per Engine.	Accommodation.	Flight Deck Crew.	Dimensions. Span. ft. in.	Dimensions. Length. ft. in.	Dimensions. Height. ft. in.	Wing Area. sq. ft.	Cruising Speed Range. m.p.h.	All-up Weight. lb.
SR/45 . .	Saunders-Roe	10	Proteus	3,500 s.h.p.	100 passengers	—	220 0	118 0	—	—	380	315,000 (140 tons)
BRABAZON 1	Bristol	8	Centaurus	2,470 b.h.p.	Standard 70–100 seats. Max. 120 seats	7	230 0	177 0	50 0	5,317	250–285	290,000
UNIVERSAL .	General Aircraft	4	Hercules 261	1,950 h.p.	56–90 seats *	4	162 0	99 2	31 0	2,916	150–200	95,000
TYPE 175 .	Bristol	4	Centaurus 663	—	Standard 42 seats. Max. 50 seats	3	130 0	115 0	34 6	1,775	310	90,000
HERMES 5 .	Handley Page	4	Theseus	2,290 equiv. h.p.	40, 52 or 63 seats	5	113 0	96 10	29 11	1,408	322–349	84,000
HERMES 4 .	Handley Page	4	Hercules	2,100 b.h.p.	Standard 40 seats. Max. 63 seats	5	113 0	96 10	29 11	1,408	252–300	82,000
TUDOR 4 .	A. V. Roe	4	Merlin 621	1,740 b.h.p.	Standard 32 seats	5	120 0	85 3	24 0	1,421	240–282	82,000
†HERMES 6 .	Handley Page	4	Hercules 763	2,000 b.h.p.	Standard 40 seats. Max. 63 seats	5	113 0	97 0	—	1,408	250–298	82,000
TUDOR 2 .	A. V. Roe	4	Merlin 621	1,725 b.h.p.	Standard 40 seats. Max. 60 seats	5	120 0	105 7	24 0	1,421	260–280	80,000
‡TUDOR 3 .	A. V. Roe	4	Merlin 621	1,740 b.h.p.	Day-and-night 8 passengers. Private cabin 2 passengers	5	120 0	79 3	24 0	1,421	265–282	78,761

* 30,000 lb. payload as freighter (max.). † A development of the Hermes 4. ‡ This is a specially equipped version of the Tudor 1.

TABLE V. (*contd.*)—BRITISH CIVIL AIRCRAFT—IN WEIGHT ORDER.

Name	Makers.	Engine(s). No.	Engine(s). Name.	Engine(s). Power per Engine.	Accommodation.	Flight Deck Crew.	Dimensions. Span. ft. in.	Dimensions. Length. ft. in.	Dimensions. Height. ft. in.	Dimensions. Wing Area. sq. ft.	Cruising Speed Range. m.p.h.	All-up Weight. lb.
SOLENT	Short Bros.	4	Hercules 759	2,000 b.h.p.	Standard 40 seats plus 4,750 lb. payload	—	112·10	88·7	—	1,687	220–250	78,000
HERMES FREIGHTER	Handley Page	4	Hercules 101	1,675 b.h.p.	Max. payload 16,000 lb.	5	113 0	81 8	22 6	1,408	244–303	75,000
SANDRINGHAM	Short Bros.	4	Twin Wasp	1,000 b.h.p.	21–45 seats	5	112·10	86·3	9·9 beam at chine	1,687	171–179 knots	60,000
AMBASSADOR	Airspeed	2	Centaurus	2,700 b.h.p.	40 passengers (standard)	2 or 3	115 0	80 9	18 10	1,200	240–312	52,000 (max.)
APOLLO (MERLIN)	Armstrong Whitworth	4	Merlin 35	1,280 b.h.p.	Standard 24–27 seats. Max. 31 seats	3 or 4	92 0	70 11	—	986	285 (max. cruise)	43,150
APOLLO (R-1830)	Armstrong Whitworth	4	P. & W. R-1830	1,200 b.h.p.	Standard 24–27 seats. Max. 31 seats	3 or 4	92 0	70 11	25 1	986	250–270	43,000
VISCOUNT	Vickers	4	Dart	1,300 s.h.p.	Standard 32 seats	3	89 0	74 6	26 9	885	240–260	41,000
*TYPE 170	Bristol	2	Hercules 672	1,690 b.h.p.	Standard 32 seats. Max. 36 seats	3	108 0	68 4	21 6	1,487	162–207	40,000
APOLLO	Armstrong Whitworth	4	Mamba	1,010 s.h.p.	Standard 24 seats	3 or 4	92 0	70 11	25 1	986	276–305	39,500
VIKING FREIGHTER	Vickers	2	Hercules 730	1,975 b.h.p.	Max. payload 11,000 lb.	3	89 3	65 2	19 6	882	210–264	36,500

* As Freighter: Max. 11,383 lb. payload.

TABLE V. (*contd.*)—BRITISH CIVIL AIRCRAFT—IN WEIGHT ORDER.

Name.	Makers.	Engine(s). No.	Engine(s). Name.	Power per Engine.	Accommodation.	Flight Deck Crew.	Dimensions. Span. ft. in.	Length. ft. in.	Height. ft. in.	Wing Area. sq. ft.	Cruising Speed Range. m.p.h.	All-up Weight. lb.
VIKING 1B	Vickers	2	Hercules 634	1,690 b.h.p.	Standard 24 seats. Max. 34 seats	3	89 3	65 2	19 6	882	210 (recommended)	34,000
*MARATHON	Miles	4	Gipsy Queen 70	330 b.h.p.	14–22 seats	2 or 3	65 0	52 1	13 7	500	180–200	18,000
PRINCE	Percival	2	Leonides	570 b.h.p.	Standard 8 seats. Max. 10 seats	2	56 0	42 11	16 1	365	140–177	10,650
SEALAND 2	Short Bros.	2	Leonides LE4M	520 b.h.p.	5–8 seats	1 or 2	68 0	42 0	5 9 beam	388	150–168 (knots)	10,130
SEA OTTER	Vickers Supermarine	1	Mercury 30	805 b.h.p.	Standard 4 seats	2	46 0	39 9	—	610	100	10,000
SEALAND 1	Short Bros.	2	Gipsy Queen 70	345 b.h.p.	5–8 seats	1 or 2	59 0	42 0	5 3 beam	354	157 (knots)	9,100
DOVE	de Havilland	2	Gipsy Queen 70	330 b.h.p.	8–11 seats	2	57 0	39 6	13 4	335	165–179	8,500
CONSUL	Airspeed	2	Cheetah 10	410 b.h.p.	4–6 seats	1 or 2	53 4	35 4	11 1	348	133–156	8,250
PRESTWICK PIONEER	Scottish Aviation	1	Gipsy Queen 30	250 b.h.p.	Pilot and 3 passengers	—	49 9	34 9	9 9	390	114–124	4,250
TRIBIAN	Sponson Developments	2	Gipsy Major	145 h.p.	3–4 passengers	1	44 6	34 0	13 2	283	153	4,214
AEROCAR	Portsmouth Aviation	2	Cirrus Major 3	155 b.h.p.	5 seats	1	42 0	26 3	10 7	—	162	4,200

* The Marathon is being built by Handley Page (Reading) Ltd. for British European Airways Corp.

TABLE V. (*contd.*)—BRITISH CIVIL AIRCRAFT—IN WEIGHT ORDER.

Name.	Makers.	Engine(s). No.	Engine(s). Name.	Power per Engine.	Accommodation.	Flight Deck Crew.	Span. ft. in.	Length. ft. in.	Height. ft. in.	Wing Area. sq. ft.	Cruising Speed Range. m.p.h.	All-up Weight. lb.
*PROCTOR 6	Percival	1	Gipsy Queen 32	251 b.h.p.	2 passengers	1	39 6	28 6	—	202	126–130	3,750
PROCTOR 5	Percival	1	Gipsy Queen 2	208 b.h.p.	Pilot and 3 passengers	—	39 6	28 6	7 3	202	146 (max. cruise)	3,500
DESFORD	Reid & Sigrist	2	Gipsy Major 1	130 b.h.p.	Pilot and passenger	—	34 0	25 6	8 1	186	120–142	2,300
SATELLITE	Planet	1	Gipsy Queen 31	250 b.h.p.	4 or 5 seats	1	32 6	26 3	9 3	153	191 (max.)	2,905
AVIS	Auster	1	Gipsy Major 10	145 b.h.p.	Pilot and 3 passengers	—	36 0	23 2	—	185	100	2,550
NEWBURY EON	Elliotts	1	Cirrus Minor 2	101 b.h.p.	2 passengers with Cirrus	1	37 0	25 0	9 9	173	100 (Cirrus)	1,950 (Cirrus)
		1	or Gipsy Major 10	145 b.h.p.	3 passengers with Gipsy						116 (Gipsy)	2,350 (Gipsy)
SUPER-ACE	Chrislea	1	Gipsy Major 10	145 b.h.p.	Pilot and 3 passengers	—	36 0	21 6	7 7	177	110	2,342
PRIMER	Fairey	1	Gipsy Major 10	145 b.h.p.	Instructor and pupil	—	32 10	27 6	6 11	151·5	122 (Gipsy)	1,960
		1	or Cirrus Major 3	155 b.h.p.							125 (Cirrus)	
CHIPMUNK	de Havilland	1	Gipsy Major 10	145 b.h.p.	Pilot and passenger	—	34 4	25 5	7 6	172·5	122–124	1,900

* This is the floatplane version of the Proctor 5.

TABLE V. (*contd.*)—BRITISH CIVIL AIRCRAFT—IN WEIGHT ORDER.

Name.	Makers.	Engine(s). No.	Engine(s). Name.	Engine(s). Power per Engine.	Accommodation.	Flight Deck Crew.	Dimensions. Span. ft. in.	Dimensions. Length. ft. in.	Dimensions. Height. ft. in.	Dimensions. Wing Area. sq. ft.	Cruising Speed Range. m.p.h.	All-up Weight. lb.
AUTOCRAT	Auster	1	Cirrus Minor 2	101 b.h.p.	Pilot and 2 passengers	—	36 0	23 5	6 6	185	100	1,850
ARROW	Auster	1	Continental C–75	75 b.h.p.	Pilot and passenger	—	36 0	22 9	6 6	185	87	1,450

NOTE.—The following types are not included in this table :—
Avro Tudor 7 experimental airliner (four Bristol Hercules engines). Avro Tudor 8 experimental jet airliner (four Rolls-Royce Nene turbojets).
Vickers-Armstrongs Viking experimental jet airliner (two Rolls-Royce Nene turbojets). de Havilland D.H.108 research aircraft.
The figures quoted in the table are all maxima, and are not necessarily related.

TABLE V. (*contd.*)—HELICOPTERS.

Name.	Makers.	Engine(s). No.	Engine(s). Name.	Engine(s). Power per Engine.	Accommodation.	Flight Deck. Crew.	Dimensions. Span. ft. in.	Dimensions. Length. ft. in.	Dimensions. Height. ft. in.	Dimensions. Rotor dia. ft. in.	Cruising Speed Range. m.p.h.	All-up Weight. lb.
AIR HORSE	Cierva	1	Merlin 24	1,620 b.h.p.	—	—	—	—	—	47 0	125	17,500
WESTLAND SIKORSKY S.51	Westland	1	Leonides WS.51L	525 b.h.p.	3 passengers	1	57 6	41 1¾	12 11	48 0	85	5,374
*TYPE 171	Bristol	1	Leonides	550 b.h.p.	5 passengers	1	—	—	11 8	49 0	100	5,200
†GYRODYNE	Fairey	1	Leonides	515 b.h.p.	3 or 4 passengers	Pilot	—	19 2	10 1	52 0	110	4,800
SKEETER	Cierva	1	‡Jameson	100 b.h.p.	Pilot and passenger	—	—	—	—	—	—	—

* Tail rotor dia. 9 ft. 7 in. † Disc area 2,123 sq. ft. ‡ Production series will have a more powerful engine.

SECTION XXXI

PART II

AERO ENGINES

Aero engines are internal combustion engines of either the piston or turbine type. The piston engine, both air and liquid cooled, with spark ignition has been the most widely used type to date. Its rival over a long period of years has been the compression ignition engine, but where high power and light weight are required—as they very commonly are—the spark ignition type has always led. The struggle is not yet over but its importance has become less with the rapid advance in recent years of the internal combustion turbine engine (see p. 448). The piston type of aero engine is not likely to survive for powers exceeding 4,000 h.p. and some experts incline to put the figure lower than that.

Because of the conditions of flight, the aero engine has differed from other I.C. engines by reason of the demands for: (1) great power in a single unit; (2) small volume and low head resistance; (3) low weight; (4) great reliability; (5) high power output for a short time at take-off; (6) high percentage of power at high altitude and therefore low air pressure and temperature. Designers have been able to satisfy these conflicting demands to a remarkable degree. Engines developing 3,500 h.p. from 24 cylinders are available.

Piston Engines.

The cylinders of piston engines are arranged either *radially* or *in line*. The *radial* arrangement used only for *air-cooled* types, may have 3, 5, 7, 9, 14 or 18 cylinders with powers ranging from 50 to over 2,000 h.p. Up to 9 cylinders the engine consists of one row but the 14- and 18-cylinder engines consist of two rows with the rear row staggered behind the front row for better cooling.

The radial arrangement is not used for *liquid cooled* engines, these being always in *in-line* banks of up to 6 with the axis of the bank parallel to the direction of flight to reduce head resistance to a minimum. The bank may be arranged with the cylinders vertical and above the crankcase, in which case it is referred to as an *upright* engine, or they may be under the crankcase, an *inverted* engine. This type has the advantage of greater visibility for the pilot. Other ways of arranging the banks are in a 'vee' (usually 60°, upright or inverted) an 'H' (with two crankshafts), a 'double vee' (also with two crankshafts, included angle 150°), or an 'X.'

The *in-line* bank of cylinders may also be used for *air-cooled* engines with either 4 or 6 cylinders per bank. There may be either one, two or four banks arranged in the upright, inverted, inverted 'vee,' 'H,' or opposed flat systems.

Mechanical Efficiency.

Direct determination of mechanical efficiency by the simultaneous measurement of both the indicated and the brake horse-power is not practicable. Accurate phasing is so extremely difficult that any attempt to obtain the indicated mean effective pressure from the card is in practice quite impossible.

The following method is therefore employed: an accurate measurement of the total losses, both friction and pumping losses, is obtained when motoring the engine immediately firing has ceased so that temperature and other conditions are approximately identical to the conditions when the engine is running under its own power. This figure for the losses is then added to the observed brake horse-power of the engine and the sum quoted as the indicated horse-power. The figure obtained for i.h.p. is then used for the determination of the mechanical efficiency of the engine.

The mechanical efficiency of a radia engine is higher than that of an in-line (or vee) type engine as shown in Table I.

TABLE I.—MECHANICAL EFFICIENCY.

Type of Engine.	Mechanical Efficiency.
	Per Cent.
Radial direct drive, normally aspirated	92·0
Radial, geared, normally aspirated	89·5
Radial, geared, fully supercharged	83·0
Vee, direct drive, normally aspirated	89·5
Vee, geared, normally aspirated	87·0
Vee, geared, fully supercharged	80·5

The total losses of an engine may be divided into (*a*) frictional losses, (*b*) pumping losses; typical figures being given in Table II, the losses being expressed as the equivalent brake mean effective pressure.

TABLE II.—SUB-DIVISION OF TOTAL LOSSES.

Engine Speed. R.P.M.	Pumping Loss. Lb. per Sq. In.	Piston Friction. Lb. per Sq. In.	Bearing Friction and Auxiliaries. Lb. per Sq. In.	Total Loss. Lb. per Sq. In.
1,000	2·5	5·5	1·0	9
1,500	4·0	8·5	1·5	14
2,000	5·0	12·0	2·0	19

Thermal Efficiency.

Strictly, thermal efficiency means the efficiency with which the heat actually generated by combustion in the cylinder is converted into mechanical work.

For practical purposes, however, thermal efficiency is taken as the ratio of the heat equivalent of the work done by the engine to the heat supplied or, expressed in another way, as the equivalent of the inverse of the fuel consumption per horse-power. This definition has been standardised in the report of the Heat Engines Trials Committee of the Institution of Civil Engineers wherein the brake thermal efficiency is defined as follows :—

$$\text{Brake thermal efficiency} = \frac{100 \times \text{B.H.P.} \times 2{,}546}{(\text{Calorific value (gross) of the fuel} \times \text{fuel consumption lb. hr.})}$$

The Committee adopted the higher (or gross) calorific value of the fuel as a figure to be used in all calculations, since the gross value is a direct experimental determination whereas the lower (or net) value is the gross value less a quite arbitrary deduction.

The effect of using the gross value on the figure of thermal efficiency is given in the report as follows :—

	Thermal Efficiency.	
	On Net Value.	On Gross Value.
	Per Cent.	Per Cent.
Internal combustion engine	35	32

The calorific values and latent heat of various fuels are given in Table III.

TABLE III.

Fuel.	Specific Gravity at 15° C.	Gross Calorific Value.		Latent Heat of Evaporation. B.Th.U. Lb.
		B.Th.U. Lb.	B.Th.U. Imp. Gal.	
Aviation gasoline	0·704–0·782	19,500–20,100	142,000–152,250	140
Motor benzole .	0·880	18,250	157,000	168
Methyl alcohol .	0·829	9,900	82,100	500
Ethyl alcohol .	0·798	11,800	94,500	406

Standard aviation gasoline to Air Ministry Specification D.T.D. 224 has an average gross calorific value of 20,000 B.Th.U. lb. (net value 18,750 B.Th.U. lb.) and a modern aircraft engine at normal power and speed having a fuel consumption of 0·55 lb. b.h.p. hr. gives a brake thermal efficiency of 22·5 per cent. (or reckoned on the net value of the fuel 24·7 per cent.).

The Ideal Air Cycle.

The efficiencies of the standard engine using air as the working agent and following the ideal constant volume cycle are given in Table IV and the theoretical limits for the thermal efficiency and i.m.e.p. in Table V. These efficiencies are usually used when comparing the efficiencies obtained on test of actual engines.

TABLE IV.—AIR STANDARD EFFICIENCY.

$$n = 1 - \left(\frac{1}{r}\right)^{\gamma-1}$$

Ratio of Compression. r	Thermal Efficiency. $1-\left(\frac{1}{r}\right)^{0\cdot396}$
2	0·240
3	0·345
4	0·424
5	0·473
6	0·509
7	0·539
10	0·600
20	0·695

TABLE V.—IDEAL FIGURES FOR THERMAL EFFICIENCY AND INDICATED MEAN PRESSURE.

Compression Ratio.	Limiting Thermal Efficiency. Per Cent.			Limiting Indicated M.E.P. Lb. per Sq. In.		
	15 per Cent. Weak.	Correct.	20 per Cent. Rich.	15 per Cent. Weak.	Correct.	20 per Cent. Rich.
4·0	29·0	27·9	28·3	119·0	133·0	135·5
4·5	31·2	30·0	30·5	127·5	143·0	147·0
5·0	33·0	31·8	32·3	135·5	152·0	156·0
5·5	34·6	33·3	33·9	143·0	159·0	163·5
6·0	36·0	34·7	35·3	148·0	163·5	171·0
6·5	37·3	35·9	36·5	153·0	171·5	176·0
7·0	38·5	37·1	37·7	158·0	177·0	182·0
7·5	39·6	38·1	38·8	163·0	182·0	187·0
8·0	40·5	39·0	39·7	166·5	186·0	191·0

(*Engines of High Output*—H. R. Ricardo.)

The Influence of Compression Ratio in Piston Engine Combustion.

The thermal efficiency is greatly affected by the compression ratio and as a general rule the highest ratio that can be satisfactorily employed should always be used.

The quantity of residual gas left in the clearance space diminishes as the ratio is increased. Since the presence of exhaust gas in the cylinder reduces the rate of combustion, a reduction of the amount left in the clearance space has the effect of increasing the combustion rate. The reduction of the quantity of hot exhaust gas also reduces the rise in compression temperature with increase of compression ratio. The temperature of the residual gas may be taken as 850° C. and the temperature at the end of compression at different ratios after allowing for the proportion of residual exhaust gas, considering the swept volume of the cylinder to be 1 cu. ft., are given in Table VI.

TABLE VI.

Compression Ratio.	Volume Residual Gas at 850° C. Cu. Ft.	Vol. at N.T.P. Cu. Ft.	Vol. of Charge. N.T.P.	Temp. of Mixture before Compression. °C.	Max. Temp. of Compression. °C.
4	0·33	0·08	0·82	139	376
5	0·25	0·061	0·82	122	399
6	0·20	0·049	0·82	111	419
7	0·166	0·04	0·82	102	437

(*Internal Combustion Engine*—D. R. Pye.)

The effect of compression ratio on power, maximum pressure, thermal efficiency and heat to cooling water is shown in fig. 1.

THE TURBINE ENGINE.

As the burning gases in the piston engine are employed to rotate an airscrew so as to produce a large diameter air jet, the dynamic reaction from which propels the aircraft, it is natural to ask whether the burning gases could not produce a suitable jet for this purpose without requiring the intervention of an airscrew. That question has been asked—and answered, largely by the efforts of Sir F. Whittle. The burnt gases from the combustion chamber, with or without an admixture of air, can provide a jet which, though much smaller and much faster than that from an airscrew, will give the necessary thrust for propulsion, and continue to do so at air speeds so high as to embarrass any airscrew, by reason of the latter's grave loss of efficiency when the speed of sound is approached.

For metallurgical reasons there is at present a rather low temperature limit to the cycle of operations, and this leads to the fuel economy of jet-propulsion units being low at speeds of less than 400 m.p.h.; but when the speed rises to 600 m.p.h. the rate of fuel consumption becomes less than that of the piston engine. Greater economy at lower speeds is obtainable by taking much more power out of the turbine than is needed to drive the air compressor, and using the balance to add an airscrew jet to the now reduced one from the exhaust. It must be remembered that the power required by the air compressor is usually far more than the net power produced by the unit, but that since this power flows in a closed circuit the only loss is that due to mechanical inefficiency. Close study of mechanical details is therefore amply repaid.

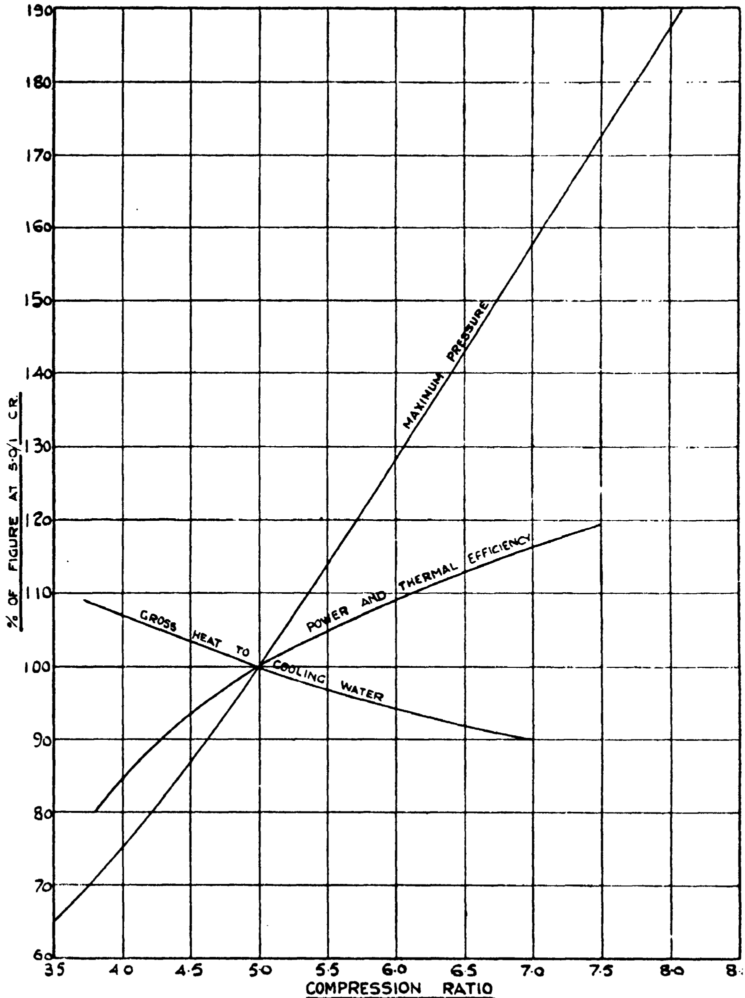
One of the unsolved problems is whether the centrifugal compressor or the axial compressor is the better. Both are in use. The former is easier to design and build but, as the latter leads to a less overall diameter and therefore to lower head resistance, it is likely to be preferred.

The turbine engine normally works on the constant-pressure-cycle, just as the normal piston-engine operates on the constant-volume-cycle, and it might be thought that the same ideal thermal efficiency formula could hardly apply to both. As it happens, however, the same formula does apply to both and is that given in Table IV on p. 447; moreover, the efficiency so calculated is identical with the ratio of the temperature rise on compression to the final compression temperature, a useful result readily applied to the turbine engine, and showing directly the great importance of as high a compression ratio as can safely be allowed.

At present this ratio is limited by metallurgical considerations in turbine and combustion chamber design, consequently the thermal efficiency is lower than that of a piston engine and the fuel consumption higher. The turbine is, however, less affected by high altitudes, and in the jet type has no airscrew to suffer from the effect of an approach to sonic speeds. Hence for very high speed and high altitude work the turbo-jet combination is highly efficient. An intermediate type is the turbine applied to an airscrew drive, and this may become popular for civil transport types owing to the combination of reasonable economy with a welcome freedom from vibration and bothersome noise.

A curious type of which more may be heard in the future is the Ram-Jet Engine (or 'athodyd'). This makes use of the principle that at exceedingly high air speeds the intake air is so effectively

EFFECT OF COMPRESSION RATIO ON POWER, THERMAL EFFICIENCY, MAXIMUM PRESSURE AND HEAT FLOW TO COOLING WATER OF PISTON ENGINES.



(H. R. Ricardo.)

FIG. 1.

compressed by being 'brought to rest' relatively to the aircraft that no mechanical compressor (or turbine for driving it) is needed and the engine consists solely of a combustion chamber and an exhaust jet. It is simple and cheap, and may well prove useful in certain small pilotless aircraft of military type. It is simpler even than the rudimentary flap-type impulse engines fitted to the 'flying bombs' of the 1939–45 war.

Volumetric Efficiency.

There are two ways of considering the volumetric efficiency of a piston engine:—

(a) The ratio of the volume swept by the piston to the volume of mixture drawn in per stroke in the pressure and temperature of the surrounding air;

(b) The ratio of the mixture drawn in per stroke to the quantity that would be contained in the swept volume of the cylinder at N.T.P.

Neglecting the effect of the exhaust gas in the clearance space by definition (b) above

$$\text{Volumetric efficiency} = \frac{273}{\text{Temp. °C. absolute of the induction charge}}$$

In theoretical calculations this definition should always be used, but for practical engine tests the Heat Engine Trials Committee have standardised definition (a) and the volumetric efficiency is calculated as follows:—

$$\left.\begin{matrix}\text{Volumetric}\\ \text{efficiency}\end{matrix}\right\} = \frac{\text{Vol. of inlet air per min. at the temp. and pressure of the engine air intake}}{\text{Total swept vol. of all the power cylinders per min. during the power stroke.}}$$

In addition to the limitations imposed by valve area, valve timing, compression ratio, engine speed and the form of induction system, the volumetric efficiency is dependent on variable factors such as cylinder temperature, latent heat of evaporation of the fuel, mixture strength, and the heat received by the mixture from the induction system. It is therefore difficult to give an average figure, but for a normally aspirated engine the volumetric efficiency should lie between 75 per cent. and 85 per cent.

The *Adiabatic Efficiency* of a blower is the ratio of the power theoretically required for the adiabatic compression of a given weight of air between specified pressure limits to the power input to the air during compression, assuming that no frictional losses occur.

M = Weight of air lb. minute.
J = Joule's equivalent (1,400 ft. lb. per centigrade heat unit).
C_p = Specific heat of air at constant pressure (0·238).
γ = The ratio of the specific heats of air (1·41).
T_1 = Absolute temperature of air (° C.) at intake.
T_2 = " " " " " " delivery.
P_1 = " pressure of air at intake.
P_2 = " " " " " delivery.

$$\frac{J \times C_p}{33{,}000} = 0{\cdot}0101 \quad \frac{\gamma - 1}{\gamma} = 0{\cdot}291$$

Power for Adiabatic Compression.

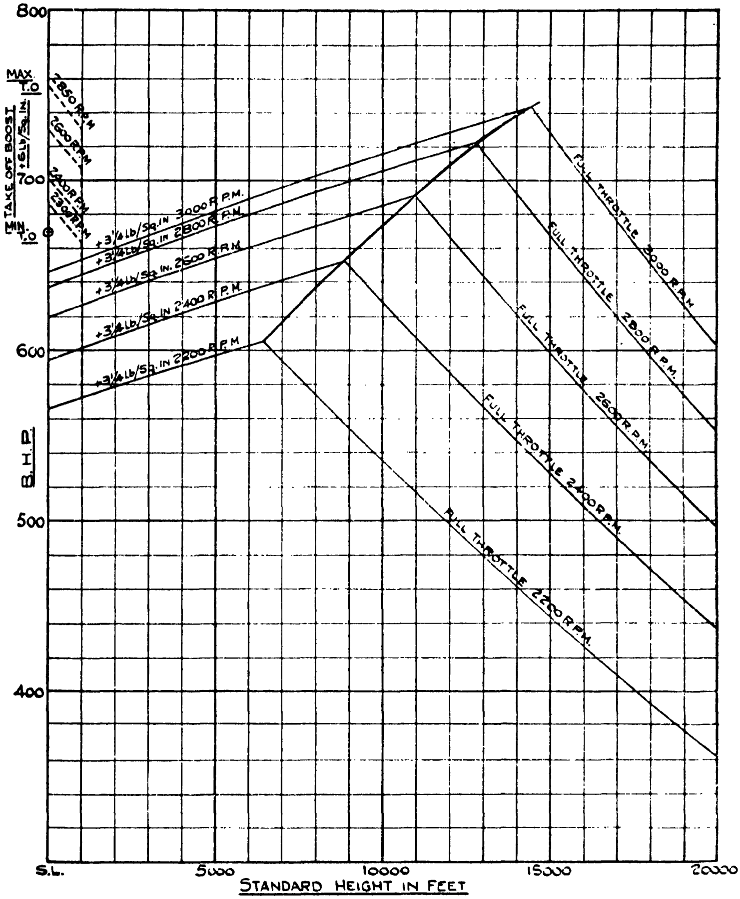
$$\text{Adiabatic H.P.} = \frac{M \times J \times C_p \times T_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{33{,}000}$$

Detonation.

Ricardo defines detonation thus: 'The phenomenon of detonation appears to be the setting up in the cylinder of an explosion wave. This occurs when the rapidity of combustion of that portion of the working fluid first ignited is such that, by its expansion, it compresses before it the unburnt portion beyond a certain rate. When the rate of temperature rise due to compression by the burning portion of the charge exceeds that at which it can get rid of its heat by conduction, convection, etc., by a certain margin, the remaining portion ignites spontaneously and nearly simultaneously throughout its whole bulk, thus setting up an explosion wave which strikes the walls of the cylinder with a hammer-like blow.'

Detonation should not be confused with 'pre-ignition,' which is a premature explosion of the charge caused by an excessively hot spot in the cylinder such as a glowing piece of carbon deposit. Detonation is affected by the maximum flame temperature, the compression pressure and the proportion of diluent exhaust gas left behind in the cylinder.

KESTREL XVI (FULLY SUPERCHARGED PISTON ENGINE). ENGINE POWER AT CONSTANT R.P.M. AT ALTITUDES.



(*Rolls Royce Ltd.*)

FIG. 2.

Aviation Fuel.

Aviation fuel can be divided into two classes: volatile and non-volatile. The former class comprises the petrols and the latter the Diesel oils. Both are composed of a complex mixture of hydrocarbons. Petrol is a distillate of crude petroleum boiling between 140° and 400° F. containing three series of hydrocarbons; paraffins, naphthenes and aromatics. Aromatics are the most desirable, naphthenes next, while the paraffins are worst. Resistance to detonation is the most important characteristic of fuel. An average aviation fuel distilled from petroleum would contain about 50 per cent. paraffins, 30 aromatics and 20 naphthenes. As well as from petroleum, aviation fuel can also be made from coal.

The constant search for methods of improving fuel economy and increasing the power of engines has called for higher compression ratios and greater boost pressures, these in turn necessitating higher resistance to detonation. The 'octane number' of a fuel is a measure of its resistance to detonation and is defined as 'the percentage by volume of iso-octane (2, 2, 4 tri-methyl pentane) in a mixture of iso-octane and normal heptane which matches the fuel in anti-detonation value when tested by the standard C.F.R. Motor Method.'

The demand for high octane fuels has been more than the ordinary distillation products of petroleum could satisfy in quality and it has been found that fuels above 87 octane could best be made by breaking down the petroleum molecule into gaseous hydrocarbons, rebuilding them into high anti-detonation fuels and adding tetra ethyl lead. Experimental work had established the fact that the addition of this substance in small quantities greatly raised the resistance to detonation. The gaseous hydrocarbons may come from gas wells, oil wells, the distillation of crude, or the 'cracking' processes. Fuels may be developed by blending with base gasoline, or using catalytically-cracked or hydrogenated gasoline. Fuels of 90 octane can be made by adding T.E.L. to certain types of aviation base gasoline, but fuel of higher octane must include a blending agent such as iso-pentane or iso-octane. The use of leaded fuels is now almost universal for aero engines, the only ones which do not call for it being some of the smaller engines of about 100 h.p. which are designed to run on first quality unleaded motor spirit of about 70 or 74 octane.

The oil industry has become chemically a most complicated affair in the last ten years and aviation fuel has advanced a long way from being just a distillation product of petroleum. Alkylation is an important modern development. The following are extracts from a paper published in 1941 by H. R. Tate of the Standard Oil Development Company:

'The research laboratories of the oil industry have found that the branched-chain paraffins—iso-paraffins—more nearly approach the ideal aviation fuel than any type of hydrocarbons occurring in crude oil or capable of being synthesised in large volumes from refinery raw materials. The iso-paraffins in the aviation gasoline boiling range are characterised by high calorific value, high clear octane numbers, good lead susceptibility, excellent chemical stability, and freedom from gum deposition in storage or in the induction system of aircraft engines. The classic example of iso-paraffins is iso-octane, the primary reference standard for octane ratings. Blends of technical iso-octane and straight-run or hydrogenated gasolines in the form of 100 octane fuels have been produced in increasing quantities. In 1937 approximately 170,000 barrels of 100 octane aviation gasoline were produced and it is estimated that by the end of 1940, the yearly production of such fuels containing iso-octane will reach 4 or 5 million barrels.

'Oil technologists were not content with the cold and hot sulphuric acid and other polymerisation processes for making technical iso-octane . . . Research was therefore directed toward the coupling of the olefin with an iso-paraffin to produce a branched chain, saturated paraffinic compound. Previous work had shown the possibility of this direct coupling—alkylation—by means of metallic halide catalysts, but the catalyst costs were too high. The research staffs . . . worked out processes whereby the coupling . . . was effected by the catalytic action of sulphuric acid. The alkylation process is a commercially-proven method. . . . Nineteen commercial plants are in operation in U.S.A. . . . The total output is approximately 6,800,000 barrels per year of aviation alkylate.

'Present indications are that 100 octane is not the ceiling for high-output engines. Fuels of over 100 octane can readily be made from butene alkylates plus moderate concentrations of T.E.L. The 300° F. end-point butene alkylates with 3 c.c. T.E.L. per U.S. gal. have octane numbers well over 100 (one example being equivalent to iso-octane plus 0·52 c.c. T.E.L. per U.S. gal.).'

A typical 100 octane fuel made from butene alkyate blending agent and a 74 octane straight-run gasoline shows end points of 105° and 270° F. The amount of alkylate required varies from 50 to 70 per cent. depending on the octane rating of the base. (The octane rating used in U.S.A. is determined by the A.S.T.M. method or the Army method. In the above extracts from the paper, the rating is A.S.T.M.)

During the year 1939, before the outbreak of war, 87 octane fuel (DTD 230) was the standard for British military engines in combat aircraft, but some squadrons were operating experimentally with 100 octane. During 1940 the Merlin-engined Spitfire and Hurricane fighters used 100 octane exclusively. As the war went on, bombers started to use it for take-off and, though no official statement has been made, it is likely that more and more extensive use was made of it for continuous operation.

TABLE VII.—SPECIFICATIONS FOR AVIATION FUELS.

Country	Great Britain.		America.		France	
Specification No.	D.T.D. 224	D.T.D. 230	Y–3,557–G.	100 Octane blend.'	3401/2.A.A.	3401/2.A.B.
Minimum Octane No.	77	87	92	100	70	85
Method	C.F.R. Modified Motor Method.	C.F.R. Modified Motor Method.	U.S. Army.	U.S. Army.	C.F.R. Modified Motor Method.	C.F.R. Modified Motor Method.
Maximum lead concentration T.E.L. ml. Imperial gallon	Nil.	4	7	3·6	Nil.	3·6
Ml. litre	Nil.	0·88	1·54	0·79	Nil.	0·79
Specific gravity maximum	0·79	0·79	—	—	0·78	0·78
Distillation Range:						
10 per cent. recovery	Below 75° C.	Below 75° C.	Below 75° C.	Below 75° C.	Below 75° C.	Below 75° C.
20 „ „ „	—	—	—	—	Above 75° C.	Above 75° C.
50 „ „ „	Below 100° C.	Below 100° C.	Below 100° C.	Below 100° C.	Below 100° C.	Below 100° C.
90 „ „ „	„ 150° C.	„ 150° C.	„ 135° C.	„ 135° C.	„ 150° C.	„ 150° C.
Final boiling point maximum	180° C.	180° C.	—	—	180° C.	180° C.
Reid vapour pressure maximum at 100° F.	7 lb. sq. in	7 lb. sq. in.	7 lb. sq. in.	7 lb. sq. in.	0·5 kg. sq. cm.	0·5 kg. sq. cm.
Freezing point	Below — 50° C.	Below — 60° C.	Below — 60° C.	Below — 60° C.	Below — 45° C.	Below — 45° C.
Actual gum—maximum	10 mgms. per 100 c.c.	10 mgms. per 100 c.c.	—	—	6 mgms. per 100 c.c.	6 mgms. per 100 c.c.
Potential gum—maximum	10 mgms. more than actual gum	10 mgms. more than actual gum	10 mgms. per 100 c.c.	10 mgms. per 100 c.c.	—	—

TABLE VIII.

Fuel Specification.	Octane No. C.F.R. British Air Ministry Method.	Engine Output. B.H.P. Litre.	Engine Weight. Lb. B.H.P.	Fuel Consumption. Lb. B.H.P. Hr.
D.T.D. 134	73	20·5	1·76	0·55
D.T.D. 224	77	22·7	1·58	0·53
D.T.D. 230	87	28·0	1·29	0·49
—	100	35·0	1·05	0·45

A usual fuel for air transport use is 87 octane, though it is likely that higher quality will be used in increasing proportion in the immediate future.

Fuel quality affects power very much. By changing from 87 to 100 octane fuel, the take-off power of the Merlin X was raised by 20 per cent. On a Merlin II the B.M.E.P. of 165 lb./sq.in. with 87 octane could be raised to 205 with 100 octane.

Ethyl Fluid.

Tetra Ethyl Lead $Pb(C_2H_5)_4$ has the following characteristics:—

Specific gravity at 20° C.	1·659
Boiling point	200° C. (with decomposition)
Freezing point	− 156° C.

The ethyl fluid at present used for aviation purposes is known as '1.T.Mix' and is coloured blue. It has the following composition:—

Tetra ethyl lead $Pb(C_2H_5)_4$	61·42 per cent. by weight.
Ethylene dibromide	35·68 „ „ „ „
Dye	0·17 „ „ „ „
Kerosene and impurities	Balance
Specific gravity of fluid at 20° C.	1·755

The proportion of T.E.L. in ethyl fluid, by volume, is 65·5 per cent.

Ethyl fluid is completely soluble in petrol and will not separate out under any normal storage conditions. Some idea of the value of tetra ethyl lead in suppressing detonation will be gathered from the fact that the addition of the first cubic centimeter (1·0 c.c.) to a gallon (4,545 c.c.) of petrol can increase the anti-knock value of that petrol by as much as 15 octane numbers, the actual amount depending on the characteristics of the particular petrol. The maximum allowable concentration of lead in any fuel for commercial purposes is limited to 3·6 c.c. per Imperial gallon although for military or other government purposes this limit is raised to 7 c.c. Imperial gallon. The addition of lead to petrol does not sensibly alter any of the characteristics of the spirit and the lead has no effect until combustion commences.

In quoting the concentrations in a petrol it is always the amount of tetra ethyl lead that is given and not the amount of fluid, since it is the lead that is the active anti-knock agent. Although tetra ethyl lead is poisonous, in its diluted state in petrol there is no danger whatever, and fuels containing lead up to the maximum permitted concentration can be handled in the same manner as ordinary petrols.

Fuel of High Flash Point.

Research has been directed towards developing a fuel of high flash point in order to reduce fire risk after accident. This has been loosely called 'safety fuel.' Such fuels have been produced with boiling points in the 300° to 425° F. range and flash points of 110° or 115° F. The heating value and anti-detonating qualities approximate to those of normal fuels; octane ratings up to and greater than 100 have been attained in them. High flash fuels may be of the aromatic type or branched-chain paraffins.

Vapour Lock at High Altitude.

Boiling of the petrol with the formation of vapour lock in the fuel system is one of the problems which high altitude flight brings in its train. Reduction of the atmospheric pressure at altitude

allows the fuel to vaporise at much lower temperatures than at sea-level pressures. To combat this it is necessary to use a fuel with a low vapour pressure, increase the pressure within the fuel tank, or keep the temperature of the fuel low. The first method is not promising as an auxiliary fuel of high vapour pressure would be required for starting and also because high octane fuels with low vapour pressure are not readily available.

The pressure inside the tank could be increased by applying a supercharge pressure or by 'self-supercharging.' This consists in closing the tank vent just after the fuel has started to boil and so generating a pressure which prevents it boiling at any higher altitude. With a fast-climbing fighter, the tankful of fuel does not cool as quickly as the aeroplane climbs, and this is very conducive to vapour lock. If no other steps are taken to prevent vapour lock, it may be necessary to cool the tank contents on the ground but so far this has not been attempted.

With a fuel of 7 lb./sq. in. Reid vapour pressure if the aeroplane left the ground at 38° C. and the fuel did not cool, boiling would start just below 20,000 ft. With a supercharge pressure on the tank of 4 lb./sq. in., boiling could be prevented up to 40,000 ft.

Cold Corrosion.

When engines are stored subsequent to running on leaded fuels, serious corrosion of the stee cylinders has, in certain circumstances, been found to occur. This corrosion varies with the absolute humidity of the atmosphere, and is therefore much more prevalent under tropical conditions.

A large number of compounds have been tested as inhibitors of this corrosion by the Ethyl Gasoline Corporation, and the following mixture, known as E.G. 174, has been found to be effective in all conditions:—

	Per Cent.
Commercial triethanolamine	5
Aluminium stearate	10
Normal butanol	10 to 12
Lard oil (3½–7½ per cent. free acidity as oleic acid)	Balance

The mixture is somewhat difficult to manufacture as it tends to be unstable and throws down solids unless manufactured with care. It can be obtained commercially.

High Temperature Liquid Cooling.

The outlet temperature of a normal water-cooled engine is limited by the boiling point of water, and when flying at about 5,000 feet this means a limitation to about 80° C. The rate of heat dissipation depends on the temperature difference between this outlet temperature and the temperature of the surrounding air. A comparatively small temperature difference means that a large radiator has to be used which imposes a substantial 'cooling drag' and anything that can be done to increase the value of the temperature difference will obviously reduce this drag since the necessary dissipation of the waste heat will be obtained from a smaller radiator.

In order to increase the temperature difference, high boiling-point liquids are now being used, and of these ethylene glycol is the most important. The characteristics of this liquid are given in Table IX.

TABLE IX.

ETHYLENE GLYCOL $C_2H_4(OH)_2$.

Density at 25° C.	1·11
,, ,, 100° C.	1·05
Viscosity at 25° C.	17·0
,, ,, 100° C.	0·020
Specific heat	0·675
Boiling point °C.	178
Freezing point ° C.	−17
Flash point ° C.	124
Latent heat at 100° C. . . .	240

The heat transfer coefficient of water is approximately 4·7 times that of ethylene glycol.

With the use of glycol in the engine jackets the actual metal temperatures increase not only due to the higher liquid temperatures employed, but also due to the relatively poor heat transfer

of glycol compared to water and to the much lower evaporative effect which, when water is in use, serves to safeguard any 'hotspots' in the cylinder or head. The use of glycol therefore introduces serious difficulties in engine design and the greatest care must be taken to provide adequate liquid flows and velocities if cracking and distortion of the engine structure are to be avoided. It is usual to limit the outlet temperature of the glycol to not more than 120° C. to 130° C., since experience has shown that an appreciable margin between the maximum temperature reached by the liquid and its boiling point must be maintained if trouble with hotspots in the engine is to be avoided.

The increase in the temperature difference, however, in a glycol cooled system does permit a much smaller radiator being used with an appreciable increase in aircraft performance, particularly at high flying speeds.

Evaporative Cooling.

In evaporatively cooled systems the circulating water is introduced into the cylinder jackets at the boiling point at any altitude, boiling being prevented by maintaining a pressure in the jackets and also by ensuring that the rate of circulation is kept high. The steam separator is at atmospheric pressure, and from here the water is returned to the engine and the steam led to the condenser where its latent heat is dissipated and the condensate returned to the cylinder An engine, the Rolls Royce Kestrel, has passed an official type test evaporatively cooled, and certain types of aircraft installation flew successfully in the 1930 decade.

On older types of machines of comparatively low wing loading, sufficient surface is available for the condensers and, therefore, almost dragless cooling is achieved, but the general performance of these machines is so low by modern standards that the complete elimination of cooling drag effects comparatively little improvement. In modern high performance aircraft the surface available for cooling is not only very limited but also very difficult to reach from the engine which involves serious complications in the system. Since there is almost no reduction of cooling drag if the steam is condensed in a conventional honeycomb radiator the evaporative cooling system cannot be considered promising for modern high performance aircraft, and no engines employing this system are at present in use in this country.

Pressure Cooling.

Pressure water cooling is a later way of obtaining a small radiator surface through the use of a high temperature coolant in the liquid form, the boiling point of water being raised by subjecting it to pressure. Water is a better coolant than glycol because : (1) Its viscosity is lower, its latent and specific heats are higher and its heat-transfer coefficient greater ; (2) it has less tendency to 'creep' through joints ; (3) it is more easily obtainable in an emergency than glycol.

A 30 per cent. solution of glycol is recommended so that freezing does not occur on the ground or during a glide with engine off. A pressure-cooled system is similar in all respects to a water-cooled system with the addition of a spring-loaded double-acting relief valve instead of a vent on the header tank. With an open vent the drop in boiling point with a 30 per cent. solution from sea level to 12,000 ft. is 12·5° C. but with a relief valve set to 15 lb./sq. in. above atmospheric the drop is only 6·5° C. At 25 lb./sq. in. it is 5° C. A pressure-cooled system is affected by altitude changes to a less degree than an open one. A maximum pressure of 20 lb./sq. in. is needed to give the same area radiator as for a glycol-cooled installation with an outlet temperature of 130° C.

Controlled Cooling.

Radial air-cooled engines are cowled with a cowling of the NACA or Townend ring type. The engine may develop its maximum power either at its maximum forward speed, or at a very low forward speed as in the take-off or climb. So if the cowling is designed for the former case, the engine is liable to overheat in the latter. It has therefore been found necessary to control the flow of air by means of 'gills' on the exit of the cowling. These restrict the exit during high speed flight and may be adjusted to give the correct amount of cooling air for any condition. A cooling fan may also be used.

The radiator of the *liquid-cooled* engine is enclosed in an air duct so designed that the stream of air is slowed down to about one-third of its velocity just before going through the cooling tubes. After emerging from these tubes the air increases in velocity by reason of the duct narrowing and it is speeded up to its original velocity as nearly as possible so that it emerges again into the main stream with a minimum of turbulence. This procedure is necessary in order to reduce cooling drag as the increasing speed of aircraft was making cooling very expensive in terms of drag.

The heat which is imparted to the air stream as it goes through the tubes causes it to expand and so increases its velocity. This sets up a 'jet effect' which gives a forward thrust tending to reduce the drag of the cooling system. This effect becomes increasingly important with increase in forward speed of aircraft and it may be that, with speeds somewhat over 400 m.p.h., cooling will be accomplished without drag.

Air Cooled v. Liquid Cooled.

Though in the smaller sizes, up to about 400 or 500 h.p., the air-cooled engine seems to be established to the exclusion of the liquid-cooled, in the larger sizes the struggle still continues and engines of both types are in everyday use. U.S.A. has concentrated most on air-cooled, while German and British development has been in both types.

The liquid-cooled has smaller head resistance, but its cooling system adds complication; therefore it has been widely adopted in fighters and not widely adopted in airliners. It might seem that engines of greater power could be built as liquid-cooled units for it is difficult to see how more then 18 cylinders could be arranged radially and cooled by air. The air-cooled engine has the advantage in weight per h.p. as reference to Table XI shows, but at full throttle the liquid-cooled has a better fuel consumption. This advantage, however, does not hold so much at cruising conditions. Power required to cool the engine is approximately equal for the two types at about 3 per cent. of take-off power, though for the most modern types it is more like 2 per cent. From this brief summary it will be seen that neither type has a clear cut supremacy over the other in the larger sizes of engine from 500 to 1,500 h.p.

Installation Weights.

The following weights per h.p. of engine and installation come from a paper published in 1941 by John G. Lee, of United Aircraft Corporation. They refer to the larger size of aero engine in about the 800 to 1,200 h.p. range.

TABLE X.—ENGINE INSTALLATION WEIGHTS.

	Air-cooled.	Liquid-cooled.
Weight of engine installation:		
Airscrews and controls	0·302 to 0·320	0·302 to 0·320
Starting system (less batteries)	0·020 „ 0·045	0·020 „ 0·045
Engine mounting	0·045 „ 0·080	0·045 „ 0·080
Cowling	0·063 „ 0·100	0·063 „ 0·100
Oil system (including tanks)	0·040 „ 0·060	0·040 „ 0·060
Fuel system (less tanks)	0·020 „ 0·040	0·020 „ 0·040
Exhaust system	0·027 „ 0·040	0·027 „ 0·040
Radiators and coolant	— —	0·275 „ 0·300
Miscellaneous (ducts, controls, etc.)	0·015 „ 0·040	0·015 „ 0·040
Total normal installation weight	0·532 „ 0·725	0·807 „ 1·025
Dry weight of engine	1·26 „ 1·35	1·10 „ 1·26
Engine and installation—lb. per h.p.	1·792 „ 2·075	1·907 „ 2·285

Other items. Intercoolers and ducts may weigh 0·090 to 0·095 lb. per h.p.; turbo-superchargers, 0·109 to 0·132.

Supercharging.

The power of an engine falls off with decrease of atmospheric density since the quantity of charge entering the cylinder each stroke is proportional to the density. So the power of an aero engine falls off with increase of altitude unless it is supercharged.

Supercharging consists in fitting the engine with a pump which forces the explosive charge into the cylinders under a pressure greater than atmospheric. Superchargers were first fitted in order to maintain sea-level power at altitude but they were soon used to increase sea-level power by

increasing the manifold pressure above sea-level atmospheric pressure. Take-off powers with a manifold pressure of 6 lb. per sq. in. and more are now quite common.

Superchargers may be divided into two main types: the centrifugal blower and the positive displacement blower (the Roots type and the vane type). As far as modern aero engines are concerned, the centrifugal supercharger is fitted to the exclusion of all other types. It may be gear-driven from the engine, driven by a fluid drive or driven by a turbine run by the exhaust gases. Before 1941 no exhaust turbo-supercharger had been put into production and all superchargers had been gear-driven.

The centrifugal supercharger consists of a rotating impeller (running at speeds which may be between 15,000 and 25,000 r.p.m. in different engines); a casing which encloses it and contains diffuser blades; and a collector duct. The air enters at the impeller duct and is given a very high tangential velocity by the impeller. As the air passes through the fixed diffuser blades the kinetic energy is converted into pressure energy.

A supercharged engine is designed to deliver its maximum power at a certain altitude referred to as its 'rated altitude.' This is, of course, full throttle power; above this altitude the full throttle power falls off just as for an unsupercharged engine. Below the rated altitude full throttle cannot be used as it would result in a manifold pressure being built up which would be too great for the engine. (It might be too great because it would cause detonation or because the power developed would be too great for the structural strength of some of the parts or because the cooling system could not deal with the excessive quantity of heat generated.) So below the rated altitude the engine must be throttled to keep the manifold pressure to the maximum allowable. This results in the sea-level power usually being less than the power at rated altitude.

Owing to the power taken to drive the supercharger and due also to the fact that there is a rise in temperature of the air during compression, a supercharged engine at its rated altitude delivers at the same crankshaft speed and manifold pressure less power than a similar unsupercharged engine at sea level, although this loss in power is offset by the gain due to the increase of the difference between the exhaust back pressure and the induction manifold pressure at altitude. A supercharger capable of maintaining sea-level power at 12,000 ft. absorbs nearly 5 per cent. of the engine power on the ground.

The use of a take-off power, higher than the maximum continuous rated power at sea level, is allowed on the strict condition that it is only permissible for a short time, usually one or three minutes. For such a short time only can the engine deal with the greatly increased heat flow.

The first type of supercharger developed was geared to the engine through a train of gears causing it to run at several times engine speed (modern engines have ratios varying between 6 and 12). The higher the rated altitude of the engine, the greater would be the excess manifold pressure at sea level if the throttle were opened wide. So, for high-rated engines it became very desirable to run the supercharger slower at sea level to reduce the waste of power due to throttling. This called for a *two-speed supercharger*, necessitating a two-speed gearbox between engine and supercharger the lower supercharger speed being used at low altitudes.

High rated altitudes also called for another development, that of the *two-stage supercharger*. There is a limit to the pressure difference which it is efficient to handle in one compression stage, and as the increase of temperature in the compressed charge becomes very great it is necessary also to have an intercooler between the two stages.

The *exhaust turbo-supercharger* has been recognised for many years as the most desirable because the pressure-difference driving it (and therefore the power absorbed by it) is lowest at sea level and increases with altitude. Also it utilises some heat energy of the exhaust, much of which would otherwise be wasted. But the practical difficulties of making a turbine which would stand up to the very great temperatures of the exhaust gases has prevented this type appearing in production until 1941. But the Wright Cyclone engines of the high-altitude bomber, the Boeing Flying Fortress, have turbo-superchargers which have been developed by the General Electric Company in U.S.A. The exhaust turbo-supercharger is able to maintain sea-level power to a higher altitude than any other type and this is evident from the following extracts from a 1940 paper by Erold Pierce of the Wright Aeronautical Corporation:

'The two-speed supercharger rather falls by the wayside when any considerable altitude performance is desired. . . . For operation at 20,000 ft. or higher at rated power the two-speed engine (supercharging without intercooling) is entirely inadequate. . . . Below 25,000 ft., it would appear that the method of supercharging will depend upon the critical altitude for which operation is desired. In the range from sea level to 15,000 ft., the two-speed engine has the field almost to itself because of installation simplicity coupled with maximum power for take-off and reasonable power output at altitude. Above 15,000 ft. the choice of two-stage or turbo depends upon installation complexity and the availability of equipment. The increased power output of the turbo-supercharged engine compared with that of the two-stage engine appears most desirable and worth considerable installation complication.' Above 25,000 ft. he indicates that the exhaust turbo-supercharger is supreme.

Exhaust Valves.

The power developed by aero engines in production in 1941 exceeded 100 h.p. per cylinder and the exhaust valve had to be made capable of dealing with very great heat flows because of the exhaust gas being at temperatures of 850° or 1,000° C. Steels have been developed which are capable of continuous work at over 850° C., these temperatures being encountered in truck engines, but due to refinements in design aero engine valves can be kept down to a maximum of about 700° C. Typical exhaust valve temperatures are shown in fig. 3. Valves can now be obtained which have a life of over 5,000 hours in engines which have maximum r.p.m. of 3,000 and B.M.E.P. of 200 lb. per sq. in.

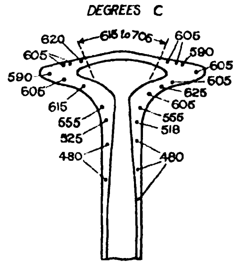


FIG. 3.—Exhaust Valve Temperatures.

This is due to three major developments: sodium cooling; development of the one-piece forging technique giving the hollow-head valve; and advances in cylinder head design. Three minor developments also contribute: use of austenitic steels; substitution of austenitic steel for bronze in the seat insert; and use of stellite facing on the valve seat.

Sodium is used in the hollow stem of the valve because it is a good conductor of heat, wets steel, is light, has no vapour pressure at operating temperatures, and melts at a low temperature (97·5° C.). Its boiling point is 880° C. and its heat conductivity about 9 times that of austenitic valve steel. Specific gravity is 0·95. In smaller valves 60 per cent. of the hollow space is filled with sodium, so allowing it to splash about and conduct heat from the crown to the stem and so to the valve guide. In larger valves 40 or 50 per cent. of the cavity is filled. Care must be taken in filling that the sodium is not contaminated, as any oxide of sodium will attack the steel. Before filling, the interior is carefully hand-polished to remove all scratches to prevent the start of fatigue cracks. After filling the end of the stem is swaged closed.

Improvements in forging technique have done much to reduce valve failures by keeping the grain of the metal in the right direction and much more is now known about valve shape design. The provision of a 'button' onthe crown of the valve is no longer good practice as it concentrates stress. Valve stems are usually nitrided and honed, with a tip of cobalt-chrome welded on the end. Such valves can handle 130 h.p. each. In marking valves no stamping is permissible as it may start a fatigue crack; only the electric pencil is allowed and only curved lines, no straight ones. Marking is done near the tip.

Austenitic steels have been adopted for exhaust valves because they have a high hot strength, good corrosion resistance and high impact value. KE 965 and TPA are two such steels. Their analyses are:

	C.	Mn.	P.	S.	Cr.	Ni.	Si.	Mo.	W.
KE 965 .	·35–·50	1·5 max.	—	—	12–16	10 min.	1–2·5	—	2–4
TPA .	·40–·50	·70 max.	·03 max.	·025 max.	13–15	13–15	·30–·80	·50 max.	1·75–3

Bronze seat inserts have been superseded for high-output aero engines by steel seats because of corrosion and loosening. It is necessary that the material of the insert have a high coefficient of expansion approximating to that of the aluminium alloy cylinder head so that loosening will not occur and so that the contact between the two will facilitate heat flow. The Firth NMC steel has a good resistance to corrosion and scaling at high temperature, is very tough and 'work

hardens' to a high degree. Other insert steels are Silcrome No. 9 and TPA. Their analyses follow:

	C.	Mn.	P.	S.	Cr.	Ni.	Si.	Mo.	W.
NMO	·5–·6	5·0	—	—	3·5	12	·5	—	—
Silcrome No. 9	·40–·50	·7 max.	·03 max.	·03 max.	13–15	13–15	2·75–3·25	·50 max.	1·75–3·0
TPA	·40–·50	·7 max.	·03 max.	·025 max.	13–15	13–15	·3–·8	·50 max.	1·75–3·0

The coefficient of expansion of NMO steel between 200° and 300° C. is 0·0000223.

Differential seat angle is a feature of modern design. To allow for changes of shape due to heating, the included angle of the valve face is made 1° greater than that of the faces of the insert.

Stellite is a material which has a very high degree of hot hardness and resistance to corrosion. It is used widely all over the world for facing both valves and inserts. Consisting largely of chromium and cobalt, Stellite No. 6 has this composition: C, 1·2; Cr 26·7; Si 2·7; W, 4·0; Co, 65·0 per cent. Another material, Brightray, is also used for the same purpose, this having the composition: C, 0·27; Cr, 19·75; Ni, 76·4; Fe, 2·2; Mn, 1·77 per cent.; with small amounts of others.

Stellite or Brightray is applied by oxy-acetylene flame to the valve or insert surface by a 'puddling' process which is more of a high temperature brazing process than a welding process. The melting point of Stellite is about 1,270° C.

In general, American design conforms to two valves per cylinder whereas both Rolls-Royce and Bristol have four, which of course leads to smaller size valves. The two-valve design leads to less complicated valve gear. A typical Bristol exhaust valve is made of KE 965 steel, sodium-cooled with 40 per cent. of the stem cavity filled with sodium. The small size of the valve prevents

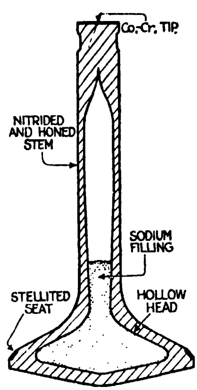


FIG. 4.—Cross Section of Modern Hollow-head Exhaust Valve

the adoption of a hollow head. The seat has ·040 in. of Stellite and the head of the valve has ·020 in. of a material similar to Brightray. American practice does not generally Stellite the seat inserts as they claim that Stellited valves work equally well with inserts with or without Stellite on them. Stellited inserts are common European practice. The most modern aero engines have enclosed valve gear and forced lubrication.

Cleaning Exhaust Valves.

To remove lead deposits and scale from the heads of exhaust valves after they have been in service, a method has been developed by the Ethyl Gasoline Corporation. This consists in electrolysing a fused mixture of 60 per cent. sodium hydroxide and 40 per cent. anhydrous sodium

carbonate by weight, using the valve to be cleaned as cathode. To prevent any reaction between oily material on the valve and the electrolyte, the valve must first be cleaned by heating in a flame or washing in petrol. The salt is maintained at from 430° to 480° C. by a burner or electric heat and a current of one-half ampere passed from a 6-volt storage battery. Time of immersion depends on the thickness of the deposit, but 10 minutes is usually sufficient, after which the valve is easily wire-brushed.

Inlet Valves.

Inlet valves do not have to operate at such high temperatures as exhaust valves, as shown in fig. 5. It is therefore not general practice to have them sodium-cooled, though this is occasionally

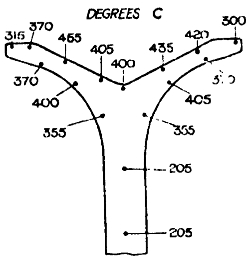


FIG. 5.—Inlet Valve Temperatures.

done. The hollow stem is general, however, for reasons of lightness, but the hollow head is not necessary. As well as KE 965 and TPA, other steels used are SAE 7260 and Silcrome No. 1 of the following composition :

	C.	Mn.	P.	S.	Cr.	Si.	W.
SAE 7260 . .	·5–·7	·3 max.	·035 max.	·04 max.	·5–1·0	—	1·5–2·0
Silcrome No. 1 .	·4–·5	·2–·6	·02 max.	·02 max.	8–9	3–5	—

It is both British and American practice to use aluminium bronze for the material of the inlet valve seat inserts. This has the composition of Cu, 89 ; Al, 10·5 ; Fe, 0·50 per cent. American valve guide material is a bronze consisting of Cu. 80 to 86 per cent. ; Al or Sn, 10 to 11 ; and other constituents.

Crankcases.

Crankcases may be made of alloys of aluminium or magnesium or, a new development for radial air-cooled engines in U.S.A., of steel. The small in-line air-cooled engines with powers up to about 200 h.p. frequently have crankcases of Elektron or other cast magnesium alloy. Larger engines, both in-line and radial, usually have crankcases of aluminium alloy. Two alloys suitable for casting are RR 50 and RR 53. Radial engine crankcases can also be made as forgings in two pieces out of aluminium alloys. But in U.S.A. forged aluminium alloy crankcases are going out of favour and it has been found that a steel crankcase can be made of the same weight. This has the advantage of higher fatigue strength and better strength at high operating temperatures.

Pistons and Piston Rings.

Pistons are die or sand castings or forgings of aluminium alloy and a late development in piston design to aid cooling is to have 'stalactites' of metal projecting downward from the crown of the piston. Piston speeds have increased greatly from figures such as 1,700 ft. per min. in 1921 to 2,600 in 1939. Now piston speeds of 3,000 ft. per min. in both air- and liquid-cooled engines are in common use. Hottest part of the piston is the centre of the crown and this can reach 300° or 350° C. in modern designs. Maximum gas pressure loads usually lie between 450 and 550 lb. per sq. in. Aluminium alloys frequently used are RR 53 and RR 59, and Y alloy. These contain Cu, Mg, Si, Ni, Fe.

Piston rings are of cast iron and the latest development is to coat them to reduce scuffing. Wear can be reduced by half by a plating of tin or a coating of iron oxide or phosphate or ferrous

sulphide. A usual design is to have three compression rings at the top of the piston and an oil control (scraper) ring at the bottom of the skirt. In the latest Wright 'Uniflow' design, the piston has five upper rings and one lower, this one being 'inverted' to act as a pump, so giving a good oil supply to the skirt. The oil thus pumped is conducted away through drainage holes to the inside of the piston. Oil consumption is reduced by this design.

Cylinders and Cylinder Heads.

Cylinder heads of both air- and liquid-cooled engines are of aluminium alloy such as RR 50 or RR 53 (for sand casting) or Hiduminium 59 or Y alloy (for forged and machined heads). Aluminium alloy is necessitated both for lightness and good heat conduction. Temperatures in an air-cooled head may range from 375° C. (exhaust valve seating) down to 100° C. Corresponding temperatures for liquid-cooled are 350° and 100° C. Adequate finning is most important to ensure cooling and advances have been made both in design and in casting technique to enable the very high power of the modern cylinder to be developed.

The barrels of air-cooled cylinders are usually machined from hollow forgings of Ni-Cr-Mo steel (DTD 228) suitable for nitriding for hardness to minimum value of 600 Vickers diamond. Connection between the barrel and the head is usually by screwing and shrinking, though the head may be held to the crankcase by long steel bolts.

The barrel of a liquid-cooled engine such as the Rolls-Royce Merlin is of high carbon steel and is of the 'wet' type, that is, is in contact with the coolant. A liquid-tight joint is made at the bottom with a rubber ring and a gas-tight joint at the top with soft aluminium. The assembly is held together with long bolts. A design with 'dry' cylinder liner may also be adopted.

Cylinder blocks for both types are of aluminium alloy castings such as RR 50.

Bearing Metals.

In early designs of aero engines the common bearing material, a tin base babbitt, was used but this has been superseded in high output engines by copper-lead and more lately still by cadmium-silver. Work is proceeding on the use of aluminium-tin alloys.

Tin base babbitt can be used up to a maximum permissible unit pressure of 1,000 lb. per sq. in. for standard quality bearings and 1,500 for higher quality. The maximum PV (rubbing factor) should not exceed 32,000 and 42,500 respectively; oil reservoir temperature 235° F. Composition: Sn 89, Sb 7·5, Cu 3·5, Pb 0·25 max. per cent.

High lead babbitt can be used up to 1,800 lb. per sq. in. with a maximum PV of 40,000 and oil reservoir temperature of 225° F. Composition: Pb 82 to 86, Sb 9 to 11, Sn 5 to 7, Cu 0·25 max. per cent. With both the babbitts minimum crankshaft hardness is not important.

Copper-lead can be used over 1,800 lb. per sq. in. with a maximum PV of 90,000 and upwards and oil temperature of 260° F. Composition: Cu 60, Pb 40 per cent. Minimum crankshaft hardness, 300 Brinell.

Cadmium-silver can be used over 1,800 and up to 3,850 lb. per sq. in. with maximum PV 90,000 and oil temperature of 260° F. Composition: Cd 98·75, Cu 0·50, Ag 0·75 per cent. Minimum crankshaft hardness, 250 Brinell.

Spark Plugs.

The conditions under which spark plugs have to operate have been constantly increasing in severity, particularly due to the use of leaded fuel, the increase of cylinder head temperature and the enclosing of the plug in a metal cover to 'screen' it. As mica is attacked by the products of combustion of leaded fuel, mica plugs are now superseded for high duty engines by ones with an insulating body of sintered aluminium oxide known as 'sinterkorund' or 'sintox.'

Sleeve Valve Engines.

The Bristol Aeroplane Co. has led the world in the development of the sleeve valve aero engine and has a range of air-cooled radials of various powers. These engines have a single sleeve which both rotates and reciprocates.

Advantages claimed for the sleeve valve design are:

1. By the elimination of the exhaust valve a 'hot spot' is removed, so enabling a higher compression ratio to be used (in some cases up to one unit higher) with fuel of the same octane rating.

2. A flatter consumption 'loop' is possible without running into troubles often associated with poppet valves at weak mixtures.

3. Its volumetric efficiency at high speeds is better than that of the poppet valve because its 'breathing' is better.

BRITISH PISTON AERO-ENGINES.

Name.	Makers.	Cylinder.			Maximum Power Rating.	Remarks.
		No.	Arrangement.	Cooling.		
LEONIDES 501–4	Alvis	9	Radial	Air-cooled	440–540 b.h.p.	
CHEETAH 15 and 25	Armstrong Siddeley	7	Radial	Air-cooled	405 b.h.p.	
GRENADIER	Blackburn (Cirrus Division)	6	Inverted in-line	Liquid-cooled	310 b.h.p.	
BOMBARDIER	Blackburn (Cirrus Division)	4	Inverted in-line	Air-cooled	180 b.h.p.	
CIRRUS MAJOR 3	Blackburn (Cirrus Division)	4	Inverted in-line	Air-cooled	155 b.h.p.	
CIRRUS MINOR 2	Blackburn (Cirrus Division)	4	Inverted in-line	Air-cooled	100 b.h.p.	
CENTAURUS	Bristol	18	Two-row radial	Air-cooled	2,550–3,100 b.h.p.	
HERCULES	Bristol	14	Two-row radial	Air-cooled	1,780–2,100 b.h.p.	
GIPSY QUEEN	de Havilland	6	Inverted in-line	Air-cooled	250–355 b.h.p.	
GIPSY MAJOR	de Havilland	6	Inverted in-line	Air-cooled	145–197 b.h.p.	
JAMESON	Jameson	4	Horizontally opposed	Air-cooled	106 b.h.p.	Not yet in production.
MONACO	Monaco	4	Horizontally opposed	Air-cooled	100 b.h.p.	
SABRE	Napier	24	H-type horizontal	Liquid-cooled	2,615–3,055 b.h.p.	
EAGLE 22	Rolls-Royce	24	Flat H-form layout	Liquid cooled	3,500 b.h.p.	
GRIFFON	Rolls-Royce	12	60° Vee layout	Liquid-cooled	1,990–2,350 b.h.p.	
MERLIN	Rolls-Royce	12	Upright Vee layout	Liquid-cooled	1,240–1,795 b.h.p.	

BRITISH GAS-TURBINE AERO-ENGINES.

Name.	Makers.	Type.	No. of Combustion Chambers.	Flow System.	Turbine.	Maximum Power Rating (Static).	At R.P.M.
PYTHON	Armstrong Siddeley	Turboprop.	11	Axial	Two-stage	3,670 shaft h.p. + 1,150 lb. thrust	8,000
MAMBA	Armstrong Siddeley	Turboprop.	6	Axial	Two-stage	1,010 shaft h.p. + 310 lb. thrust	15,000
PROTEUS	Bristol	Turboprop.	8	Axial and centrifugal	Single-stage	3,200 shaft h.p. + 800 lb. thrust	10,000
THESEUS	Bristol	Turboprop.	8	Axial and centrifugal	Single-stage	2,200 shaft h.p. + 590 lb. thrust	8,200
GHOST	de Havilland	Turbojet	10	Centrifugal	Single-stage	5,000 lb. thrust	10,000
GOBLIN	de Havilland	Turbojet	16	Centrifugal	Single-stage	3,000 lb. static thrust–3,300 lb. static thrust	10,200 (Goblin 2) 10,750 (Goblin 3)
BERYL	Metropolitan-Vickers	Turbojet	Annular	Axial	Single-stage	3,850 lb. thrust	7,750
NAIAD	Napier	Turboprop.	5	Axial	Two-stage	1,500 shaft h.p. + 241 lb. thrust	18,250
NENE	Rolls-Royce	Turbojet	9	Axial	Single-stage	4,500 lb. thrust–5,000 lb. thrust	12,000 (Nene 1) 12,300 (Nene 2)
DERWENT 5	Rolls-Royce	Turbojet	9	Axial	Single-stage	3,600 lb. thrust	14,700
CLYDE	Rolls-Royce	Turboprop.	9	Axial and single-stage centrifugal	—	3,020 shaft h.p. + 1,225 lb. thrust	6,000
DART	Rolls-Royce	Turboprop.	7	Centrifugal	Two-stage	1,000 shaft h.p. + 310 lb. thrust	14,500

4. It allows a much neater design than the poppet valve, particularly for the cylinder head, and the number of moving parts is smaller.

5. It is less susceptible to the troubles resulting from the use of leaded fuel and needs less maintenance between complete overhaul periods. (The poppet valve engine continues to make advances and is little if any behind the sleeve valve in these respects.)

Compression Ignition Engines.

The compression-ignition engine, properly developed, can have several advantages over the petrol engine. These are: (1) reduced fire hazard; (2) lower fuel consumption, which gives either greater range with the same payload or greater payload for the same range; (3) lower cost of fuel per gallon and, even more so, per mile; (4) absence of ratio interference due to ignition system; (5) absence of carbon monoxide in the exhaust gases; (6) freedom from carburettor icing.

The engine does not operate with the typical Diesel cycle of constant-pressure combustion but has a highly peaked indicator diagram rather like that of the spark-ignition engine. It should therefore not be called a Diesel engine but simply a 'compression ignition' engine. A comparison of the pressures in C.I. and S.I. engines (quoted by Maleev) follows:

	C.I.	S.I.
Compression ratio . . .	15	7·5
Maximum combustion pressure .	1,450 lb. per sq. in.	880
Mean effective pressure . .	112 „	135

The higher combustion pressure necessitates larger dimensions for some of the parts, thereby making the engine heavier. S.I. engines are now close to 1·0 lb. per h.p. at take-off, but C.I. engines are not likely to be less than 40 or 60 per cent. heavier. So their advantage in lower specific fuel consumption is offset by heavier engine weight.

Petrol Injection.

Petrol injection was adopted whole-heartedly by Germany just before the start of the war in 1939 and her combat types of aircraft were fitted with petrol injection systems instead of carburettors. There was one pump for each cylinder and this supplied the petrol to the cylinder through an injection nozzle. Advantages claimed for petrol injection are these: (1) more accurate metering of the fuel to each cylinder so giving better distribution and better specific fuel consumption. (2) A fuel of lower volatility can be used. (3) The system is free from the deposition of ice such as can occur in a carburettor. (4) Accelerations caused by aerobatics and air fighting have no effect on an injection system as they have on a carburettor.

As against these advantages the pumps have to be very accurately made as they meter out very small quantities of fuel and precision work of a high order is called for in making these reciprocating pumps which consist of a plunger about ¼ in. diameter in a barrel about 6 in. long. This has been the main obstacle to more widespread adoption of the system. A partial adoption of petrol injection is occurring in U.S.A. This is with a single pump injecting into the manifold as in the Stromberg injection carburettor. Small engines are also adopting it.

The 'Dynamic Damper.'

The *dynamic damper* fitted to the crankshaft of radial engines has done a great deal to reduce crankshaft stress as it reduces angular deflections to about one-tenth of the value they would have without it. The important angular vibration in a crankshaft is that due to the torque from gas pressure and it therefore has a frequency $\frac{N}{2} \times \frac{n}{60}$ per second, where N = number of cylinders in single row engine, and n = r.p.m of the engine. If this vibration happens to be in resonance with the natural frequency of the crankshaft, the angular deflection can be multiplied many times; for a 9-cylinder engine the stress may be 5 times as great as that resulting from a steady application of the maximum torque. This cannot be tolerated, particularly as the stress may be repeated as much as 1,000,000 times in 2 hours.

Mathematics indicated that it was necessary to find an 'absorber' with a frequency equal to that of the forcing vibration and with a restoring force varying as the square of the r.p.m. A simple pendulum attached to and forming part of the counterweight of the crankshaft has these

properties. As its weight is effective as part of the counterweight, the pendulum adds no weight to the engine. For a 9-cylinder engine, $\frac{r_1}{L} = (4 \cdot 5)^2$. This means that L must be very small (a

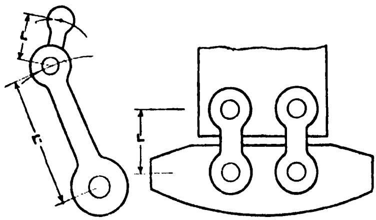


FIG. 6.—Principle of the Dynamic Damper.

fraction of an inch) which can be accomplished by hanging the pendulum on two links as indicated in fig. 6. Actually, in the design as evolved by the Wright Company, the pendulum part of the counterweight is supported on two rollers which are loose in their holes in both the pendulum and the crank cheek, so ensuring a very short pendulum length. This 'dynamic damper' was one of the most important developments of the late 1930's.

SECTION XXXII

RAILWAYS

PART I

PERMANENT WAY

RAILS — FISHPLATES AND BOLTS—SLEEPERS — BALLAST — MAINTENANCE — RAIL CHAIRS — ARRANGEMENT OF PERMANENT WAY — GAUGES — LIGHT RAILWAYS — SUPERELEVATION—CROSSINGS—CURVES (pp. 469–514)

(By C. D. Morgan, Permanent Way Assistant, Western Region, British Railways)

PART II

LOCOMOTIVES AND ROLLING STOCK

(Revised by H. Holcroft, Member of Council, Institution of Locomotive Engineers.)

LOCOMOTIVES — RAIL MOTOR CARS — CARRIAGES AND WAGONS — HEATING RAILWAY CARRIAGES — BRAKES (pp. 515–584)

PART III

ELECTRIC RAIL TRACTION (pp. 585–593).

(By W. A. J. Sykes, A.M.I.Mech.E.)

PART IV

RAILWAY SIGNALLING (pp. 595–611)

(By O. S. Nock, B.Sc., A.M.I.C.E., M.I.Mech.E., M.I.R.S.E.)

SECTION XXXII

RAILWAYS

PART I

PERMANENT WAY.

RAILS — FISHPLATES AND BOLTS — SLEEPERS — BALLAST — MAINTENANCE — RAIL CHAIRS — ARRANGEMENT OF PERMANENT WAY — GAUGES — LIGHT RAILWAYS — SUPERELEVATION — CROSSINGS — CURVES.

(By C. D. Morgan, Permanent Way Assistant, Western Region, British Railways.)

RAILS.

With the principal exception of Great Britain, where bull-head rails in cast-iron chairs are still the standard for railway track, the flat-bottom rail—with or without baseplates—is in universal use. The merits of each type of rail have been debated from time to time and it is of interest to record that British Railways are laying in a considerable mileage of flat-bottom rail for purposes of comparison under similar conditions, particularly from the maintenance point of view, since the initial cost of flat-bottom track with heavier rails is undoubtedly higher than that for bull-head track under present circumstances in this country. Further details of these experimental lengths are given in the next section, 'Permanent Way on British Railways.'

The general increase in speeds and axle loads in the years prior to 1939 had emphasised the necessity for a very high standard of track maintenance and there was a tendency to increase the weight of new rails everywhere. In Great Britain some miles of 100-lb. bull-head rail were laid in but the greatest increases were to be seen in the United States of America where rails weighing 131 lb. per yd. soon became common with 150 lb. rails being used on certain lines. In comparing these figures it must be borne in mind that, whereas very high speeds were attained in both countries, axle loads in the U.S.A. reached 34 tons, whereas the heaviest British locomotives

had maximum axle weights of about 23 tons. Other factors also need to be taken into consideration. For instance, in Great Britain track maintenance is normally possible throughout the year whereas thousands of miles of American track receive no attention for months on end due to the extremes of temperature making it unsafe or impossible to fettle the track, and the permanent way must therefore be designed to withstand this lack of attention, particularly the stresses associated with the heaving of the road bed during the winter months.

Permanent Way on British Railways.*

The present standard for main lines is the 95 B.S.R. bull-head rail in 60-ft. lengths joined by four-bolt fishplates. The rails are supported in cast-iron chairs, weighing 46 lb. each, which are secured to creosoted softwood sleepers (8 ft. 6 in. long, 10 in. wide, 5 in. deep), by means of either three chair screws or two through bolts. In the former case the chairs usually have plain smooth bases, although various types of projections have been experimented with as anti-spread measures. Where through bolts are used the underside of the chair is cast with transverse serrations which engage corresponding recesses adzed on the top face of the sleeper. The bolts are inserted from below with the nuts on top for tightening purposes and have a large square ribbed steel washer under the head to prevent the bolt turning. The rails are held in the chairs by means of tapered hardwood or spring steel keys driven as far as practicable in the direction of traffic.

The usual number of sleepers is 2,112 per mile of road but an additional one or two sleepers per rail length are generally used where the formation is soft, in tunnels and on curves sharper than ½-mile radius. As mentioned in a later section a considerable mileage of track has been laid with welded rails up to 240 ft. in length. It should also be recorded that various designs of semi-supported rail joints are being tried with the object of obtaining smoother running and, in some instances, the special joints provide for additional expansion of long welded rails.

In 1936 the London, Midland & Scottish Railway Company laid in for trial purposes a tota of about five miles of track with B.S. 110 lb. flat-bottom rail, and in 1939 this was increased by another six miles, the A.R.E.A. 131 lb. flat-bottom rail being used in the latter case. Both tests included trials with various designs of baseplates—rolled and drop-forged steel and cast iron—with a variety of fastenings. The London & North Eastern Railway Company also laid many miles of 110 lb. flat-bottom rails and about two miles of the same section were laid on the Great Western Railway in 1945.

In 1945 a new flat-bottom rail section, weighing approximately 113 lbs. per yd., was designed by the railway companies jointly and details are shown in fig. 1. A considerable mileage of this rail has been laid on the Western, London Midland and Southern Regions of British Railways during the last few years.

Fig. 1 also shows the cast-iron base plate and elastic rail spike fastening adopted for the experimental lengths. The following comparison of the moments of inertia for various rail sections now in use is interesting.

95 R.B.S. Bull-head	35·2 ins.⁴
110 B.S. Flat-bottom	57·2 „
113 lb. „	66·0 „
131 A.R.E.A. Flat-bottom	88·5 „

In addition to their greater vertical stiffness as girders the flat-bottom rail sections are by reason of the wide flange far more rigid laterally than the bull-head rail and one of the principal objects of the tests now being made in Great Britain is to obtain comparable data in regard to the maintenance of bull-head and flat-bottom track respectively under similar service conditions.

Concurrently with the above all Regions of British Railways are laying a proportion of switches and crossings in flat-bottom rail.

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* See also p. 514.

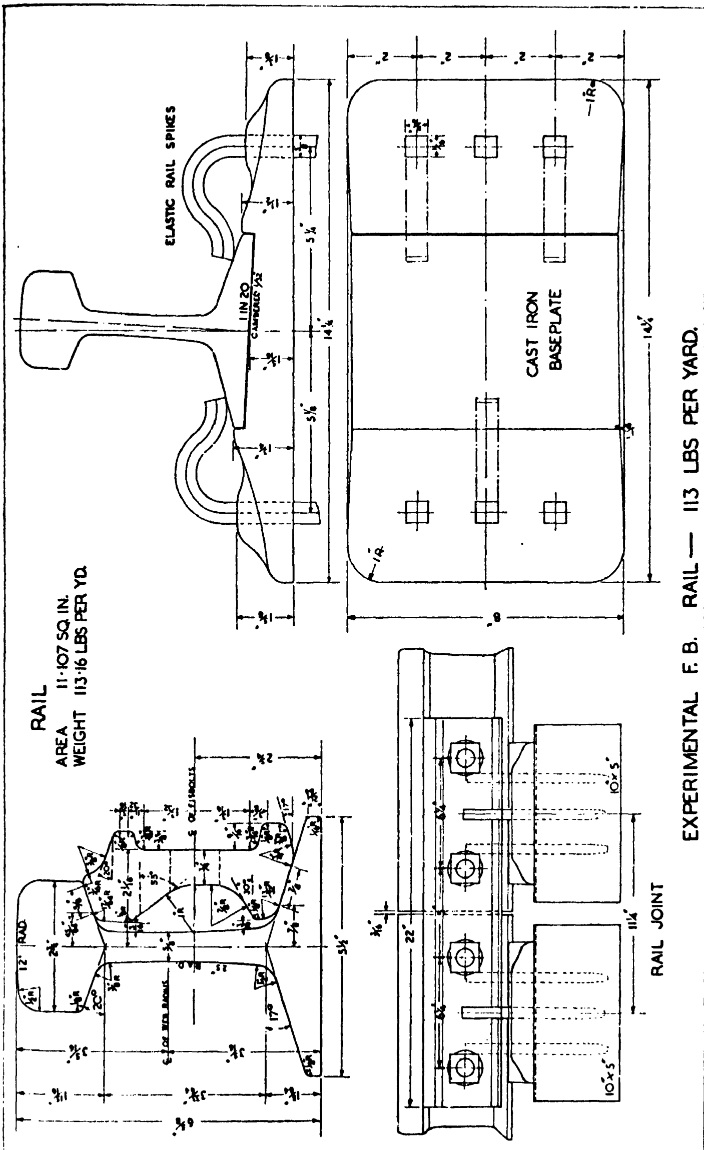

RAIL
AREA 11·107 SQ. IN.
WEIGHT 113·16 LBS PER YD.
ELASTIC RAIL SPIKES
1 IN 20
CAST IRON BASEPLATE
RAIL JOINT
EXPERIMENTAL F. B. RAIL — 113 LBS PER YARD.

Fig. 1.

BRITISH STANDARD SPECIFICATION AND SECTIONS OF BULL-HEAD RAILWAY RAILS.*

(No. 9—1935.) (*Abstract.*)

2. The steel for the rails shall be of the best quality made by the Open Hearth or Acid Bessemer process, as may be specified or required by the engineer (or the purchaser), and shall show on analysis that in chemical composition it conforms to the limits given for such processes respectively in Tables 1 and 2.

The steel shall conform to the limits of chemical composition appearing in Tables 1 and 2 under the headings of 'Higher Carbon' and 'Medium Manganese' respectively, as may be specified or required by the engineer (or the purchaser).

TABLE 1.—STEEL MADE BY THE OPEN HEARTH PROCESS.

Element.	Acid.		Basic.	
	Higher. Carbon.	Medium Manganese.	Higher Carbon.	Medium Manganese.
	Per cent.	Per cent.	Per cent.	Per cent.
Carbon . .	0·50 to 0·60	0·45 to 0·55	0·55 to 0·65	0·50 to 0·60
Manganese . .	0·90 (max.)	0·90 to 1·20	0·90 (max.)	0·90 to 1·20
Silicon . .	0·10 to 0·30	0·10 to 0·30	0·10 to 0·30	0·10 to 0·30
Phosphorus .	0·06 (max.)	0·06 (max.)	0·05 (max.)	0·05 (max.)
Sulphur . .	0·06 (max.)	0·06 (max.)	0·05 (max.)	0·05 (max.)

TABLE 2.—STEEL MADE BY THE ACID BESSEMER PROCESS.

Element.	Higher. Carbon.	Medium Manganese.
	Per cent.	Per cent.
Carbon . .	0·45 to 0·55	0·40 to 0·50
Manganese . .	0·90 (max.)	0·90 to 1·20
Silicon . .	0·10 to 0·30	0·10 to 0·30
Phosphorus .	0·06 (max.)	0·66 (max.)
Sulphur . .	0·06 (max.)	0·06 (max.)

6. A falling weight test shall be made in the following manner :—A piece of rail 5 feet (1·52 m.) long, taken from one of the rails selected by the engineer or the inspector (or the purchaser), shall be placed in a horizontal position with the head uppermost upon the two steel bearers of the British Standard Falling Weight Testing Machine, the bearers being 3 feet 6 inches (1·07 m.) apart at their centres, and having their upper surfaces curved to a radius of 3 inches (76·20 mm.).

The test shall comprise two blows delivered midway between the bearers from a falling iron weight. The height of the drop for the various sections of rails shall be as tabulated below. The blows shall be sustained without fracture, and the permanent set measured on the specified distance between the centres of the bearers resulting from the two blows must not exceed the limits given in the table, p. 474.

* By permission of the British Standards Institution.

BRITISH STANDARD BULL-HEAD RAILWAY RAILS.

B.S. Section No. 100.
100 lbs. per yard (49·61 kgs. per metre).

B.S. Section No. 80.
80 lbs. per yard
(39·68 kgs. per metre).

B.S. Section No. 60.
60 lbs. per yard
(29·76 kgs. per metre).

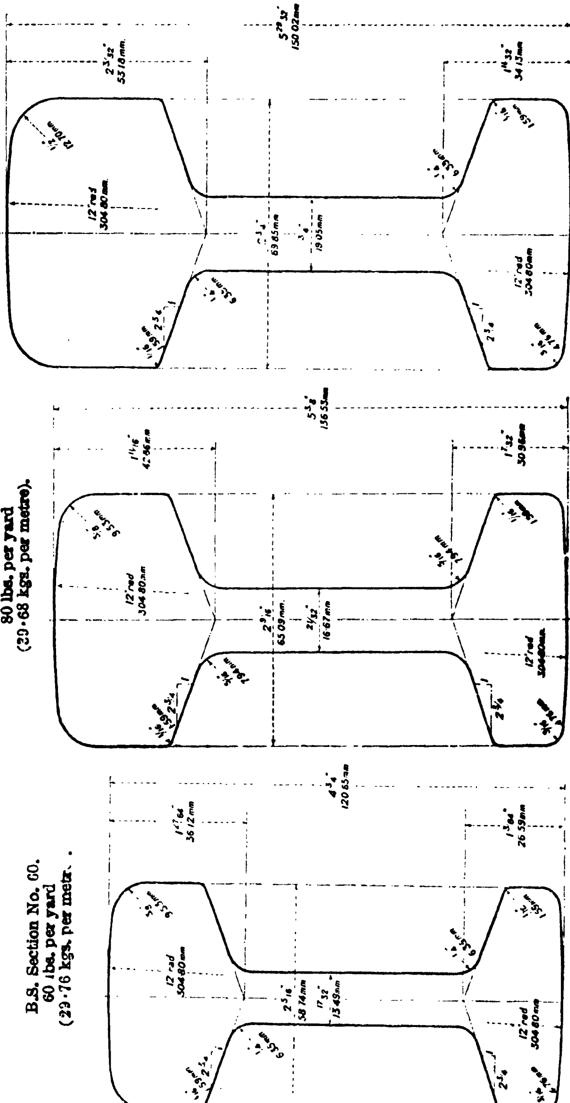


FIG. 4.

FIG. 3.

FIG. 2.

Falling Weight of 20 cwts. (2,240 lbs. = 1,016 kgs.)
Centres of bearers, 3 feet 6 inches (1·07 m.) apart.

No. of B.S. Section and Nominal Weight in lbs. per yard (kg. per metre).	Height of Drop.		Maximum Permanent Set resulting from the two blows.	No. of B.S. Section and Nominal Weight in lbs. per yard (kg. per metre).
	1st Blow.	2nd Blow.	Higher Carbon and Medium Manganese Rails.	
	ft. (m.)	ft. (m.)	in. (mm.)	
60 (29·76)	5 (1·52)	10 (3·05)	3·7 (93·98)	60 (29·76)
65 (32·24)	5 (1·52)	12 (3·66)	3·7 (93·98)	65 (32·24)
70 (34·72)	6 (1·83)	12 (3·66)	3·7 (93·98)	70 (34·72)
75 (37·20)	6 (1·83)	12 (3·66)	3·7 (93·98)	75 (37·20)
80 (39·68)	6 (1·83)	15 (4·57)	4·0 (101·60)	80 (39·68)
85R (42·16)	6 (1·83)	15 (4·57)	4·0 (101·60)	85R (42·16)
90R (44·64)	7 (2·13)	18 (5·49)	4·1 (104·14)	90R (44·64)
95R (47·13)	7 (2·13)	20 (6·10)	4·1 (104·14)	95R (47·13)
100 (49·61)	7 (2·13)	20 (6·10)	3·9 (99·06)	100 (49·61)

14. All ingots used in the manufacture of the rails shall be not less than 200 square inches (1,290·32 cm.²) in area at the larger end. The ingot shall be so fed into the cogging rolls that the bottom end arrives at the shears or hot saws first. From each end of the bloom and/or rail bar sufficient crop shall be cut to ensure that all unsound portions have been removed.

24. Dimensions and Weights of 'B.S.' Bull-Head Railway Rails.

No. of B.S. Section and Nominal Weight in lbs. per yard (kgs. per metre).	Height of Rail.	Width of Head.	Calculated Weight of Rail before Drilling. Lbs. per yard (kgs. per metre).	No. of B.S. Section and Nominal Weight in lbs. per yard (kgs. per metre).
	ins. (mm.)	ins. (mm.)		
60 (29·76)	4¾ (120·65)	2 5/16 (58·74)	59·79 (29·66)	60 (29·76)
65 (32·24)	4⅞ (123·83)	2⅜ (60·33)	64·58 (32·04)	65 (32·24)
70 (34·72)	5 (127·00)	2 7/16 (61·91)	70·13 (34·79)	70 (34·72)
75 (37·20)	5⅛ (130·18)	2½ (63·50)	74·56 (36·99)	75 (37·20)
80 (39·68)	5⅜ (136·53)	2 9/16 (65·09)	79·49 (39·43)	80 (39·68)
85R (42·16)	5 15/32 (138·91)	2 11/16 (68·26)	84·88 (42·11)	85R (42·16)
90R (44·64)	5 35/64 (140·89)	2¾ (69·85)	89·77 (44·53)	90R (44·64)
95R (47·13)	5 23/32 (145·26)	2¾ (69·85)	94·59 (46·92)	95R (47·13)
100 (49·61)	5 29/32 (150·02)	2¾ (69·85)	99·84 (49·53)	100 (49·61)

27. The standard length of the rails for straight line shall be 30 feet (9·14 m.), 36 feet (10·97 m.), 40 feet (12·19 m.), 45 feet (13·72 m.) and 60 feet (18·29 m.).

29. The rails shall be the specified length at a temperature of 62° Fahr. (16·67° C.). Any rail may be rejected which is more than three-sixteenths (3/16) of an inch (4·76 mm.) above or below the length specified at that temperature, whether for straight or curved line.

Areas of Head, Web, and Foot of 'B.S.' Bull-head Railway Rails.

B.S. No. and Nominal Weight.		Head.			Web.			Foot.			Whole Section.			
		Area.			Area.			Area.			Area.		Calculated Weight.	
lbs. per yd.	kgs. per metre.	sq. in.	mm.^{2}	Per Cent.	sq. in.	mm.^{2}	Per Cent.	sq. in.	mm.^{2}	Per Cent.	sq. in.	mm.^{2}	lbs. per yd.	kgs. per metre
60	29·76	2·71	1749	46·4	1·29	830	21·80	1·87	1208	31·80	5·87	3786	59·79	29·66
65	32·24	2·95	1906	46·6	1·36	880	21·50	2·02	1303	31·90	6·34	4089	64·58	32·04
70	34·72	3·21	2068	46·7	1·45	938	20·70	2·22	1435	32·60	6·88	4441	70·13	34·79
75	37·20	3·43	2212	46·8	1·53	989	20·80	2·36	1521	32·40	7·32	4722	74·56	36·99
80	39·68	3·62	2334	46·4	1·73	1119	22·10	2·45	1582	31·50	7·80	5034	79·49	39·43
85R	42·16	4·07	2623	48·79	1·81	1165	21·67	2·46	1588	29·54	8·33	5375	84·88	42·11
90R	44·64	4·27	2752	48·42	1·97	1268	22·30	2·58	1665	29·28	8·81	5685	89·77	44·53
95R	47·13	4·48	2892	48·27	1·97	1268	21·17	2·84	1830	30·56	9·28	5990	94·59	46·92
100	49·61	4·91	3169	50·12	1·97	1268	20·05	2·92	1886	29·83	9·80	6322	99·84	49·53

31. The holes for fishbolts shall be drilled through the web from the solid at each end of the rail of the sizes and in the position shown in the British Standard Specification for Steel Fishplates for Bull-head Railway Rails (Report No. 47—1928) or on a drawing to be supplied by the engineer (or the purchaser).

These holes shall be at right angles to the web of the rail, and clean cut, all burrs being carefully removed, and shall be checked by the manufacturer with suitable templates and gauges to be furnished by him and at his expense, and approved by the engineer (or the purchaser).

32. Should any of the holes vary from the correct size or position more than one-thirty-second ($\frac{1}{32}$) of an inch (0·79 mm.) the rail or rails in which such deviation occurs may be rejected.

British Standard Specification and Sections of Flat-Bottom Railway Rails.*

(No. 11—1936.) (*Abstract.*)

Table I (*Abbreviated*).—Dimensions.

B.S. No. and Nominal Weight.	A	B	C	D	E	F	G	J	K
lb. per yd.	in.	in.	in.	in.	in.	in.	in.	in.	in.
25R	$2\frac{7}{8}$	$2\frac{3}{4}$	$1\frac{1}{2}$	$\frac{1}{4}$	$\frac{29}{32}$	$\frac{7}{16}$	$1\frac{13}{64}$	$\frac{5}{16}$	$\frac{3}{16}$
30R	$3\frac{1}{8}$	3	$1\frac{5}{8}$	$\frac{19}{64}$	1	$\frac{15}{32}$	$1\frac{19}{64}$	$\frac{5}{16}$	$\frac{3}{16}$
35R	$3\frac{3}{8}$	$3\frac{1}{4}$	$1\frac{3}{4}$	$\frac{21}{64}$	$1\frac{3}{32}$	$\frac{1}{2}$	$1\frac{25}{64}$	$\frac{5}{16}$	$\frac{3}{16}$
40R	$3\frac{5}{8}$	$3\frac{1}{2}$	$1\frac{7}{8}$	$\frac{23}{64}$	$1\frac{9}{64}$	$\frac{17}{32}$	$1\frac{33}{64}$	$\frac{3}{8}$	$\frac{7}{32}$
45R	$3\frac{7}{8}$	$3\frac{3}{4}$	$1\frac{31}{32}$	$\frac{3}{8}$	$1\frac{7}{32}$	$\frac{9}{16}$	$1\frac{39}{64}$	$\frac{3}{8}$	$\frac{7}{32}$
50R	$4\frac{1}{8}$	$3\frac{15}{16}$	$2\frac{1}{16}$	$\frac{25}{64}$	$1\frac{19}{64}$	$\frac{19}{32}$	$1\frac{23}{32}$	$\frac{3}{8}$	$\frac{7}{32}$
55R	$4\frac{5}{16}$	$4\frac{1}{8}$	$2\frac{5}{32}$	$\frac{27}{64}$	$1\frac{11}{32}$	$\frac{5}{8}$	$1\frac{51}{64}$	$\frac{3}{8}$	$\frac{7}{32}$
60R	$4\frac{1}{2}$	$4\frac{5}{16}$	$2\frac{1}{4}$	$\frac{7}{16}$	$1\frac{13}{32}$	$\frac{21}{32}$	$1\frac{7}{8}$	$\frac{3}{8}$	$\frac{7}{32}$
65R	$4\frac{11}{16}$	$4\frac{7}{16}$	$2\frac{5}{16}$	$\frac{15}{32}$	$1\frac{29}{64}$	$\frac{11}{16}$	$1\frac{31}{32}$	$\frac{7}{16}$	$\frac{1}{4}$
70R	$4\frac{7}{8}$	$4\frac{5}{8}$	$2\frac{3}{8}$	$\frac{1}{2}$	$1\frac{1}{2}$	$\frac{23}{32}$	$2\frac{3}{64}$	$\frac{7}{16}$	$\frac{1}{4}$
75R	$5\frac{1}{16}$	$4\frac{13}{16}$	$2\frac{7}{16}$	$\frac{33}{64}$	$1\frac{9}{16}$	$\frac{47}{64}$	$2\frac{1}{8}$	$\frac{7}{16}$	$\frac{1}{4}$
80R	$5\frac{1}{4}$	5	$2\frac{1}{2}$	$\frac{17}{32}$	$1\frac{39}{64}$	$\frac{49}{64}$	$2\frac{13}{64}$	$\frac{7}{16}$	$\frac{5}{16}$
85R	$5\frac{7}{16}$	$5\frac{3}{16}$	$2\frac{9}{16}$	$\frac{35}{64}$	$1\frac{43}{64}$	$\frac{25}{32}$	$2\frac{9}{32}$	$\frac{7}{16}$	$\frac{5}{16}$
90R	$5\frac{5}{8}$	$5\frac{3}{8}$	$2\frac{5}{8}$	$\frac{35}{64}$	$1\frac{23}{32}$	$\frac{13}{16}$	$2\frac{23}{64}$	$\frac{1}{2}$	$\frac{3}{8}$
95R	$5\frac{13}{16}$	$5\frac{9}{16}$	$2\frac{11}{16}$	$\frac{9}{16}$	$1\frac{25}{32}$	$\frac{53}{64}$	$2\frac{7}{16}$	$\frac{1}{2}$	$\frac{3}{8}$
100R	6	$5\frac{3}{4}$	$2\frac{3}{4}$	$\frac{9}{16}$	$1\frac{27}{32}$	$\frac{27}{32}$	$2\frac{1}{2}$	$\frac{1}{2}$	$\frac{3}{8}$
110	$6\frac{1}{4}$	6	$2\frac{7}{8}$	$\frac{19}{32}$	$1\frac{29}{32}$	$\frac{29}{32}$	$2\frac{5}{8}$	$\frac{9}{16}$	$\frac{3}{8}$

* By permission of the British Standards Institution.

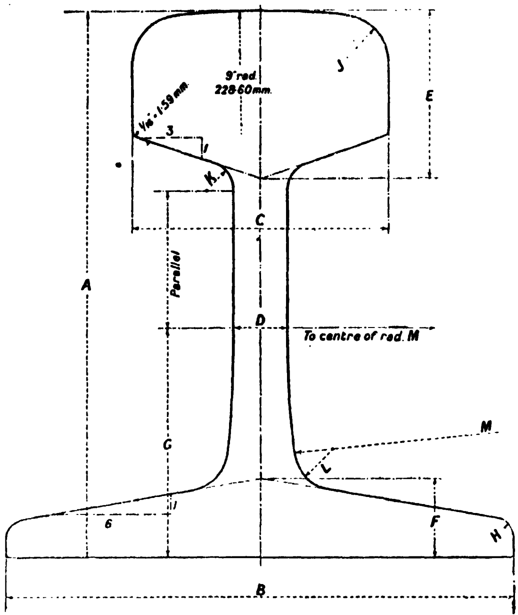


FIG. 5.—British Standard Flat-Bottom Railway Rails. Key to Table of Dimensions.

BRITISH STANDARD FLAT BOTTOM RAILWAY RAILS.

TABLE II (*Abbreviated*).—AREAS OF HEAD, WEB, AND FLANGE.

B.S. No. and Nominal Weight. lb. per yd.	Head.		Web.		Flange.		Whole Section.	B.S. No. and Nominal Weight. lb. per yd.
	Area. sq. in.	Per cent.	Area. sq. in.	Per cent.	Area. sq. in.	Per cent.	Area. sq. in.	
25R	1·129	46·04	0·438	17·82	0·886	36·14	2·453	25R
30R	1·355	45·96	0·563	19·10	1·030	34·94	2·948	30R
35R	1·598	46·49	0·656	19·09	1·183	34·42	3·437	35R
40R	1·771	45·14	0·805	20·52	1·347	34·34	3·923	40R
45R	1·996	45·21	0·898	20·33	1·522	34·46	4·416	45R
50R	2·242	45·66	0·978	19·92	1·690	34·42	4·910	50R
55R	2·420	44·76	1·120	20·71	1·867	34·53	5·407	55R
60R	2·644	44·90	1·193	20·25	2·053	34·85	5·890	60R
65R	2·804	44·00	1·342	21·05	2·227	34·95	6·373	65R
70R	2·978	43·30	1·470	21·37	2·430	35·33	6·878	70R
75R	3·193	43·45	1·588	21·61	2·568	34·94	7·349	75R
80R	3·375	42·95	1·698	21·62	2·783	35·43	7·856	80R
85R	3·598	43·09	1·825	21·85	2·928	35·06	8·351	85R
90R	3·781	42·83	1·886	21·37	3·161	35·80	8·828	90R
95R	4·026	43·15	1·992	21·35	3·313	35·50	9·331	95R
100R	4·274	43·56	2·063	21·07	3·471	35·37	9·811	100R
110	4·585	42·47	2·279	21·10	3·934	36·43	10·798	110

WEIGHT OF RAILS PER MILE OF SINGLE TRACK.

Area.	Weight per Yard.	Weight per Mile.	Area.	Weight per Yard.	Weight per Mile.
Sq. ins.	Lb.	Tons.	Sq. ins.	Lb.	Tons.
5	51·0	80	7½	76·5	120
5¼	53·5	84	8	81·5	128
5½	56·0	88	8½	86·5	136
5¾	58·5	92	9	91·5	144
6	61·1	96	10	102·0	160
6½	66·0	104	11	112·0	176
7	71·0	112			
No. of 30 ft. rails per mile = 352			No. of 45 ft. rails per mile = 234		
No. of 40 ft. rails per mile = 264			No. of 60 ft. rails per mile = 176		

Area of rail in square inches multiplied by 10·188 equals weight in lb. per yard.

Weight of rail per yard in lb. multiplied by $\frac{11}{7}$ equals tons per mile of single track.

Weight of rail in kgms. per metre multiplied by 2 equals weight in lb. per yard (approximately).

STRENGTH OF RAILS.

The following formula provides for the dynamic impact effects of engine hammer blow and lurching and affords a means of assessing the relative strength of rails :—

$$f = \frac{Wl}{5 \cdot 3\ Z}\left(\frac{V^2}{c} + \frac{V}{k} + 0 \cdot 5\right)$$

where f = max. stress in rail foot (tons per sq. in.).
W = ,, static axle weight (tons).
l = sleeper spacing (ins.).
Z = section modulus—tension (ins.^{3}).
V = max. speed (miles per hour).

The co-efficients 'c' and 'k' vary with individual classes of locomotives according to the maximum wheel hammer blow and diameter of driving wheels. Typical values for some express passenger locomotives of British Railways are given below.

Region.	*Locomotive.*	*c.*	*k.*
Eastern and N. Eastern	4-6-2 3 cyl. A-4	51,960	715
	4-6-0 2 ,, B-1	62,130	660
London Midland	4-6-2 4 ,,	89,780	725
	4-6-0 3 ,,	36,240	725
Southern	4-6-0 4 ,, 'Lord Nelson'	49,550	705
	4-6-0 2 ,, 'King Arthur'	44,300	705
	4-4-0 3 ,, 'Schools'	72,450	705
	4-6-2 3 ,, 'Merchant Navy'	—	660
Western	4-6-0 4 ,, 'Castle'	51,580	715
	4-6-0 4 ,, 'King'	60,530	700
	4-6-0 2 ,, 'County'	23,540	670

SPECIAL STEELS FOR RAILWAY RAILS.

Experiments are continually being carried out in endeavours to produce rails capable of giving a longer life in the track before their removal is indicated owing to loss of weight by abrasion, corrosion, etc.

Rails rolled from steel containing from 12 per cent. to 14 per cent. manganese appear to meet the requirement for a greater abrasive resistance but owing to their comparatively high initial cost are only economical when, in the ordinary course, the rails have a very short life and their frequent renewal entails heavy labour charges, such as in the case of complicated blocks of crossing work or in electrified lines.

For situations in which they are subjected to the action of chemical fumes, water, etc., rails containing from 0·25 per cent. to 0·30 per cent copper have been tried, and in one instance where such rails were laid in the vicinity of water troughs, the records show that after two and one-third years' life they had lost in weight about 20 per cent. less than ordinary Bessemer Acid steel rails laid at the same time for comparison.

Experiments are still being conducted with rails containing various percentages of chromium, but no definite conclusions have yet been arrived at as to their comparative superiority or otherwise over ordinary rails.

In the U.S.A. crossings (frogs) made from ordinary rails with manganese (cast) steel inserts are used extensively and have been used to some extent by British Railways.

Rails—Heat Treatment.

Many railways specify the Sandberg Regulated Sorbitizing process for rails to be used in situations where rapid wear is experienced. Briefly this treatment comprises a partial quenching of the treads of the rails, after rolling, by means of an atomized water spray followed by retarded cooling from about 550° C. to 300° C., after which the rails are allowed to cool out in the normal manner. The Sandberg treatment has the effect of considerably hardening the treads of the rails whilst the webs and feet retain their normal structure, but experience has shown that such rails are prone to develop a corrugated surface unless laid in lines carrying intensive traffic.

It is contended that rails are particularly prone to develop incipient internal fissures whilst cooling between the above-mentioned range since the tensile strength of the metal has been shown to be at its lowest at a temperature of about 400° C. The effect of slow cooling is to even out the temperature as between the external and internal metal of the rails until they are sufficiently cool to resist this tendency. Retarded cooling is now specified by British Railways for ordinary rails and undoubtedly reduces internal residual stresses in the steel.

Various other methods of heat treatment are applied to rails on the Continent, whilst in America rail ends are also hardened in the track by means of the oxy-acetylene flame followed by suitable quenching.

Maximum Allowable Axle Load.

For Light Railways.

Weight of rail in lb. per yard	30	35	40	45	50	55	60	65	70
Max. axle loading in tons	4 to 6	5 to 7	6 to 8	7 to 9	8 to 10	9 to 11	11 to 13	13 to 15	15 to 17

(*Ministry of Transport.*)

Steel Conductor Rails for Electrified Track.

Typical modern specification:

	Per cent.
Carbon	0·06 max.
Silicon	0·05 „
Manganese	0·10 to 0·20
Sulphur	0·03 max.
Phosphorus	0·04 „

Resistance not exceeding 16 microhms per 100 lb./yd. (6·5 times that of pure copper of equal ength and section area) at 60 ° Fahr.

The tensile breaking strength of such rails is approximately 20 tons per sq. in.

Short Rails in Curves, etc.

Owing to the length of the outer rail of a curve exceeding that of the inner, it is necessary in order to keep the rail joints square to insert at intervals on the inside of the curve special rails, usually from 2 inches to 5 inches shorter than the standard length.

The necessary decrease in rail length is given by:

$$d = \frac{GL}{R}$$

Where G = gauge of track,
L = length of outer rail,
R = radius of outer rail.
(All in feet.)

Also $R = \frac{GL}{d}$

For 60 ft. rails and 4 ft. 8½ in. gauge, d in inches $= \frac{3390}{R}$

On the former Great Western Railway such special rails are laid opposite one another in pairs to the extent of about 10 per cent. of the total length of track when long lengths of straight or very flatly curved lines are laid. The object of this is to obviate the cutting of short rails for curves when the main line material is eventually recovered and relaid in secondary lines or branches, as the latter as a rule contain a larger proportion of sharp curves than the main lines.

Fishplates.

British Standard Specification and Sections for Steel Fishplates for Railway Rails. (No. 47—1928.) (*Abstract.*)*

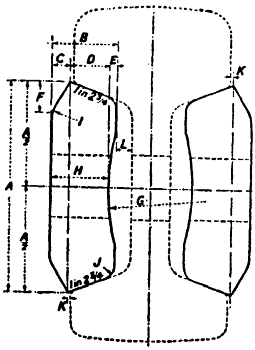


Fig. 6.

Table I.—Dimensions of Sections of Fishplates for Bull Head Rails. (Fig. 6.)

No. of B.S. Section of Rail.	70	75	80	90R & 95R	100	No. of B.S. Section of Rail.	70	75	80	90R & 95R	100
	in.	in.	in.	in.	in.		in.	in.	in.	in.	in.
A	$3\frac{7}{32}$	$3\frac{7}{32}$	$3\frac{7}{16}$	$3\frac{1}{4}$	$3\frac{1}{4}$	G	$4\frac{1}{16}$	$4\frac{1}{16}$	$4\frac{1}{16}$	$4\frac{1}{16}$	$4\frac{1}{16}$
B	1	1	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{1}{8}$	H	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{15}{16}$	1	1
C	$\frac{9}{32}$	$\frac{9}{32}$	$\frac{5}{16}$	$\frac{21}{64}$	$\frac{21}{64}$	I	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$
D	$\frac{19}{32}$	$\frac{19}{32}$	$\frac{5}{8}$	$\frac{43}{64}$	$\frac{43}{64}$	J	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$
E	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	†K	$\frac{1}{16}$	$\frac{1}{32}$	$\frac{3}{64}$	$\frac{3}{64}$	$\frac{3}{64}$
F	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{17}{32}$	$\frac{17}{32}$	†L	$\frac{17}{64}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$

The steel for fishplates to B.S.S. 47—1928 must show on analysis that it conforms to the following limits:

	Class A. Steel.	*Class B. Steel.*
Carbon	0·20 to 0·30 per cent.	0·30 to 0·42 per cent.
Manganese	Not to exceed 0·80 per cent.	
Silicon	" " " 0·15 "	
Phosphorus	" " " 0·075 "	
Sulphur	" " " 0·075 "	

Standard C or D tensile test pieces must give the following results:

	Class A. Steel.	*Class B. Steel.*
Breaking strength, tons per sq. in.	28 to 35	36 to 42
Minimum elongation, per cent. .	22	20

Worn and Reconditioned Fishplates.

Extensive use of Wonham True Temper Tapered Rail Joint Shims is now made in this country and America in connection with the maintenance of worn rail ends and fishplates.

* By permission of the British Standards Institution.
† K and L are given to the nearest $\frac{1}{64}$ inch.

To meet the various conditions of wear usually encountered these shims or packings are made in several lengths, from 9 to 16 ins., and vary in thickness at the centre from 0·03 to 0·15 ins., tapering towards the two ends. A special stepped type of shim is also available for use where unequal wear on adjoining rail ends obtains. (See fig. 7.)

These shims are proving very satisfactory in restoring dished joints and also enable fishplates to be re-used which would otherwise be scrapped.

Some railway companies recondition worn fishplates by re-forming them in suitable dies to fit the worn fishing surfaces of the rails.

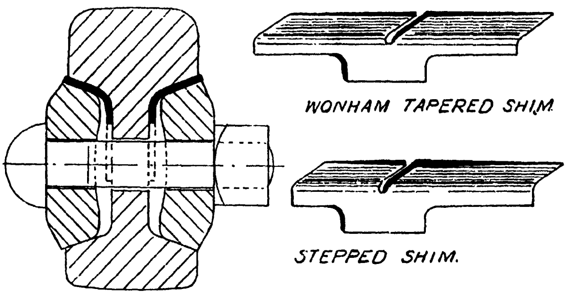


FIG. 7.

CHECK RAILS AND WIDENING OF GAUGE ON CURVES.

In accordance with the Ministry of Transport Requirements (1928), curves of ten chains radius or under in passenger lines are provided with continuous check rails. Flatter curves are sometimes similarly checked if the speeds over them are high and if they have superelevation adverse to the curvature which, although undesirable, is occasionally necessary in the case of turn-outs from the outside of a curved main line.

For curves flatter than ten chains (660 ft.) radius no widening of gauge is usual, but for sharper curves the following represents average although not universal practice.

Radius of Curve.	Gauge Widening.
From 10 chains to 8 chains	$\frac{1}{4}$ inch
Below 8 chains	$\frac{7}{16}$ inch

At level crossings, the usual practice is to provide check rails giving 2-in. flangeway clearance, although in some cases wooden guards are used in place of check rails.

The inside fishplates on continuous check rails are generally grooved to provide flange clearance when the running rails become worn.

FISHBOLTS AND NUTS.—ABSTRACT FROM B.S. 64—1946.

Tensile Test.—Shall be made either on a test piece taken from the bar or on a test piece turned from the finished bolts. The test shall be carried out in accordance with the provisions of B.S. 18 and shall comply with the requirements set out below :—

Material.	Ult. Tensile Stress. Tons per sq. in.	Min. Elongation Per Cent. on			
		Test Piece A.	Test Piece B.	Test Piece B1.	Test Piece C or Subs.
Material for bolts. Test piece cut from bar . . .	35 to 40	—	18	22	23
Test piece from finished bolt .	35 to 42	—	—	—	20
Material for nuts . .	35 to 40	20	—	—	25

Bend Test.—The bend test piece shall when cold be capable of being bent, either by pressure or by blows from a hammer, round a bar equal to its diameter or thickness until its sides are parallel without showing any signs of fracture.

Screw Threads.—Shall be either B.S.W. or B.S.F. as specified by the purchaser who shall also state to which of the following alternatives the bolts and nuts are required to conform:

(*a*) The bolts and nuts shall pass the unscrewing test. Threads cut to B.S.W. form.

(*b*) The threads shall be cut either to B.S.W. or B.S.F. form and shall confirm to the ' medium fit ' tolerances specified in B.S. 84—1940.

Unscrewing Tests.

First Test.—The nut shall be screwed on the fishbolt until it is flush with the end of the bolt and a spanner as described below shall then be affixed to the nut in a horizontal position and weighted as specified.

Second Test.—The nut shall then be screwed further on to the fishbolt until the end of the bolt projects four threads beyond the top of the nut and the spanner weighted as before shall be again similarly applied to the nut. Under each test the nut shall carry neither less nor more than the minimum and maximum weights specified in the following table.

Test.	Dia. of Fishbolt.	Test Weight in lb.	
		Min.	Max.
A	$\frac{1}{2}$ in. and $\frac{9}{16}$ in.	3	15
	$\frac{5}{8}$ in. and $\frac{11}{16}$ in.	4	20
B	$\frac{3}{4}$ in.	3	15
	$\frac{13}{16}$ in. and $\frac{7}{8}$ in.	4	20
	$\frac{15}{16}$ in. and 1 in.	5	25
	$1\frac{1}{8}$ in.	6	30

For tests A the spanner shall weigh $7\frac{1}{2}$ lb. and the test weight shall be suspended at a distance of 2 ft. 6 in. from the centre of the nut.

For tests B the spanner shall weigh 10 lb. and the distance between the centre of the nut and the point of suspension of the test weight shall be 3 ft.

Wooden Sleepers.

In normal times British Railways make large use of creosoted softwood sleepers, usually Baltic Redwood (*Pinus Sylvestris*) or Douglas Fir (*Pseudotsuga Douglasii*), although hardwoods such as Jarrah are preferred by certain Lines for electrified track. The standard sizes are 8 ft. 6 ins. long, 10 ins. wide, 5 ins. thick, and where special joint sleepers are used they are 12 ins. wide.

In 1938-9 a considerable number of French Maritime Pine (Landes Fir) sleepers were imported and these, if well selected, are considered to be very little if any inferior to Baltic Redwood.

Restrictions on imports from the Baltic and Mediterranean ports in war time have necessitated the use of larger numbers of Canadian Douglas Fir sleepers. Owing to the difficulty of effectively creosoting this species of timber by the ordinary pressure process, disappointing results had frequently arisen from the use of Douglas Fir sleepers in the past, but a longer life is anticipated from later supplies which have been made to a more rigid specification and subjected to incising before creosoting.

The incising operation consists in passing the sleepers through a machine provided with four rollers each equipped with removable cutters, designed to part the fibres of the wood without crushing or breaking, and if creosoting is carried out immediately, it is found that uniform penetration results with this otherwise refractory timber.

Incising and creosoting are carried out in accordance with B.S.S. 913—1940.

Whilst pressure creosoting is still the most used preservative treatment for railway sleepers, extensive trials are being undertaken with various water soluble salts. One of these treatments known as Tanalizing (originally Wolmanising) is being very largely used for crossing timbers on one Region of British Railways. Sleepers have also been treated experimentally with belcure preservative.

Another result of the restricted imports of sleepers is the use of home grown softwoods and hardwoods, principally Scots fir, larch, oak, beech and elm, for less important branches and siding work. These can be utilised uncreosoted when required for more or less temporary sidings, but under such conditions their life is short compared with properly treated sleepers.

Crossing timber is usually 12 ins. × 6 ins. and 14 ins. × 6 ins., and varies in length from 8 ft. 6 ins. to 40 ft., whilst longitudinal bridge timbers may be 16 ins. × 7 ins. upwards where they are required to take standard chaired track.

LENGTH OF WOODEN SLEEPERS.

	Ft. Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.
Gauge . . .	5 6	4 8½	3 6	metre	2 6	2 0
Sleeper length .	9 0 to 10 0	8 0 to 9 0	6 0 to 6 6	6 0 to 6 6	5 0	4 0

Metal Sleepers.

Whilst wooden sleepers predominate throughout the world there are considerable mileages of track laid on steel or cast-iron sleepers.

Cast-iron sleepers of the circular pot type have been used in India for very many years and, during the last two decades, other types of cast-iron sleepers have been designed and laid with both flat-bottom and bull-head rail. The basic design of steel sleepers for flat-bottom track has remained practically unaltered for over forty years but improved types of fastenings have been developed, and fig. 8 shows the 'loose jaw' pattern which is commonly used in India. Gauge is maintained

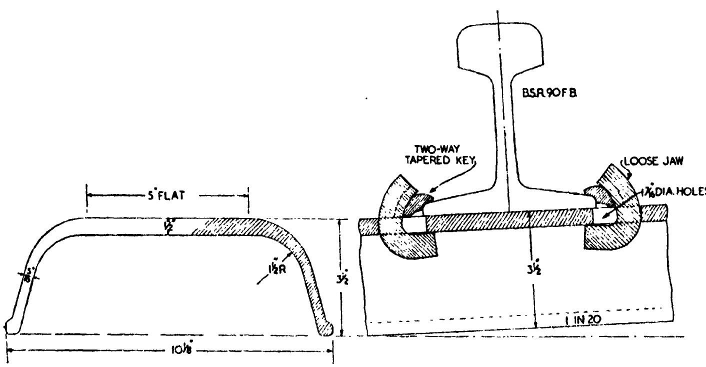


FIG. 8.

or, where necessary, widened to a limited extent on curves, by adjustment of the tapered steel keys which are driven in opposite directions.

Between 1929 and 1936 considerable numbers of steel sleepers were laid in Great Britain. Generally speaking, these have given fairly satisfactory service but many lengths have suffered from extensive corrosion and in others some distortion of the trough section has occurred with the result that premature renewal has been necessary.

The following is a brief description of some of the designs of steel sleepers laid in by the former Great Western Railway Company, which still has a considerable mileage of steel-sleepered track on its system.

(1) G.K.N. Composite Type (fig. 9).—This consists of the usual rolled and pressed mild steel trough section, the chairs, which are of cast iron, being cast in position on the plate by means of

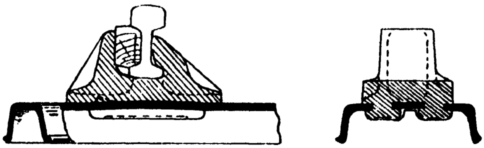


FIG. 9.—G.K.N. Composite Steel Sleeper.

two elongated snugs projecting on the underside. It is 8 ft. 6 ins. long by 3 ins. deep and was manufactured in two widths, 11 ins. and 10¼ ins., the weights of the complete sleepers being 245 lb. and 228 lb. respectively.

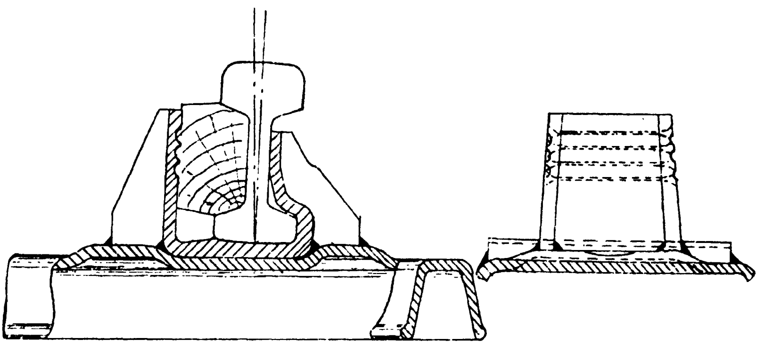


FIG. 10.—Workington Steel Sleeper.

(2) Workington Type (fig. 10).—This sleeper also is 8 ft. 6 ins. long but is 11¼ ins. wide overall and 2¾ ins. deep, weighing 196 lb. The chairs are pressed from rolled steel plate and are welded to the sleeper with a continuous fillet round the base.

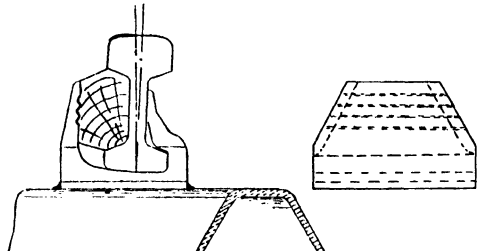


FIG. 11.—Dorman Long Steel Sleeper.

(3) Dorman Long Type (fig. 11).—This is somewhat similar to the previous design but weighs only 180 lb.

(4) Sandberg Type (fig. 12).—The Sandberg sleeper is pressed from a special rolled section having a deep central rib, the jaws being punched and pressed from the metal of the sleeper plate.

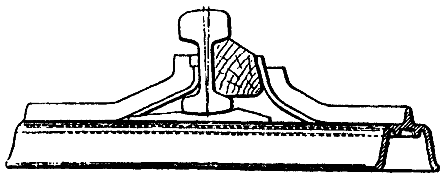


FIG. 12.—Sandberg Steel Sleeper.

A loose-bearing plate forms the rail seating and is prevented from moving laterally by a slot which engages the rail jaw. Total weight 190 lb.

PREVENTION OF RAIL CREEP.

Many 'anti-creep' devices have been submitted to tests during the past few years in an endeavour to reduce the large annual expenditure necessitated on certain lines in pulling back rails subject to creep. These comprise various forms of spring steel keys, e.g. the 'Mills,' 'Stuart,' 'Tees Side,' and 'Turplat'; also rail anchors, of which the 'Phillips,' 'Yates,' and 'Winby' are typical examples.

Steel keys or rail anchors for the prevention of creep are usually fixed at the rate of about six per 45 ft. rail and the former are of various types, some having separate tapered wedges for tightening, whilst others are made in one piece. Fig. 13 illustrates a steel key of the latter type known as the 'Turplat,' manufactured by George Turton Platts & Co., Ltd.

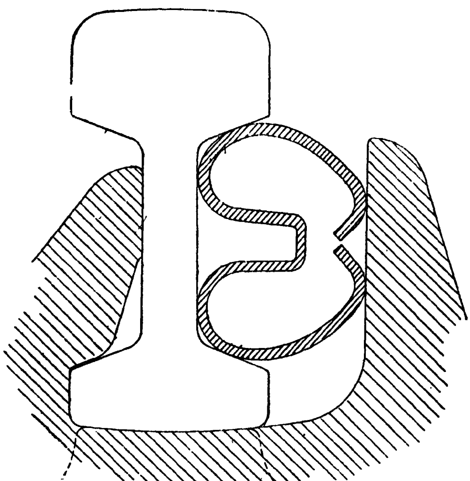


FIG. 13.—Turplat Steel Key.

In recent years steel keys have been used exclusively in some lengths of bull-head track, but experience on one railway at least indicates that it is desirable after relaying to re-use the wooden keys until the new track has settled down, after which the steel keys can be driven. Failure to adopt this practice has resulted in difficulties being experienced in connection with alignment.

Rail anchors may also be classified as one-piece or otherwise in design, and fig. 14 illustrates the 'Phillips' anchor, which belongs to the former category. This device is forced over the foot of the rail (close to a chair), which causes it to exert a vicelike grip, and by its bearing against

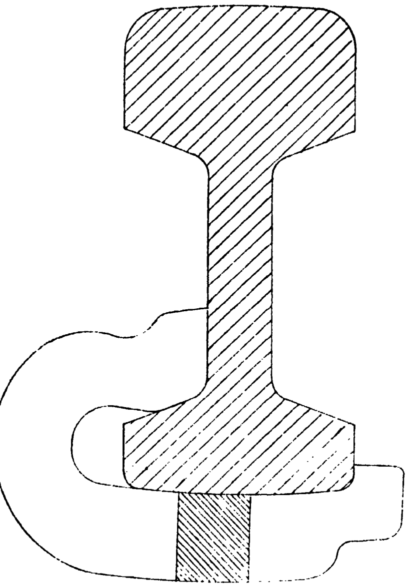


FIG. 14.—Phillips Rail Anchor.

the side of the chair, lessens the tendency for the rail to slide forward. The sleepers should, of course, be well boxed to prevent them 'skewing.'

Rail anchors are made for bull-head and flat bottom-rails.

The use of teak keys in place of oak is also stated to be beneficial in regard to the prevention of rail creep, as after a few years in the line oak keys tend to shrink.

Steel keys need less redriving than wooden ones.

Ballast.

The purpose of ballast is (*a*) to distribute the load over a larger area of the formation, (*b*) to provide a more or less uniform elastic medium between the track and the bed, (*c*) to provide rapid and efficient drainage of surface water, and (*d*) to hold the sleepers in position.

There are usually two distinct layers: the 'bottom,' consisting of stone or slag varying in size from 4 ins. to 8 ins., is laid directly on the formation to a thickness of about 9 ins.; the upper or top ballast extending from about 3 ins. below the underside of the sleepers to the level of their tops. The average size of top ballast, consisting of crushed stone, slag or gravel, varies from ½ in. to 2½ ins., but for steel sleepers the maximum is about 1½ in.

For sidings, unimportant branch lines, and for clay cuttings and banks, ashes are largely used for both bottom and top ballast.

The top ballast should extend for about 12 ins. or 18 ins. beyond the ends of the sleepers, with a 1 to 1 slope at the sides; the bottom ballast being carried about 2 ft. or 3 ft. beyond the sleeper ends.

In modern railway practice, the formation is 2 ft. below rail level, and is cambered 6 ins., dry stone cross-drains being formed as required into open jointed pipes or 'U' shaped drains (in the case of cuttings) laid in the cesses.

For single lines the width of top ballast required is about 12 ft. with 18 ft. of bottom ballast, and for double lines, top ballast 23 ft., bottom ballast 28 ft. to 30 ft.

CUBIC YARDS OF BALLAST PER MILE OF TRACK.

Side Slope of Ballast, 1 to 1. 6-in. Formation Camber.						
Depth of Ballast in Inches.	Width in Feet.					
	Single Track.			Double Track.		
	10	11	12	21	22	23
12	1,930	2,120	2,320	3,860	4,060	4,260
18	3,160	3,440	3,740	6,160	6,460	6,760
24	4,480	4,880	5,260	8,560	8,960	9,340
30	5,880	6,380	6,880	11,050	11,560	12,030

NOTE.—Allowance has been made for wooden sleepers in this table.

Blast-furnace and steel slags are much used as ballast where supplies are readily available. Cinder ballast is now being superseded by stone or slag where a sufficient supply can be obtained at an economical cost; but where the railway is subjected to subsidences cinder ballast is used, otherwise successive liftings of the tracks would cause large quantities of the ballast gradually to disappear below the ordinary level of the ballasting. The use of slag ballast is generally avoided in tunnels, because in damp and confined situations it is considered to increase corrosion of the track; it is not used on electrified lines of at least one administration, on account of certain chemical properties, and also because it is considered too retentive of moisture; this opinion is, however, not general, as it is ordinarily laid on other important electrified railways.

In course of time ballast becomes dirty and consolidated so that it interferes with the drainage and on this account should be periodically cleaned by screening. This is usually done by the use of ballast forks or screens. In some countries mechanical screening machines are employed.

PACKING OF SLEEPERS.

Hand packing of sleepers as opposed to mechanical tamping is still the general practice in Great Britain, and recent years have seen developments in this respect which make the operation more effective.

There are four principal methods of hand packing:

(i) By beater picks.

(ii) By ordinary shovel.

(iii) By small shovel and fine chippings.

(iv) Similar to (iii) but embodying precise measurement of the lift required.

The last process is known as measured packing, and the necessary equipment is supplied by Abtus, Limited. (i) and (ii) are still used to a large extent for general lifting of the track with ordinary ballast, whilst (iii) and (iv) are confined to taking out slacks, etc., in the ordinary course of maintenance.

By method (iii) the quantity of chippings to be distributed under the previously jacked up sleepers is determined by the ganger from experience and with practice gives good results, but in common with measured packing is not suitable on soft formations.

Measured packing, as now practised on a fairly extensive scale, comprises the following operations:

(i) Determining by means of adjustable graduated sighting boards the quantity of chippings required under each sleeper to bring the rail top level.

(ii) Ascertaining by the readings of 'Voidmeters' the amount of chippings required to fill up voids which allow sleepers to be depressed under traffic.

(iii) Lifting the track by jacks and spreading the required quantity of chippings evenly over the sleeper beds (under the rails) by means of specially designed small flat shovels.

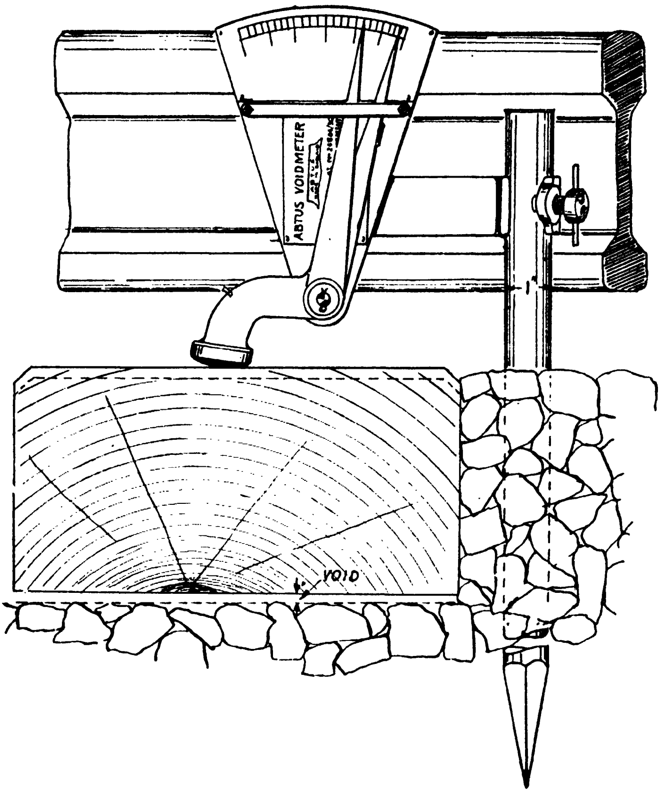


FIG. 15.

Fig. 15 shows the Voidmeter in position. If the sleeper is depressed during the passage of a train owing to a void immediately under it, the amount of such movement is indicated by the distance between the fixed and moving pointers and is read from the scale which is graduated in terms of the special canisters used for the stone chippings.

Rail and Flange Lubricators.

The use of automatic lubricators on the outer or high rail of curves to reduce side wear of rails due to flange pressure is being extended rapidly in view of the benefits derived and improved designs of lubricators now available. An efficient machine will at least double the life of the rail. Fig. 16 illustrates a design of grease lubricator now favoured in Great Britain and also available for flange rail track. Grease (graphite) is forced on the side of the rail head through a series of orifices in a plate situated close to the running face of the rail by means of two small plunger pumps which are operated by the treads of wheels.

Another type of lubricator employs oil which is fed to the rail and wheel flange by a soft felt pad. The efficiency of oilers of this design depends to a large extent on the maintenance of the felt pad which must be kept clean and soft as well as being correctly adjusted as regards position.

Special lubricators have been designed by the makers of both types referred to for greasing the rubbing faces of check or guard rails.

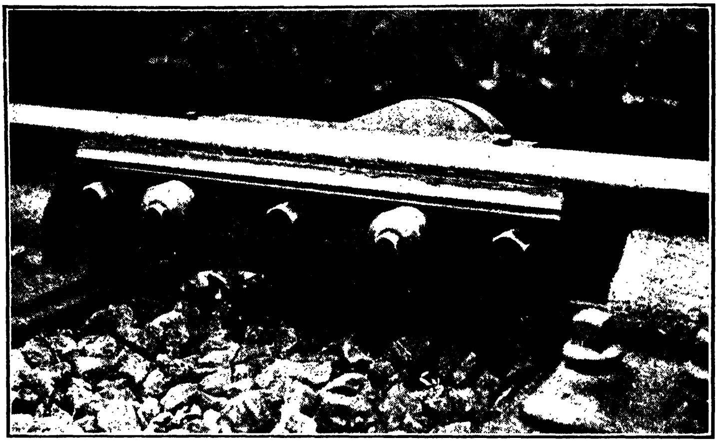


FIG. 16.—Rail and Flange Lubricator.

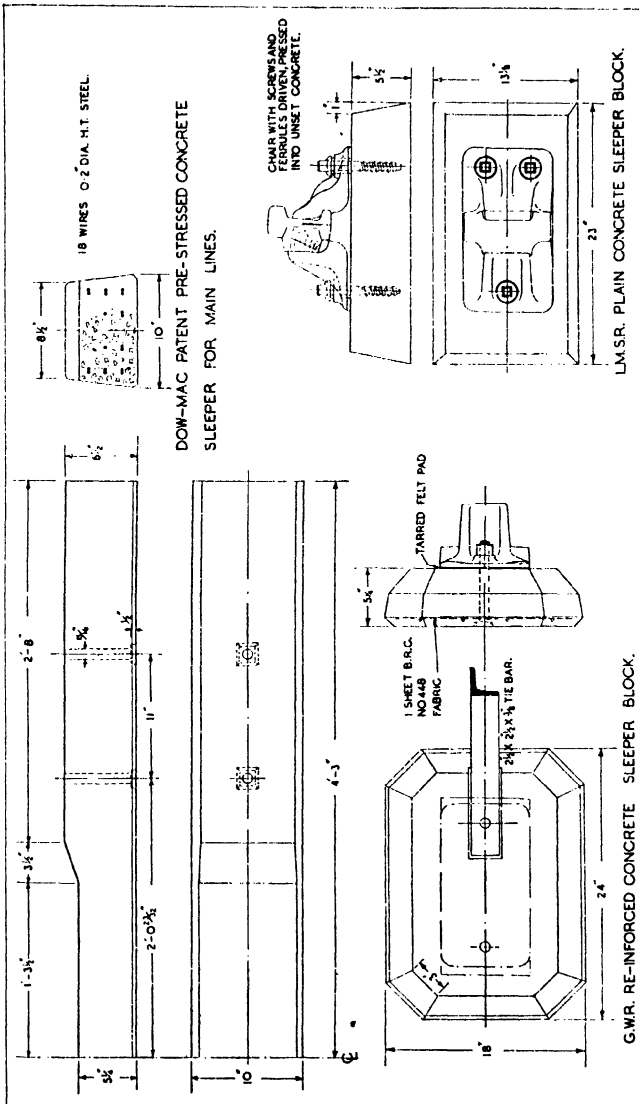
Concrete Sleepers.

The shortage of timber since 1939 owing to restrictions on imports into Great Britain has resulted in extensive trials of sleepers made in re-inforced concrete. In sidings, goods running loops and lightly operated mineral and branch lines independent concrete block sleepers have been largely used, every third or fourth pair of blocks being connected together by steel gauge ties or replaced by transverse concrete sleepers. Fig. 17 illustrates a typical block sleeper of which about 875,000 have been made and used by the former Great Western Railway Company since early 1940. The drawing shows the latest design with a single sheet of steel mesh fabric reinforcement. Fig. 17 also depicts another type of concrete block sleeper made by the London, Midland & Scottish Railway Company for siding use. These blocks were not re-inforced and the method of attaching the rail chair is interesting. First the ferrules were inserted into the holes in the casting, after which the chair screws were driven tightly into the ferrules and the whole assembly was then pressed firmly into the wet concrete in the moulds and allowed to set.

Numerous types of transverse concrete sleepers have been designed and many have been tested in the track. For sidings and large depots, etc., ordinary reinforced concrete may give a reasonably long life but for running lines where speeds exceed say 30 mile per hour the most promising results so far have been obtained with concrete sleepers embodying the principle of pre-stressing. The DOW-MAC Patent Pre-stressed Concrete Sleeper shown in Fig. 18 is a typical example of sleepers designed for main line speeds and axle loads.

All types of concrete sleepers, however, possess one disadvantage compared with timber, namely, excessive weight, which may be from four to five times that of a wooden sleeper and requiring at least six men to handle them.

B.S.S. No. 986—1944 provides for block and transverse concrete sleepers for bull-head and flat-bottom rails suitable for use in sidings, goods lines, etc., over which the speed is limited to 30 miles per hour. This specification includes requirements for both ordinary re-inforced concrete and pre-stressed concrete and provides for various means of fastening the rails or chairs.



CONCRETE RAILWAY SLEEPERS.

FIG. 17.

Cast-Iron Rail Chairs.

Abstract from Great Western Railway Specification :—

Quality of Material and Castings.—**The chairs shall be cast from a mixture of tough grey iron run from the cupola and shall be good smooth clean castings properly dressed, free from honeycomb, air bubbles, cold shuts, sponginess, and other defects.**

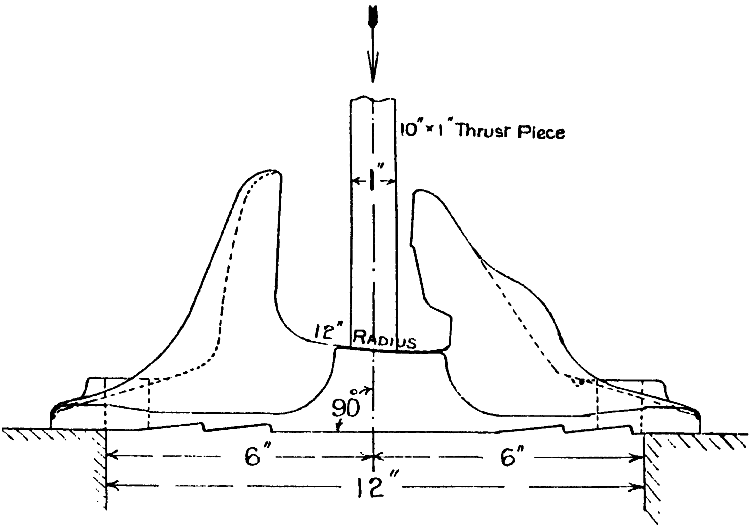


Fig. 18.—G.W.R. Rail Chair Test.

Cooling of Chairs.—**Immediately after the removal of each chair from the mould it shall be protected in such manner as shall satisfy the engineer or his deputy that its strength will not in any way be diminished by too rapid or unequal cooling.**

Variation of Weight.—**No chair will be accepted the weight of which varies more than ½ lb. above or below the computed weight.**

Tests.—**Each 25 tons of chairs cast shall be kept separate and three chairs shall be selected therefrom by the engineer or his deputy for test purposes.**

Each chair shall be supported in its normal position and for its full width on two perfectly true, level and rigid bearings, each bearing being placed six inches clear of and parallel to the centre line of the rail seat. The test load shall be fairly and uniformly applied along the full length of the centre line through a vertical thrust-piece 1 in. thick having the face in contact with the chair curved to a radius of 12 ins. in cross section (see fig. 18).

Each type of chair shall be required to sustain without fracture the following test load :—

'95' Chair	19 tons
'85' „	15 „

The test shall be continued in each case until the chair is broken.

Mechanical Platelaying Appliances.

The ever-increasing necessity for the utmost economy in maintenance and renewal expenditure has inevitably led to a more extended use of mechanical tools, etc., for various platelaying operations, and the following appliances are at present being tried on British Railways.

Mechanical Tampers.

Although extensively used in America, pneumatic, electric and self-contained petrol-engined tie or sleeper tampers have not yet been largely adopted in Great Britain. Most railways, however, are giving these tools a trial, as there are undoubtedly certain situations where their use would be beneficial. One of the chief disadvantages of the pneumatic and electric type of tamper is the necessity for some form of heavy, expensive power unit, usually a petrol engine, which has to be loaded and off-loaded at the site of the work.

Ballast Cleaning and Tamping Machines.

Self-propelled diesel-engined machines for the automatic screening of ballast are under trial in various parts of the world. The 'Matisa' machine, manufactured in Switzerland, is undergoing tests in Great Britain and North America.

A separate machine from the same factory performs automatic tamping of track, and several of these machines are now operating in this country.

Drilling, Sawing, Boring and Screwing Machines.

Portable machines for drilling and sawing rails and girders, boring sleepers and timbers, screwing home fastenings, etc., have been developed during recent years, and are useful time-saving devices for relaying and repair work.

Ratchet Jacks, Track Liners, Rail Lifters, etc.

The older types of screw jacks used for many years by platelayers are now being replaced by quick-acting ratchet jacks with trip release and the sluing of the track by small gangs has been rendered possible by the introduction of track liners, two of which can be made to do the work previously performed by six to eight men equipped with slewing bars.

Rail lifters enable 60 ft. rails to be lifted from the sleeper ends and dropped into the rail chairs by a gang of about six men where twenty-four would normally be required.

Rail creep adjusters, operated by means of right- and left-hand screw threads, enable up to 15 rails (45 ft. long) to be pulled or pushed at a time when track subjected to creep calls for adjustment.

Use of Cranes for Track Renewals.

Relaying of plain line track is now frequently performed by means of cranes. These, in conjunction with specially designed lifting beams, lift out the old track and load it on to wagons, then lay the new track previously assembled at a depot into 60 ft. lengths. This method of renewal considerably reduces the number of line 'occupations' but entails the occupation of an adjoining track in addition to the one being relaid, during the time the actual renewal is in progress. The Southern and Western Regions of British Railways have also developed special diesel-electric crane vehicles to enable relaying with pre-fabricated track to be undertaken in tunnels.

Track-Laying Machine.

One of these machines, designed and patented by Mr. A. W. Bretland, Deputy Chief Engineer, Great Southern Railways, Ireland, has been in use on that railway for several years. The special train forming the track-layer consists of engine, brake van, a number of wagons each loaded with six pairs of 45-ft. lengths of track, and empty wagons to receive the lengths of old track removed from the line, the special cantilever track-layer and wagon containing an electric winch for hauling the new track into its final position. The wagons used are ordinary rail wagons provided with side runners for the traverser.

The new track is framed up by means of crane power, complete with cross-sleepers, etc., at the depot, and the recovered track dismantled and sorted at the same place.

In Ireland the track-layer reduced labour costs in relaying by 40 per cent., and the results obtained were stated to be completely satisfactory from every point of view.

Loading and Unloading Rails Mechanically.

It is possible to effect considerable economies in the operations of loading and unloading new and old rails by a method now being used on the Great Western Railway. The rails are unloaded over the ends of the wagons by means of wire ropes and chains fastened at one end to the existing track, and at the other to the ends of rails to be unloaded. The wagon is then drawn forward by an engine and the rails slide on to the track via a 2-wheeled ramp attached to the rail wagon. Sixty-foot rails are dealt with in this manner without crippling. In the operation of loading the rails, one end of a wire rope is attached to the draw bar hook of the engine (or adjacent wagon) and the other end, to which are fastened twin chains and shackles, is carried over the truck to be loaded, and attached to the rails lying on the ground. The brake is then applied lightly to the truck, and the engine moved back, drawing the rails over a roller fixed to the end of the wagon. These two operations can be performed at considerably less cost than by the old methods of unloading by skids, etc.

Continuous Welded Rail.

During the last ten years considerable progress has been made in the direction of attaining smoother running track by the elimination of rail joints. This may be effected by the provision

of special type joints such as the Ellson (fig. 19), by welding the rails into longer lengths or by a combination of both. One of the longest lengths of continuously welded track in existence is to be found in the U.S.A. where one railroad company has over five miles broken only by a number of turnouts and insulated block joints. Many other comparatively long lengths of welded rail exist in America and other parts of the world, especially in tunnels where temperatures are generally fairly constant. In the Moffat Tunnel of the Denver & Rio Grande Western Rail Road, for instance, there is a continuous length of 18,000 ft. of rail welded by the Full-fusion Thermit process, the rails being the A.R.E.A. 112 lb. section.

The comparative lateral flexibility of the bull-head rail compared with the flat-bottom section has so far confined welding of rails in Great Britain to lengths not much exceeding 200 ft. It is now fairly well established as the result of special tests that the expansion and contraction of long lengths of welded rail due to temperature variations are limited to a length of say 40 or 50 ft. at each end of the long rail, the resistance to movement of the intermediate portion resulting in

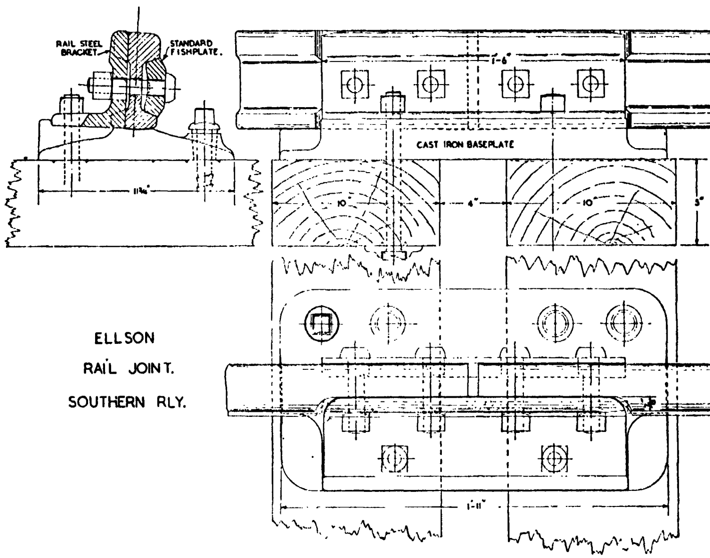


FIG. 19.

tensile or compressive stresses in the rail itself. Experiments to determine the intensity of such stresses under various conditions have been made, but further work is necessary before definite conclusions can be drawn.

The methods adopted for rail welding are either the Thermit process (with or without the insertion of a soft steel shim), the Oxy-acetylene Pressure weld or the Automatic Electric Flash-Butt Weld. The latter process has been used in England to an increasing extent for welding running and conductor rails. Generally speaking the finished lengths of the welded rail has varied from 78 ft. to 180 ft. although conductor rails have been much longer. There is every indication that the next few years will witness an extension of continuous rail welding.

The normal electric flash-butt welded rail joint will withstand a falling weight test almost if not quite as well as the rail prior to welding, whereas the Thermit process does not produce a weld capable of standing up to even a moderate drop test. In spite of this, however, Thermit welded rails are giving satisfactory service after ten or more years' life in main lines.

Resurfacing Worn Rails by Welding.

The practice of building up or resurfacing *in situ* worn switch and crossing rails is now practically universal. For a number of years prior to 1937–38 the electric arc process held the field in this respect, but modern practice favours the oxy-acetylene method, and on some railways the portable electric arc welding plant is now used almost exclusively for structural and shop welding.

Clearances on Curves.

The greatest distance by which the end of a vehicle overhangs the running face of the rail on the outside of a curve is termed the End Throw.

If

R = radius of curve;
G = gauge of track;
W = width of vehicle;
L = overall length of vehicle;
B = wheelbase or distance between bogie centres. (All in feet.)

$$\text{End throw} = \frac{W - G}{2} + \frac{L^2 - B^2}{8R}.$$

The overhang of the centre of the vehicle on the inside of a curve is known as the Centre Throw, and with the same notation

$$\text{Centre Throw} = \frac{W - G}{2} + \frac{B^2}{8R}.$$

In the case of structures adjoining the inner rail of a curve having superelevation, additional clearance should be allowed.

Permanent Way—Heights and Widths.

Ministry of Transport Requirements or Recommendations.

	Ft.	Ins.	
Outside of widest stock to face of structure	2	4	minimum
Face of signal posts, water columns, etc., between tracks spaced at 9 ft. intervals to bodywork of stock		18	minimum
Between outside heads of rails for double track	6	0	minimum
For additional single running lines or double lines alongside existing main lines	10	0	minimum
Running face of rail to structure	4	9½	
Overhead clearance between load gauge and structure		6	minimum
		12	desirable
Clearance between bodywork of stock on adjacent lines	1	6	minimum
Maximum height of existing load gauge above rail level varies from	13 ft. to 13 ft. 9 ins.		
Height of structure above rail level	15	0	desirable
Width between structures, single track	14	4	minimum
" " " double track	25	6	minimum

Note.—Additional clearances required for curved or canted track.

Total breadth on top of embankments and bottom of cuttings about 18 ft. for single, and 30 ft. for double lines.

Platforms.

The minimum clear width of any platform throughout its length to be 6 ft. At important stations the width to be not less than 12 ft., except for short distances at either end in any case of difficulty. In the case of island platforms, the minimum width for an adequate distance on each side of the centre of its length to be 12 ft. The descent at the ends of platforms to be by ramps and not by steps.

Columns for the support of roofs, and other fixed works, not to be less than 6 ft. clear from the edge of platforms. A general clear headway of not less than 8 ft. to be provided over platforms. The height of platforms above rail level may vary according to traffic and other conditions; as a rule, it should be 3 ft., but in no case less than 2 ft. 9 ins. or more than 3 ft. at permanent stations, without special approval.

The edges of the platforms to overhang not less than 12 ins., and the recess so formed to be kept clear as far as possible of permanent obstruction.

Gradients.

It is desirable to avoid constructing a station on, or providing a siding in connection with a line which is laid upon a gradient steeper than 1 in 260.

Economic System of Track Maintenance.

On branch lines where traffic is light and the sections extensive, considerable success has been attained in this country by the use of rail motor trollies and a system of 'occupation keys.' Key boxes in telephonic communication with the signal boxes are fixed at convenient intervals along the section of line and before a trolley is placed on the running line the ganger withdraws the key, which is electrically released from the signal box. The key can be removed from, and replaced in, any of the key boxes and affords complete protection to the holder, as whilst the key is withdrawn the signalman cannot allow a train to enter the section of line occupied by the trolley. On arrival at the site of work the trolley is removed clear of the line and the key replaced in the nearest convenient occupation box; the ganger then informs the signalman by telephone from

the box that the section is clear and normal working may be resumed. The key cannot be withdrawn again without the signalman's permission. The G.W.R. were the pioneers of the use of the motor trolley for track maintenance purposes in this country, and at present over 1,000 miles of track is maintained by this system.

Chemical Weed Killing.

The reduction in the strength of branch line gangs brought about by economic systems of maintenance, has made it necessary to use chemical weed killers in increasing quantities, and so far as Great Britain is concerned excellent results are being obtained by the use of 'Atlacide' (calcium chlorate), which is obtained in the form of a liquid concentrate.

The chemical is sprayed on the foliage of the weeds after mixing with water in the proportion 1 gallon of concentrate to 6 gallons of water, which represents a 7·2 per cent. solution by weight of chemicals.

On the former G.W.R. a considerable mileage of track is treated in this manner by a special weed-killing train, which, travelling at 20 m.p.h., can deal with about 25 miles of single line at one filling. In addition to the engine and brake van the train comprises three water tenders (total capacity 9,500 gallons) and a rail tank wagon containing the concentrate. The tenders are filled to six-sevenths of their capacity with water, and concentrate is pumped into them from the tank wagon until full and thoroughly mixed by steam from the engine, which also operates the spray pump on the tenders.

The action of 'Atlacide' on weed growth depends on the absorption of the chemical by the leaves and stems, hence it is necessary to delay spraying until the weeds have made sufficient growth to enable the foliage to be well covered. Fine grass is not affected to the same extent as the larger weeds.

Switches and Crossings.

***Design.*—The older types of switches, or points, in which the tongues were hinged on the fishplates at the heel necessitating the fishbolts being left slack, are now generally being superseded by 'heel-less' flexible switches, the tongues of which are made longer and held in fixed chairs for a distance of about 3 ft. 6 ins. from the heel joint, giving a minimum length of approximately 17 ft. for springing them to the required opening at the toe, which varies from 3⅜ ins. to 4½ ins. Most English railways have now adopted as standard, designs which provide for a straight tongue for the length of the side planing, the remainder of the switch being curved. The practice of the G.W.R. is to curve the tongue throughout, which gives a smaller deflection angle at the toe for the same length of switch. Switches and crossings are usually laid on creosoted cross timbers, 12 ins. or 14 ins. wide, 6 ins. deep, and varying in length from 9 to 35 ft.**

***Calculations.*—The method now generally adopted for describing the angle of a crossing (1 in N) is by the length in feet (N) along the centre line to the point where the gauge lines are 1 ft. apart (measured perpendicularly to the centre line) and this method (Centre Line Measure) is used in the formulae on pages 2039 to 2044.**

If

θ is the crossing angle measured in degrees,

$$N = \tfrac{1}{2}\cot\frac{\theta}{2}$$

and

$$\theta = 2\cot^{-1} 2N.$$

Ministry of Transport Regulations for Permanent Way.

British Standard Specifications and Sections for all details of permanent-way to be adopted, unless authority to the contrary is given. On lines on which the normal traffic is heavy and worked at high speed, the weight of new rails should not be less than 85 lbs. per yard, and, in the case of a chaired road, the common chairs should not weigh less than 45 lbs. each. On lines where traffic conditions are less severe, chairs weighing not less than 40 lbs. each may be used. The minimum length of rail, as a rule, to be 30 ft. On lines normally used for light traffic and moderate speeds, lower weights of rails and chairs may be used.

Chairs, if used, must be secured to the sleepers, at least partially, by metal bolts, screws, or spikes. With flat-bottom rails, or bridge rails, the fastenings at joints and at one or more intermediate places to consist of fang or other through bolts, and such rails, on curves with radii of 15 chains or less, to be tied to gauge by iron or steel ties at suitable intervals.

Fixed diamond crossings must not be flatter than 1 in 8 except in special circumstances. Movable diamond crossings may be at any angle and are to be treated as worked facing points.

Lengthening Life of Track.

Apart from the use of the most suitable material and design for individual components the life of track as a whole for equal traffic and atmospheric conditions depends very largely upon the standard of maintenance attained and particularly to the attention it receives during the early period of its service life.

Good drainage is of prime importance and this demands clean ballast free from dirt and weeds and, where they exist, regular attention to drains, ditches and catchpits.

The maintenance of good top, regular alignment and gauge does much to prolong the life of the track. Rail joints need special attention and wear should be taken up, if necessary by the use

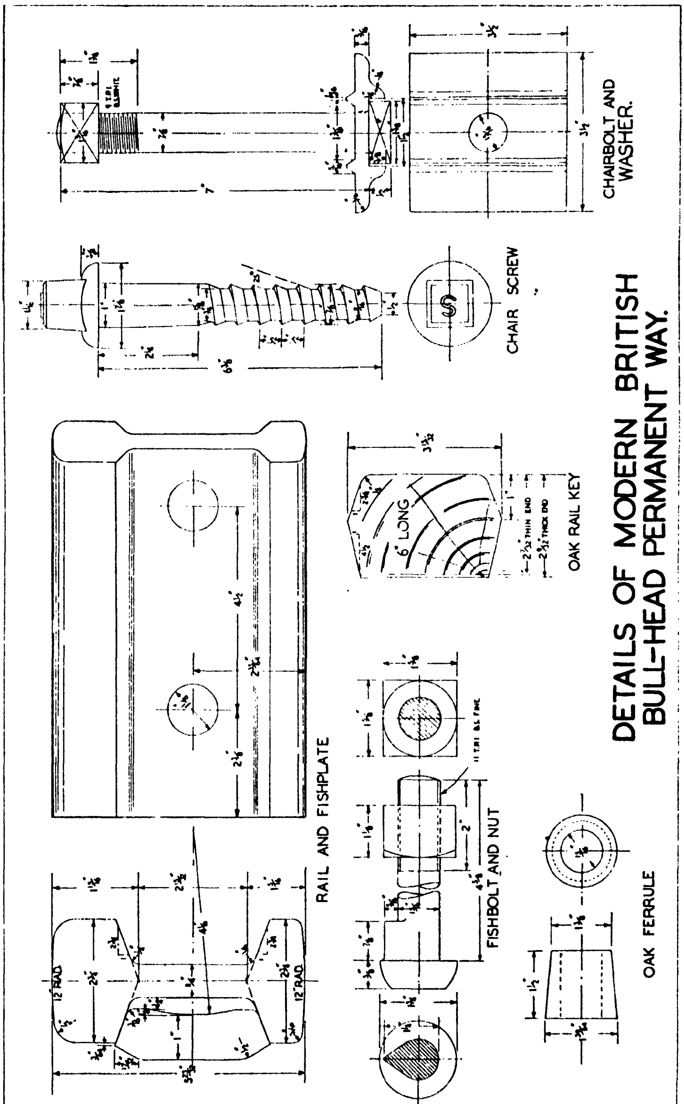

CHAIRBOLT AND WASHER.
CHAIR SCREW
6" LONG
OAK RAIL KEY
RAIL AND FISHPLATE
FISHBOLT AND NUT
OAK FERRULE
DETAILS OF MODERN BRITISH BULL-HEAD PERMANENT WAY.

Fig. 20.

DETAILS OF RAIL CHAIR AND FASTENING.

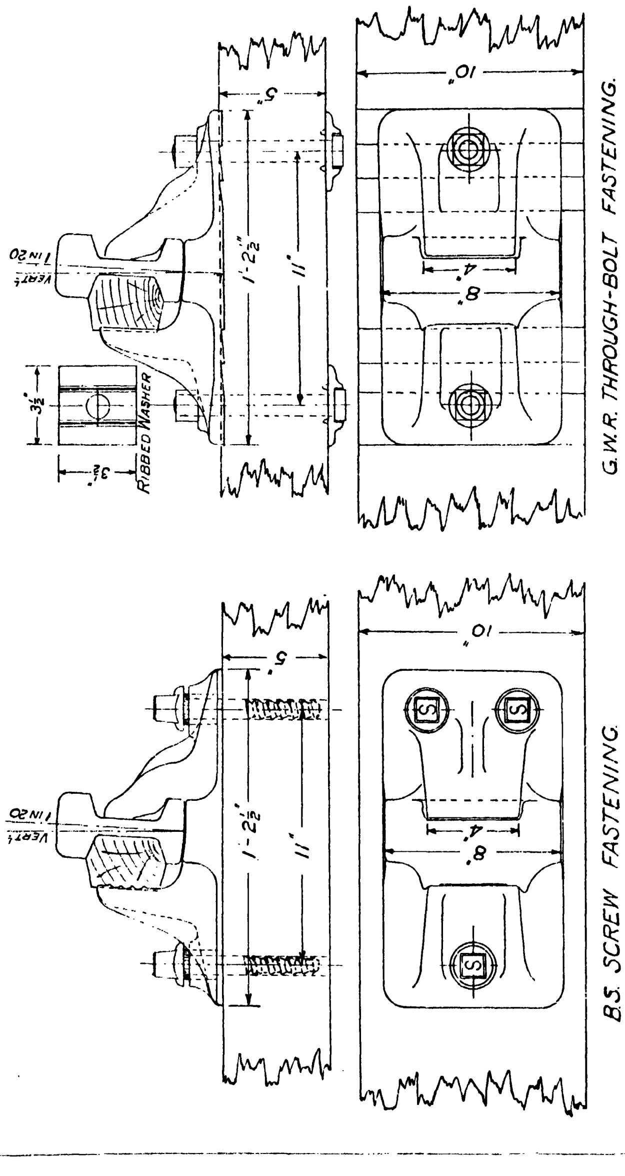
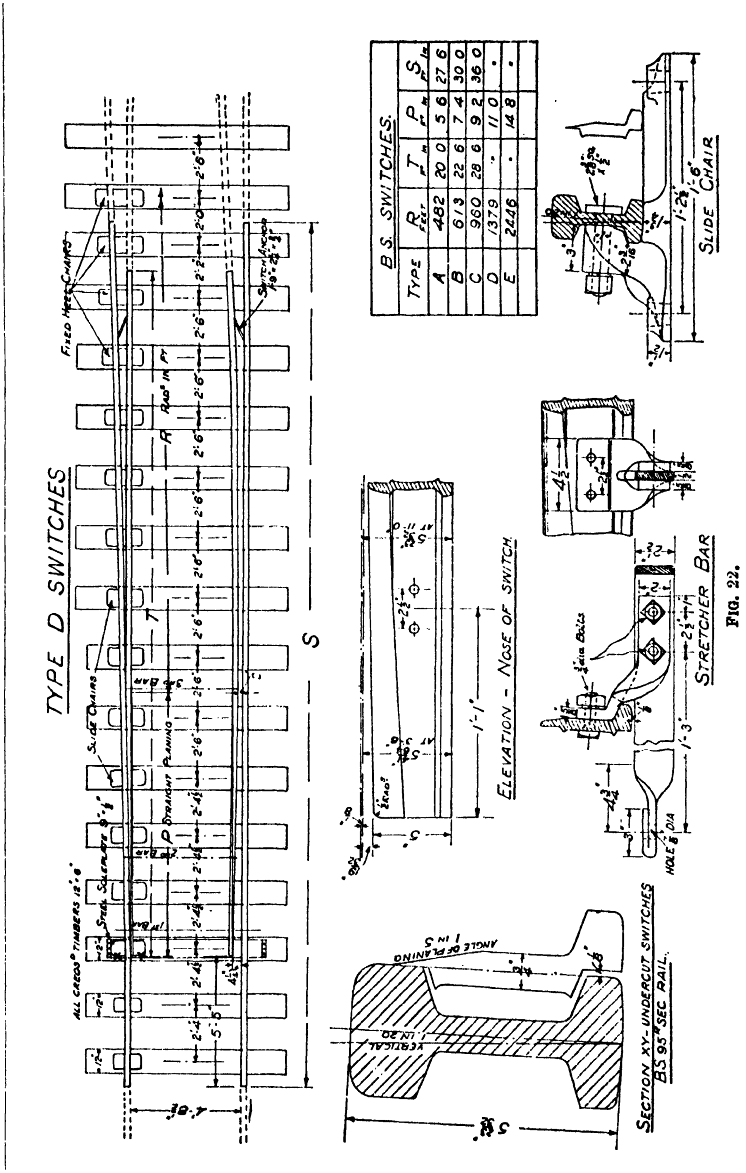


FIG. 21.



B.S. SWITCHES.				
TYPE	R FEET	T FT IN	P FT IN	S FT IN
A	482	20 0	5 6	27 6
B	613	22 6	7 4	30 0
C	960	28 6	9 2	36 0
D	1379	•	11 0	•
E	2446	•	14 8	•

FIG. 22.

of shims, and rail ends should be oiled or greased periodically to assist free expansion and contraction. Sleeper fastenings should be kept tight to minimise vertical movement between sleeper and chair or baseplate and any tendency for the sleepers to split should be corrected at an early stage if possible by the application of suitable dogs or clamps.

In the case of bull-head track wooden rail keys call for close attention in dry warm weather when they are liable to work loose due to shrinkage. New keys should be driven or the old ones re-driven with liners of the required thickness. Inefficient keying results in movement between rail and chair increasing galling of both components and causing gauge irregularities which in turn lead to intermittent side wear of the rails and lateral oscillation of rolling stock.

In some districts and especially in tunnels rail corrosion is a serious problem leading in some instances to failures attributable to corrosion fatigue—minute cracks developing at the bottom of corrosion pits due to concentration of stress.

Most railways have experimented from time to time with various methods designed to delay corrosion in such circumstances, *e.g.* painting with red lead, tarring, etc., and the Norfolk and Western Railroad (U.S.A.), has set up a special plant for spraying tunnel rails with Texas 45 Preservative Coating (containing 45 per cent. asphalt), following a thorough descaling by special oxy-acetylene flame heads. Joint bars (fishplates) are dealt with similarly.

The Great Western Railway of England is faced with a particularly difficult problem of rail corrosion in its Severn Tunnel—4 miles 624 yds. long and varying from 30 to 45 ft. from bed of river to crown. The normal practice here for many years has been to clean the new rails as thoroughly as possible by wire brushing, etc., and apply a coating of carbon tar paint, but a special test is in progress in which the rails were first flame cleaned, then chipped and wire brushed as necessary, followed whilst still warm by a coat of red lead brushed well in. This was followed by a second coating of a special tar mixture containing small proportions of tallow, lime and creosote. Similar coatings but without flame cleaning have been experimented with amongst other preservatives by the London and North Eastern Railway.

Cost of Railway Construction.

In England, taking a fairly representative section of one of the main lines, outside the Metropolis, and constructed prior to 1914, as an example, the percentages of the cost of about 40,000*l.* per mile, work out somewhat as follows :—

	Per Cent.		Per Cent.
Land and compensation	10	Sidings	3
Fencing	1½	Junctions and signals	1
Earthworks	24	Stations, including buildings (roadside only)	6½
Tunnels	12	Contingencies, including parliamentary, administration, legal and engineering expenses	6
Viaducts	8	Maintenance	½
Bridges for roads	9		
Accommodation works	2		
Culverts and drainage	5		
Permanent way, ballast, including main line	11½		
			100

From this it will be seen that, for country lines, the value of land does not form so important a factor in the total cost as is generally understood. The necessity for very easy gradients, and flat curves, in order to admit of high speeds, involves heavy earthworks, viaducts and tunnels, and expensive, well-ballasted permanent way, whilst stations and sidings have to be extensive to accommodate the traffic.

Experience has shown how to adapt the design of railways to their special surroundings. Their cost has varied from about 630,000*l.* per mile in London, and from an average of 52,378*l.* per mile in the United Kingdom, and of 8,127*l.* per mile in India, to a cost, in some parts of South Africa and South America, of 3,200*l.* to 3,500*l.* per mile, or, with equipment, of 4,000*l.* per mile—a rate at which, in new countries and under fairly normal conditions, railways can be provided, suitable for a speed of 30 miles per hour, and for heavy trains. (*Sir Douglas Fox.*)

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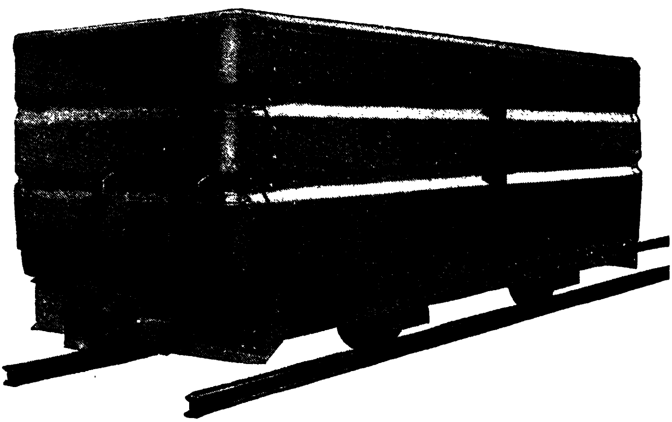

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Table of Gauges of Principal Railways in Various Countries.

Countries	Gauge
India, Ceylon, Spain, Portugal, Argentine, Chili	5 ft. 6 ins.
Russia, Estonia, Latvia, Finland, Manchuria	5 ft. 0 ins.
Great Britain, Canada, U.S.A., France, Belgium, Holland, Germany, Czecho Slovakia, Norway, Denmark, Sweden, Poland, Switzerland, Italy, Greece, Yugo-Slavia, Albania, Bulgaria, Rumania, Turkey, Mexico, Argentine, Peru, Uruquay, New South Wales, China, Manchuria, Japan, Palestine	Standard 4 ft. 8½ ins.
Palestine	3 ft. 5½ ins.
Ireland, South Australia, Victoria, Brazil	5 ft. 3 ins.
India, Ceylon, Egypt, Sierra Leone	2 ft. 6 ins.
India, Greece	2 ft. 0 ins.
South Africa, Gold Coast, Nigeria, Rhodesia, Nyasaland, Congo, Sudan, Central Australia, West ", North ", Queensland, Tasmania, New Zealand, Japan, Netherland East Indies, Manila, Newfoundland, Jersey	3 ft. 6 ins.
Federated Malay States, Burma, India, Tanganyika, Kenya, Uganda, Siam, Indo-China, Algeria, Brazil, Argentine, Chili, Iraq, Greece, France, Belgium	Metre.

Quantity of Land required for a Line of Railway.

This depends upon the width as determined by the depth of cuttings and height of embankments. Assuming average depth and height of 3 feet, and slopes of 1½ to 1, 5 to 6 acres will be required per mile of single line (4 ft. 8½ ins. gauge), made as in England, exclusive of sidings or stations. For each foot added to or deducted from the average of 3 feet given above (same slopes), 1 yard in width, or ⅜ acre per mile, must be added or deducted.

Light or Narrow-Gauge Railways.

Narrow-gauge railways will be found very suitable in districts where a standard-gauge line could not be made to pay, on account of the greater cost of construction and working of the latter. Narrow-gauge railways occupy less land, and the earthworks are of a less expensive kind

than for standard-gauge lines, as heavy embankments and cuttings can be avoided by adopting sharper curves than are admissible with the standard gauge.

For the permanent way light material can be used, the rolling stock can be constructed to give a more favourable proportion between the dead and the paying loads than is possible on standard-gauge lines, particularly if the traffic is made up of a great number of small consignments. On narrow-gauge lines the average proportion of dead load to paying load may be taken at about 1 to 2·33.

Whether steel or wood sleepers should be used is principally a question of cost, but it should be borne in mind that steel sleepers have a much greater life than wood sleepers.

The gauge and weight of rail should be determined in every case after a careful inquiry into the earning powers of the line. For the rails a weight of 25 lbs. per yard would be about the minimum. The 2-ft. gauge has the great advantage that it can be worked best in combination with portable railways which are joined on to branches of permanent line and continued to the farmyards, fields, or factories. The 2-ft. gauge is especially suitable for private lines and for lines of public utility with limited goods and little passenger traffic. It can be worked at a speed of fifteen miles per hour with curves 100 to 150 feet radius and gradients 1 in 80, for public lines; while on private lines, curves down to 60 feet radius and gradients up to 1 in 25 are admissible, according to the speed required and the amount of traffic. For lines with extensive goods traffic or principally for passenger traffic, 2 ft. 6 in. or 3 ft. gauge would be preferable.

If the goods are of a nature that they will not stand handling, the narrow-gauge wagon bodies could be made of such dimensions that three or four would fit into one main-line truck. A very simple device is also in practical use, of putting the main-line trucks on specially constructed bogies, and thus transporting them over the narrow-gauge railway.

(*A. Koppel.*)

The use of locomotives is not profitable on gauges of less than 18 inches.

The speed on narrow gauges may be as follows:—

2 ft. 0 in. gauge, 15 miles per hour.	3 ft. 0 in. gauge, 25 miles per hour.
2 ft. 6 in. gauge, 20 miles per hour.	3 ft. 6 in. gauge, 30 miles per hour.

Cost of Light Railways.

A fact which has militated against light railway construction is that the cost in Great Britain is very little less than that of building standard railways, ranging from £12,000 per mile in difficult country to £3,333 under easy conditions, while the average return on capital has been only a little over 2 per cent. On the other hand, the Decauville system of light railway adopted in the Argentine can be laid for £1,240 per mile, and the Road-Rail system, as tested in Uganda, South Africa, and Morocco, for £1,250 per mile. Another cheap form of rail transport is the Rutway, which it is said can on ordinary ground, such as the plains of India, be laid down, in lengths of 20 miles or more, at a cost of only about £1,000 per mile, including the provision of rolling stock. In this case the wheels run without any flanges on flat metal surfaces at the bottom of two deep ruts in the lowest part of the permanent way, and are confined to these ruts by the vertical sides of the rail angles and by the stone ballast heaped up along either side of each rail.

Superelevation of Outer Rail on Curves.

The effect of centrifugal force on vehicles travelling round a curve is usually counteracted, at least partially, by raising the outer rail of the curve above the level of the inner rail, thus giving the track a transverse slope.

If G = gauge (rail centres) in feet, V = speed of vehicle in miles per hour, R = radius of curve in feet, and e = amount of superelevation in inches, then theoretically

$$e = \frac{GV^2}{1 \cdot 25R}$$

In practice only three-quarters or seven-eighths of this cant is usually applied with a maximum of about 6 ins., or approximately one-tenth of the gauge, but it is now generally agreed that the desirable superelevation in any particular instance is not very critical provided the amount selected is attained from the level very gradually and maintained at a strictly uniform figure throughout the length of the curve. There is in fact a general tendency among railway engineers to-day to permit a deficiency of cant to the extent of from 2 to 3 ins. below the theoretical figure without a corresponding reduction in the permissible maximum speed.

The gain or loss of superelevation on high-speed lines should not be effected at a greater rate than 1 in 1200 where practicable, whilst 1 in 300 should be considered the absolute limiting cant gradient in all cases.

Whilst superelevation is desirable for reasons of comfort its safety value has in the past undoubtedly been largely over-rated and there are in existence many junctions which, for various reasons, have had to be laid without superelevation and which are regularly run over at speeds of 40 m.p.h. and over without noticeable discomfort or excessive rail wear.

A recent innovation consists of two-level rail chairs the use of which enables a certain amount of superelevation to be provided through junctions of fairly sharp curvature. In one such case this treatment has permitted a speed restriction of 20 m.p.h. to be relaxed to 40 m.p.h. These chairs were produced by Taylor Brothers (Sandiacre), Limited, in collaboration with the Chief Engineer of the former L.M. & S.R. Company.

The following is an extract from a table giving particulars of the superelevation provided for in the regulations of one of the Home railways according to the curvature, and speed of trains.

Radius of Curve. Chains.	Superelevation in Inches. Speed in m.p.h.						
	10.	20.	30.	40.	50.	60.	70.
5	1	4½	—	—	—	—	—
10	½	2⅛	4¾	—	—	—	—
15	—	1⅜	3⅛	5⅝	—	—	—
20	—	1	2⅜	4¼	—	—	—
25	—	⅞	1⅞	3⅜	5¼	—	—
30	—	¾	1⅝	2¾	4⅜	—	—
35	—	⅝	1⅜	2⅜	3¾	5⅝	—
40	—	½	1⅛	2⅛	3¼	4¾	—
60	—	—	¾	1⅜	2¼	3⅛	4¼
80	—	—	⅝	1	1⅝	2⅜	3¼
100	—	—	½	⅞	1¼	1⅞	2⅝
120	—	—	—	¾	1⅛	1⅝	2⅛
160	—	—	—	½	⅞	1⅛	1⅝
200	—	—	—	—	⅝	1	1¼

RADII OF CURVATURE AND VERSED SINES FOR RAILS.

Radius of Curvature in Feet.	Versed Sines for Rails, the Length in Feet being				Radius of Curvature in Feet.	Versed Sines for Rails, the Length in Feet being			
	15	16	18	20		15	16	18	20
	In.	In.	In.	Ins.		In.	In.	In.	In.
286	1 3/16	1 5/16	1 11/16	2 1/16	573	9/16	11/16	⅞	1 1/16
301	1⅛	1¼	1⅝	2	636	½	⅝	¾	15/16
318	1 1/16	1 3/16	1½	1⅞	716	½	9/16	11/16	13/16
337	1	1⅛	1 7/16	1¾	818	7/16	½	9/16	¾
358	15/16	1 1/16	1⅜	1 11/16	955	⅜	⅜	½	⅝
382	⅞	1	1¼	1 9/16	1,146	5/16	5/16	7/16	½
409	13/16	15/16	1 3/16	1 7/16	1,432	¼	¼	5/16	7/16
441	¾	⅞	1⅛	1⅜	1,910	3/16	3/16	¼	5/16
477	11/16	13/16	1	1¼	2,865	⅛	⅛	3/16	3/16
521	⅝	¾	15/16	1⅛	5,739	1/16	1/16	1/16	⅛

The above radii are those corresponding (commencing at the top) to angles of deflection from 20 to 1 deg.

Turnout from Straight Main Line.

(Turnout Curve Tangential to Straight.)

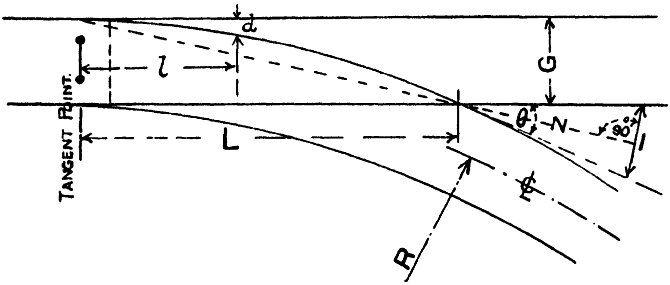


FIG. 23.

d = clearance at heel of switch (usually $4\frac{1}{2}$ ins. = 0·375 ft. for 4 ft. $8\frac{1}{2}$ ins. gauge).

$$L = 2GN = \sqrt{2RG}$$

$$l = \sqrt{(2R - d)d} = \sqrt{2Rd} \text{ approximately.}$$

$$N = \sqrt{\frac{R}{2G}} = \frac{L}{2G} = \frac{1}{2}\cot\frac{\theta}{2}$$

$$R = 2GN^2$$

The switch lengths for crossing angles from 1 in 4 to 1 in $6\frac{1}{2}$ may be about 2N, and for flatter crossings about $1\frac{1}{2}$N.

Curves with Contra-Flexure.

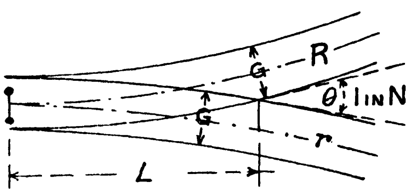


FIG. 24.

$$N = \sqrt{\frac{Rr}{2G(R + r + \frac{1}{2}G)}} = \sqrt{\frac{Rr}{2G(R + r)}} \text{ approximately.}$$

$$L = 2GN \qquad = \sqrt{\frac{2RrG}{R + r}} \text{ approximately.}$$

$$\theta = 2\cot^{-1}\sqrt{\frac{2Rr}{G(R + r + \frac{1}{2}G)}}$$

Curves with Similar Flexure.

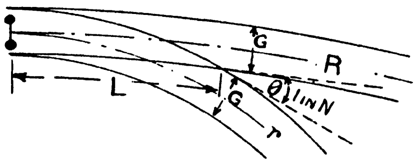


FIG. 25.

$$N = \sqrt{\frac{Rr}{2G(R - r - \frac{1}{2}G)}} = \sqrt{\frac{Rr}{2G(R - r)}} \text{ approximately.}$$

$$L = 2GN \qquad = \sqrt{\frac{2RrG}{R - r}} \text{ approximately.}$$

$$\theta = 2 \cot^{-1} \sqrt{\frac{2Rr}{G(R - r - \frac{1}{2}G)}}$$

Straight Track with Straight Switch and Curved Crossing.

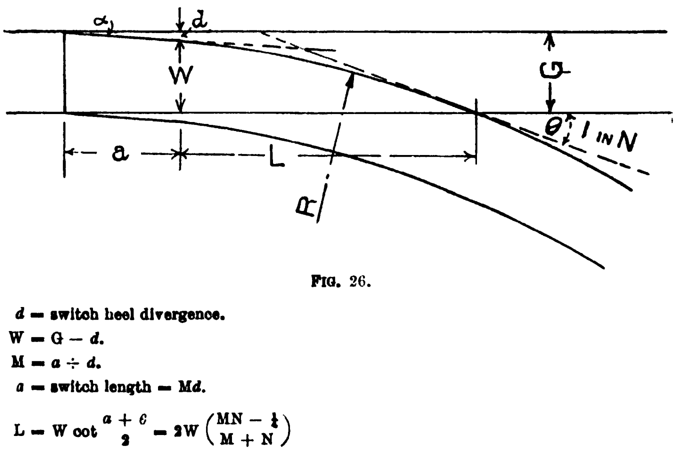


FIG. 26.

d = switch heel divergence.

$W = G - d.$

$M = a \div d.$

a = switch length = Md.

$$L = W \cot \frac{a + \theta}{2} = 2W \left(\frac{MN - \frac{1}{2}}{M + N}\right)$$

$$R = \frac{W}{\cos a - \cos \theta} = \frac{2WM^2N^2}{M^2 - N^2} \text{ (approximately)} = \frac{LMN}{M - N} \text{ approximately.}$$

$$N = M\sqrt{\frac{R - \frac{1}{2}W}{R + 2WM^2}} \text{ approximately.}$$

$$= \frac{LM + \frac{1}{2}W}{2WM - L} \text{ (approximately).}$$

$$\cos \theta = \cos a - \frac{W}{R}$$

$$N = \tfrac{1}{2} \cot \tfrac{1}{2}\theta.$$

Straight Track with Straight Switch and Straight Crossing.

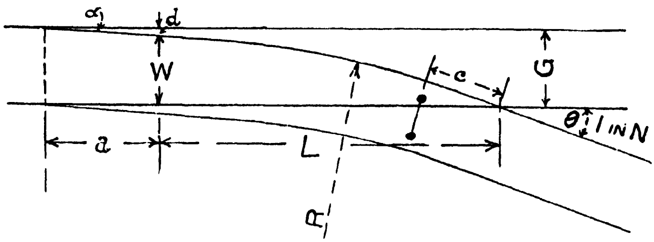


FIG. 27.

$$W_1 = W - c \sin\theta = G - \left(d + \frac{c}{N + \frac{1}{4N}}\right)$$

$$L = W_1 \cot\frac{\alpha + \theta}{2} + c\cos\theta = 2W_1\left(\frac{MN - \frac{1}{4}}{M + N}\right) + c\left(\frac{N - \frac{1}{4N}}{N + \frac{1}{4N}}\right)$$

$$R = \frac{W_1}{\cos\alpha - \cos\theta} = \frac{2W_1M^2N^2}{M^2 - N^2} \text{ approximately.}$$

$$\cos\theta = \cos\alpha - \frac{W_1}{R}$$

$$N = M\sqrt{\frac{R - \frac{1}{2}W_1}{R + 2W_1M^2}} = \tfrac{1}{2}\cot\tfrac{1}{2}\theta.$$

Crossover Roads.

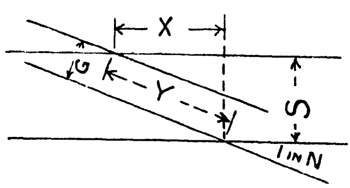


FIG. 28.

$$X = N(S - G) - \frac{S + G}{4N}$$

$$Y = N(S - G) + \frac{S + G}{4N}$$

This also applies when the tracks are curved if the length Y is curved to the same radius and the same sense as the main lines

Diamond Crossing with both Tracks Straight.

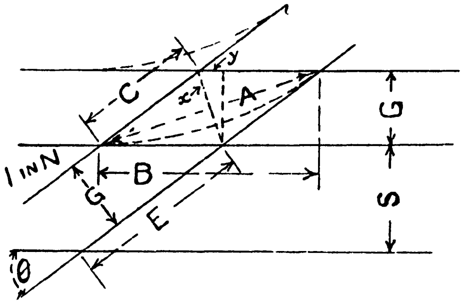


FIG. 29.

$$A = G\sqrt{4N^2 + 1} = G \operatorname{cosec} \tfrac{1}{2}\theta$$

$$C = G\left(N + \frac{1}{4N}\right) = G \operatorname{cosec} \theta$$

$$Y = \frac{G}{2N} = G \tan \tfrac{1}{2}\theta$$

$$X = \frac{A}{2N} = \frac{G\sqrt{4N^2 + 1}}{2N} = G \sec \tfrac{1}{2}\theta$$

$$E = S\left(N + \frac{1}{4N}\right) = S \operatorname{cosec} \theta$$

$$B = 2GN = G \cot \tfrac{1}{2}\theta$$

The radius (centre line) of the slip road shown dotted is $= 2GN^2$ feet.

Scissors Crossing – Straight Tracks.

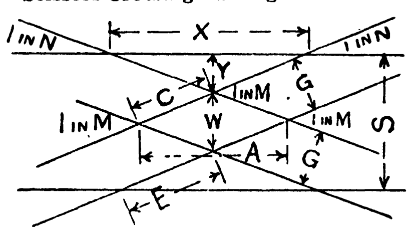


FIG. 30.

$$M = \frac{N}{2} - \frac{1}{8N} \qquad W = \frac{A}{2M} = S - 2Y$$

$$X = N(S - G) - \frac{S + G}{4N}$$

$$C = G\left(M + \frac{1}{4M}\right) \qquad A = G\sqrt{4M^2 + 1}$$

$$E = \frac{X}{2}\left(\frac{N + \frac{1}{4N}}{N - \frac{1}{4N}}\right) \qquad Y = \frac{X}{2}\left(\frac{1}{N - \frac{1}{4N}}\right)$$

Diamond Crossing. One Straight and One Curved Track.

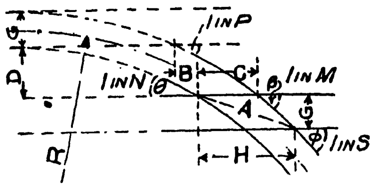


FIG. 31.

$$D = \frac{2R - G}{4N^2 + 1} = \frac{2R + G}{4M^2 + 1} - G = \frac{2R + G}{4S^2 + 1} - 2G$$

$$N = \sqrt{\frac{R - \frac{1}{2}(D + G)}{2D}} \qquad M = \sqrt{\frac{R - \frac{1}{2}D}{2(D + G)}}$$

$$S = \sqrt{\frac{R - \frac{1}{2}(D + G)}{2(D + 2G)}} \qquad P = \sqrt{\frac{R}{2G}}$$

$$C = 2M(D + G) - 2DN. \qquad H = 2S(D + 2G) - 2DN$$

$$A = \sqrt{G^2 + H^2} \qquad B = 2(DN - GP)$$

$$\theta = 2 \cot^{-1} 2N \qquad \phi = 2 \cot^{-1} 2S. \qquad \beta = 2 \cot^{-1} 2M.$$

Diamond Crossing, both Tracks Curved.

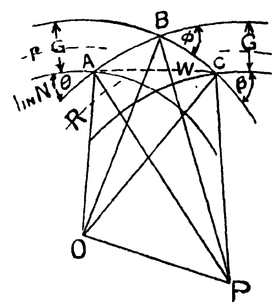


FIG. 32.

$$\cos\theta = \frac{N - \frac{1}{4N}}{N + \frac{1}{4N}} \qquad \sin\theta = \frac{1}{N + \frac{1}{4N}}$$

$$OP = \sqrt{OA^2 + PA^2 - 2 . OA . PA . \cos\theta}$$

$$\text{sin. angle } APO = \frac{OA}{OP} \sin\theta$$

$$\cos \tfrac{1}{2} \text{ angle } BPO = \sqrt{\frac{S(S - OB)}{OP . PB}}$$

$$\text{where } S = \tfrac{1}{2}(OB + OP + PB)$$

$$AB = \frac{AP}{57 \cdot 3} \times (\text{angle } BPO - \text{angle } APO)$$

$$\sin \phi = \frac{OP}{OB} \sin \text{angle } BPO$$

$$\cos \tfrac{1}{2} \text{ angle } OOP = \sqrt{\frac{S'(S' - PO)}{OP.OO}}$$

$$(\text{where } S' = \tfrac{1}{2}(OO + PO + OP))$$

$$\text{Angle } BOP = 180^\circ - (\phi + \text{angle } BPO)$$

$$BO = \frac{OB}{57 \cdot 3} (\text{Angle } BOP - \text{angle } OOP)$$

$$\sin \beta = \frac{OP}{PO} \sin \text{angle } COP$$

$$\text{Angle } AOO = \text{angle } AOP - \text{angle } COP$$

$$W^2 = OA^2 + OO^2 - 2. \quad OA.OO.\cos \text{angle } AOO.$$

TABLES OF LEADS FROM TANGENT POINT AND ANGLES OF CROSSINGS CURVING OFF A STRAIGHT LINE.

I. GAUGE, 5 FEET 6 INCHES.

Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, 1 in N.	Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, 1 in N.
400	66	1 in 6·0	1000	105	1 in 9·5
450	70	„ 6·4	1050	107	„ 9·8
500	74	„ 6·7	1100	110	„ 10·0
550	78	„ 7·0	1150	112	„ 10·2
600	81	„ 7·4	1200	115	„ 10·4
650	84	„ 7·7	1250	117	„ 10·6
700	88	„ 7·9	1300	120	„ 10·9
750	91	„ 8·2	1350	122	„ 11·0
800	94	„ 8·5	1400	124	„ 11·2
850	97	„ 8·7	1450	126	„ 11·5
900	99	„ 9·0	1500	128	„ 11·7
950	102	„ 9·3	1550	131	„ 11·9

II. GAUGE, 3 FEET.

Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, 1 in N.	Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, 1 in N.
400	49	1 in 8·1	1000	78	1 in 12·8
450	52	„ 8·6	1050	79	„ 13·2
500	55	„ 9·0	1100	81	„ 13·5
550	58	„ 9·6	1150	83	„ 13·8
600	60	„ 10·0	1200	85	„ 14·1
650	63	„ 10·3	1250	87	„ 14·4
700	65	„ 10·7	1300	88	„ 14·7
750	67	„ 11·1	1350	90	„ 15·0
800	69	„ 11·5	1400	92	„ 15·2
850	71	„ 11·9	1450	93	„ 15·5
900	74	„ 12·2	1500	95	„ 15·7
950	76	„ 12·5	1550	96	„ 16·1

III. GAUGE, 3 FEET 3⅜ INCHES (1 METRE).

Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, 1 in N.	Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, 1 in N.
400	51	1 in 7·8	1000	81	1 in 12·3
450	54	„ 8·3	1050	83	„ 12·6
500	57	„ 8·7	1100	85	„ 12·9
550	60	„ 9·1	1150	87	„ 13·2
600	63	„ 9·5	1200	89	„ 13·4
650	65	„ 10·0	1250	91	„ 13·7
700	68	„ 10·3	1300	92	„ 14·1
750	70	„ 10·7	1350	94	„ 14·2
800	73	„ 10·9	1400	96	„ 14·5
850	75	„ 11·3	1450	98	„ 14·8
900	77	„ 11·6	1500	99	„ 15·1
950	79	„ 12·0	1550	101	„ 15·3

IV. GAUGE, 4 FEET 8½ INCHES.

Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, 1 in N.	Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, 1 in N.
400	61	1 in 6·5	1000	97	1 in 10·3
450	65	„ 6·9	1050	99	„ 10·6
500	69	„ 7·3	1100	102	„ 10·8
550	72	„ 7·6	1150	104	„ 11·0
600	75	„ 8·0	1200	106	„ 11·3
650	78	„ 8·3	1250	108	„ 11·5
700	81	„ 8·6	1300	111	„ 11·7
750	84	„ 8·9	1350	113	„ 12·0
800	87	„ 9·2	1400	115	„ 12·2
850	89	„ 9·5	1450	117	„ 12·4
900	92	„ 9·8	1500	119	„ 12·6
950	95	„ 10·0	1550	121	„ 12·8

V. GAUGE, 5 FEET 3 INCHES.

Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, 1 in N.	Radius of Curve (R) in Feet.	Lead (L) in Feet.	Inclination of Crossing, 1 in N.
400	65	1 in 6·1	1000	102	1 in 9·8
450	69	„ 6·5	1050	105	„ 10·0
500	72	„ 6·9	1100	107	„ 10·2
550	76	„ 7·2	1150	110	„ 10·4
600	79	„ 7·5	1200	112	„ 10·7
650	83	„ 7·8	1250	115	„ 10·9
700	86	„ 8·1	1300	117	„ 11·1
750	89	„ 8·4	1350	119	„ 11·3
800	92	„ 8·7	1400	121	„ 11·5
850	94	„ 9·0	1450	123	„ 11·7
900	97	„ 9·2	1500	125	„ 11·9
950	100	„ 9·5	1550	128	„ 12·1

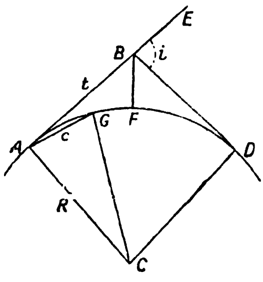


FIG. 33.

RANGING OUT CURVES.

Ranging out Curves with a Theodolite.

R = radius of curve = AC. i = angle of inclination of intersection = EBD. t = tangent AB or BD = R tan $\frac{i}{2}$.

Length of curve = AFD = ·0002909Ri.

Tangential (or 'setting off') angle (angle between c and t) in minutes for any chord c

$$= 1718{\cdot}873\ \frac{c}{R}.$$

R and c being expressed in feet or chains and i in minutes.

The angle ACG is called the 'deflection' angle, and is twice the tangential angle, *i.e.* the tangential angle is one-half the deflection angle.

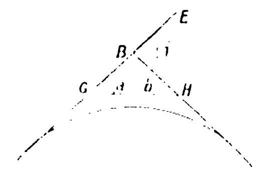


FIG. 34.

To find the Angle of Intersection and Point of Intersection where the latter is inaccessible.

If B is the inaccessible point, then—

Range the line GH.

$$i = a + b,$$

$$GB = \frac{GH \sin b}{\sin i}, \quad HB = \frac{GH \sin a}{\sin i}$$

TABLE OF TANGENTIAL ANGLES FOR CHORDS OF 1 FOOT AND 100 FEET.

Radius of Curve.	Tangential Angle. Chord 1 ft.	Tangential Angle. Chord 100 ft.	Radius of Curve.	Tangential Angle. Chord 1 ft.	Tangential Angle. Chord 100 ft.	Radius of Curve.	Tangential Angle. Chord 1 ft.	Tangential Angle. Chord 100 ft.
Ft.	Min.	Deg. Min.	Ft.	Min.	Deg. Min.	Miles.	Min.	Deg. Min.
500	3·438	5 43·77	3,000	0·573	0 57·30	⅛	2·604	4 20·44
600	2·865	4 46·48	3,500	0·491	0 49·11	¼	1·302	2 10·22
700	2·456	4 5·55	4,000	0·430	0 42·97	½	0·651	1 5·11
800	2·149	3 34·86	4,500	0·381	0 38·20	¾	0·434	0 43·41
900	1·910	3 10·99	5,000	0·344	0 34·38	1	0·326	0 32·55
1,000	1·719	2 51·89	5,500	0·313	0 31·25	1¼	0·260	0 26·04
1,100	1·563	2 36·26	6,000	0·286	0 28·65	1½	0·217	0 21·70
1,200	1·432	2 23·24	7,000	0·246	0 24·56	1¾	0·187	0 18·70
1,500	1·146	1 54·59	8,000	0·215	0 21·49	2	0·163	0 16·28
2,000	0·859	1 25·94	9,000	0·191	0 19·10	2½	0·130	0 13·02
2,500	0·688	1 8 76	10,000	0·172	0 17·19	3	0·109	0 10·85

A curve in America is expressed as an $n°$ curve (deflection angle), and is one of such a radius that a chord of 100 ft. subtends an angle of $n°$ at the centre.

$$R \text{ in feet} = \frac{5730}{n°}$$

Setting out Curves by Offsets.

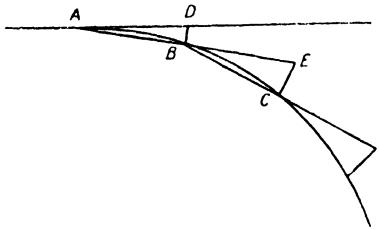


FIG. 35.

First offset BD from the tangent line.

$$DB = \frac{AD^2}{2\ \text{radius}}.$$

Offset from any chord AB produced

$$\text{to E} = EC = \frac{AE \times EB}{2\ \text{radius}}.$$

Or where AB = BE, then

$$EC = \frac{AB^2}{\text{radius}}.$$

Transition Curves.

When a train travels round a circular curve radial acceleration varies directly as the square of the speed and inversely as the radius, disappearing entirely when the radius is infinite, that is, when the track is straight. A transition curve or easement spiral (fig. 36) is, therefore, inserted between a tangent and a regular circular curve or between two curves of the same 'hand' but differing in radius in order that the radial acceleration shall be attained gradually in the length of the transition curve. The absence of suitable transition curves not only gives rise to very uncomfortable running conditions but also makes it impossible to maintain the track in its proper alignment.

Many forms of spiral have been advocated for this purpose but the curve which best fulfils all the requirements of a true transition is the spiral, the intrinsic equation of which is :—

$$l = m\sqrt{\phi}$$

It can be shown, however, that for the majority of railway purposes the cubic parabola is a very close approximation and can be set out by means of rectangular offsets from the tangent by the formulae shown in fig. 36.

The length L can be determined by one or other of the following methods :—

(*a*) If, as usual, it is required to run up a given amount of superelevation required on the circular arc in the transition length, the latter (in feet) should not be less than $V^3/17\ R$, where V is in miles per hour and R is measured in chains. This formula provides for the superelevation being attained at a maximum rate of $1\frac{1}{2}$-in. per second, which is generally recognised as reasonable.

(*b*) If there is no superelevation to be considered L should not be less than $V^3/21\ R$. This assumes that radial acceleration can be increased without discomfort at the rate of 1 ft. per second per second per second.

When using the formulae of fig. 36 it will be found that where the ratio R/L is much less than say 4, difficulties in setting out arise, especially near the junction of the transition with the circular

curve. This is due to the approximations employed in the derivation of the usual formulae and in such instances more accurate formulae should be used to ensure the best results. Such formulae are not difficult to derive from the equation to the cubic parabola but are somewhat cumbersome. If, however, 'c' is substituted for the ratio R/L the following simple relations can be established.

$$p = \frac{(4c^2 - 1)^{\frac{3}{2}}}{8c^3} \qquad Y = \frac{1}{6p} L = qL$$

$$D = \frac{c}{\sqrt{1 - 4p^2}} L = mL$$

$$S = (q + 2mp - c)L = nL$$

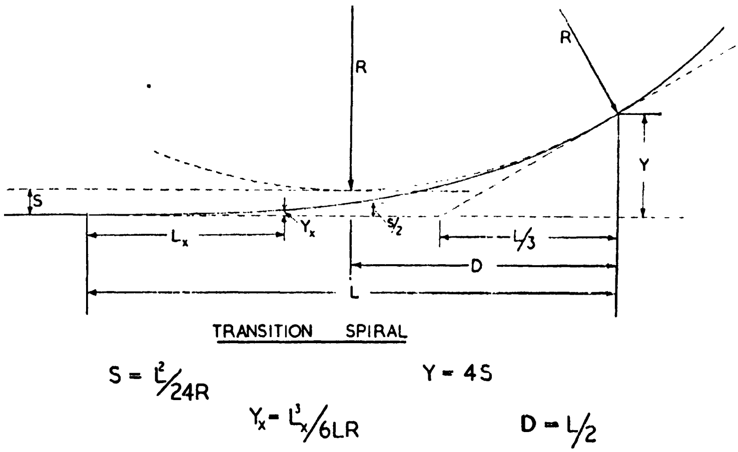


FIG. 36.

From this the following table of factors may be calculated for obtaining Y, D and S for selected values of '*c*.' To use the table select the nearest suitable ratio '*c*' for the transition to be set out and, if necessary, adjust the length L slightly so that the resulting ratio R/L corresponds exactly with one of the ratios in the table.

Ratio R/L.	Multiply L by these Factors to Obtain		
c	Y	S	D
1·5	0·132583	0·02647	0·5544
1·75	0·108215	0·02270	0·5404
2·0	0·091804	0·02001	0·5311
2·25	0·079920	0·01179	0·5246
2·5	0·070876	0·01621	0·5200
2·75	0·063741	0·01480	0·5165
3·0	0·057953	0·01360	0·5139
3·5	0·049115	0·01173	0·5102
4·0	0·042663	0·01030	0·5078
4·5	0·037734	0·00917	0·5062
5·0	0·033840	0·00828	0·5050
6·0	0·028070	0·00691	0·5035
7·0	0·023909	0·00594	0·5026
8·0	0·020956	0·00520	0·5020
9·0	0·018605	0·00462	0·5015
10·0	0·016729	0·00416	0·5013

It has been shown that the radius of a curve may be reduced 15 per cent. without the necessity for introducing a transition between the original and sharpened curves.

The alignment and superelevation for all transition curves should be marked on the ground by means of permanent concrete monument blocks so that the transition may be maintained to correct shape and reproduced accurately following renewal of the track.

Vertical Curves for Joining Gradients.

For important railways, rates of change of 0·1 per 100 ft. on summits, and 0·05 in sags should not be exceeded. On minor railways 0·2 per 100 ft. on summits and 0·1 in sags might be used.

The length of curve should be about 600 ft. at summits, and 800–1,200 ft. in sags.

(American Railway Engineering Association.)

Track Circuits—Insulated Rail Joints.

The steadily increasing mileage of track circuits installed in recent years has resulted in the necessity for providing a larger number of insulated rail joints.

Numerous arrangements have been experimented with from time to time but until a few years ago block joints on the whole consisted of ordinary steel fishplates machined on the bearing surfaces to permit the insertion of thin fibre channels between the plates and rails, the gap between adjoining rails being filled by a suitably shaped fibre end post. The fishbolt holes in the rails were enlarged to take fibre ferrules surrounding the bolts.

In heavily worked yards or where atmospheric conditions were unfavourable this type of joint gave a short life and the fibre insulation required frequent renewal.

Alternative joints now used to some extent are illustrated in figs. 37 and 38.

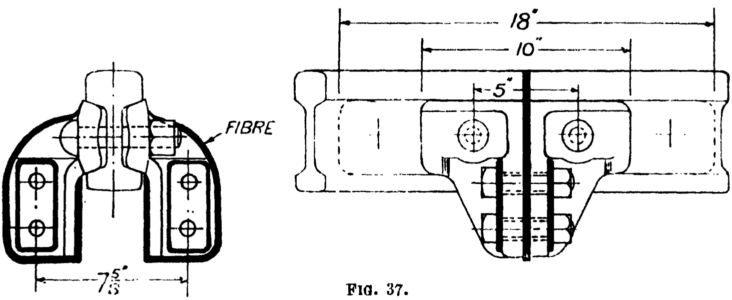


FIG. 37.

Fig. 37 is the Henry Williams drop-forged fishplate in which the insulating fibre is not subject to mechanical wear, the load on the joint being transmitted by means of side flanges secured together by bolts.

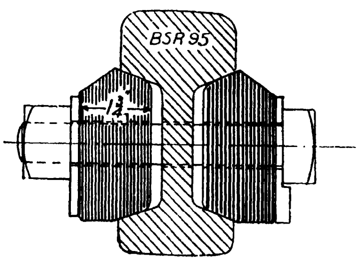


FIG. 38.

Fig. 38 shows the cross section of a pair of laminated wooden fishplates manufactured from 'Permali,' an efficient insulator made from compressed laminations of various woods impregnated with synthetic resin. This type requires no fibres and both designs obviate the use of bushes in the fishbolt holes, thus being easier and cheaper to install.

WEAR OF RAILS—CONTOUR OF TOP TABLE.

A comprehensive survey of worn rails of varying sections carried out by the Great Western Railway a few years ago led to the conclusion that whatever the original profile of the head might have been it sooner or later conformed to the average curvature shown in fig. 39.

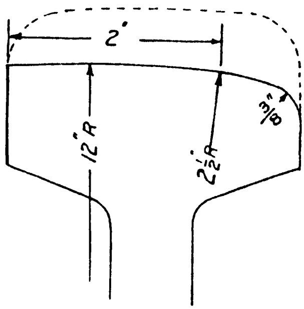


FIG. 39.

As will be seen the worn curves are considerably sharper than the 12-in. radius of a new B.S.95 R.B.H. rail, and this fact probably accounts for the bad running due to bogie hunting sometimes experienced when high speeds are attained over straight track recently relaid with new materials.

In 1935 the G.W.R. designed a new rail section (95½ lb. per yd.) with a modified head to conform as far as practicable with the curvature of this average worn rail and later embodied the same alteration in a new 100-lb. rail.

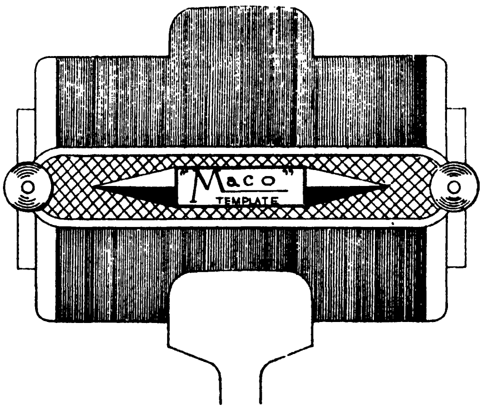


FIG. 40.

The profiles of worn rails may conveniently be ascertained by means of the 'Maco' Adjustable Template (The P. & M. Co. (England), Ltd.) illustrated in fig. 40. This comprises innumerable very fine strips of brass securely clamped in a metal framework and capable of following practically any irregular outline. The device gives male and female profiles simultaneously and pencil sections may conveniently be obtained by drawing round the template on to a suitably backed sheet of paper.

New Standard Rails—British Railways

Early in 1949 the British Transport Commission approved a recommendation from the Railway Executive to adopt flat bottom rails as the future standard for British Railways. Two sections have been designed, viz. 109 lb. for main lines and 98 lb. for less important routes, and the following table gives their salient features compared with the nearest sections in B.S.S. 11–1936.

		109 lb.	98 lb.	B.S. 110	B.S. 100	B.S. 95A
Rail—						
Overall depth	ins.	$6\frac{1}{4}$	$5\frac{5}{8}$	$6\frac{1}{4}$	6	$5\frac{13}{16}$
Width of foot	"	$5\frac{1}{2}$	$5\frac{1}{2}$	6	$5\frac{3}{4}$	$5\frac{9}{16}$
Width of head	"	$2\frac{3}{4}$	$2\frac{3}{4}$	$2\frac{7}{8}$	$2\frac{3}{4}$	$2\frac{11}{16}$
Depth of head	"	$1\frac{15}{16}$	$1\frac{13}{16}$	$1\frac{29}{32}$	$1\frac{27}{32}$	$1\frac{25}{32}$
Depth of foot	"	$1\frac{3}{16}$	$1\frac{1}{8}$	$\frac{29}{32}$	$\frac{27}{32}$	$\frac{53}{64}$
Min. web thickness	"	$\frac{5}{8}$	$\frac{9}{16}$	$\frac{19}{32}$	$\frac{9}{16}$	$\frac{9}{16}$
Fishing Angles—						
Top	1 in.	$2\frac{3}{4}$	$2\frac{3}{4}$	3	3	3
Bottom	1 in.	$2\frac{3}{4}$	$2\frac{3}{4}$	6	6	6
Sect. Area	sq. ins.	10·71	9·61	10·798	9·811	9·331
Weight	lb./yd.	109·1	97·9	110·01	99·95	95·06
M. of I. xx	ins.[4]	55·9	40·8	57·17	48·08	43·14
M. of I. yy	"	9·67	8·78	12·17		
Modulus-top	ins.[3]	16·89	13·77	17·41	15·37	14·22
Modulus-bottom	"	19·02	15·3	19·2	16·47	15·50
Height of N. Axis	ins.	2·94	2·66	2·97	2·87	2·783
Ratio—						
Depth/Width		1·136	1·023	1·041	1·043	1·033
M.I./weight		0·512	0·417	0·519	0·481	0·453
Mod./weight		0·154	0·141	0·158	0·154	0·149
Fishplates—						
Length	ins.	20	20	20	18	18
Weight (pair)	lb.	51·58	44·29	47·5	39	36
Fishbolt dia.	ins.	1	1	$1\frac{1}{8}$	1	1
Centres of holes	"	5	5	5	$4\frac{1}{2}$	$4\frac{1}{2}$
No. of fishbolts		4	4	4	4	4

SECTION XXXII

PART II

LOCOMOTIVES AND ROLLING STOCK.

LOCOMOTIVES — RAIL MOTOR CARS — CARRIAGES AND WAGONS—BRAKES.

(Revised by H. Holcroft, Member of Council, Institution of Locomotive Engineers.)

STEAM LOCOMOTIVES.

Some Recent Additions to British Locomotives.

No standard types have as yet been produced under the unification of the four Groups as British Railways. In the meanwhile each Region is continuing the construction of existing designs by the former Companies.

For the heaviest main line work the London Midland Region has adopted a 4–6–2 or ' Pacific ' type locomotive with four high-pressure cylinders. The Eastern and Southern Regions also use the same type, but with three high-pressure cylinders. Otherwise the bulk of the heavy work on British railways is performed by the 4–6–0 type, with two, three or four cylinders and wheels about 6 ft. 8 ins. diameter. The S.R. and E.R. have a number of powerful engines of the 4–4–0 type, with three high-pressure cylinders. The L.M.R. has many engines of the same type, but they are three-cylinder compounds.

For mixed traffic all the British railways have engines of the 4–6–0 type with two cylinders and wheels about 6 ft. diameter. The E.R. has also three-cylinder engines of the 2–6–2 type, and the S.R. two- and three-cylinder engines of the 2–6–0 type with 6 ft. diameter wheels.

All the British railways have a large number of fast freight engines of the 2–6–0 type with wheels about 5 ft. 6 ins. diameter, both two- and three-cylinder. The Western Region is, however, now building a two-cylinder 4–6–0 type with this wheel diameter instead of the 2–6–0 type. The S.R. have also a number of similar engines.

The heaviest mineral traffic on the E.R. is run by three-cylinder engines of the 2–8–2 or 2–8–0 types. Two-cylinder engines of the 2–8–0 type with wheels about 4 ft. 8 in. diameter have been uilt by the L.M.R. and W.R.

The 0–6–0 type is still being built by British railways in the form of a superheater engine with inside cylinders. The greater part of the freight traffic is handled by this type, and many of the older engines have been modernised by re-building and superheating.

Passenger tank engines of the 2–6–4 type, both two- and three-cylinder, have been constructed by the L.M.R. Other three-cylinder engines for freight service are those of the 2–6–4 type on the S.R. and the 2–6–2 type on the E.R. The latter is now putting into service two-cylinder tanks of the 2–6–4 type. The L.M.R. and the W.R. build the 2–6–2 type with two cylinders.

The ' Garratt ' articulated type of locomotive, which is extensively used on colonial railways, has been adopted to a limited extent on the L.M.R. for hauling heavy mineral traffic. An engine of this type, but having two sets of three cylinders, is in use on the E.R. for banking heavy mineral trains up a steep gradient.

Although no radical departures from normal locomotive design have been made on British railways, various experimental locomotives, having novel features, have been tried from time to time. The L.M.R. has a 4–6–2 type driven by a high-pressure turbine through reduction gearing. No condensing is provided in this case, the turbine exhausting at a pressure slightly above atmospheric through the blast pipe in the smoke-box.

[*cont. on p.* 522.

TENDER LOCOMOTIVES—

Type.	Class.	Service.	Cylinders. No.	Cylinders. Position.	Cylinders. Size.	Wheels. Bogie.	Wheels. Coupled.	Wheels. Trailing.	Wheel Base. Coupled.	Wheel Base. Total.
LONDON MIDLAND REGION.										
					″ ″	′ ″	′ ″	′ ″	′ ″	′ ″
4-4-0	2P	Pass.	2	In.	19 × 26	3 6½	6 9	—	9 6	22 8½
4-4-0	4P	Pass.	{1 2	In. H.P. Out. L.P.	19 × 26 21 × 26	3 6½	6 9	—	9 6	24 3
0-6-0	4F	Goods	2	In.	20 × 26	—	5 3	—	16 6	16 6
0-10-0	—	Banking	4	In & out.	16¾ × 28	—	4 7½	—	20 11	20 11
2-6-0	5F	Mxd. traf.	2	Out.	18 × 28	3 3½	5 6	—	16 6	25 6
4-6-0	Royal Scot	Hvy. pass.	3	In. & out.	18 × 26	3 3½	6 9	—	15 4	27 6
0-8-0	7F	Mineral	2	In.	19½ × 26	—	4 8½	—	18 3	18 3
2-8-0	8F	Freight	2	Out.	18½ × 28	3 3½	4 8½	—	17 3	26 0
4-6-0	5P 5F	Mxd. traf.	2	,,	18½ × 28	3 3½	6 0	—	15 0	27 2
4-6-0	5XP	Express	3	In. & out.	17 × 26	3 3½	6 9	—	15 4	27 7
4-6-2	Princess	Express	4	In. & out.	16¼ × 28	3 0	6 6	3 9	15 3	37 9
4-6-2	Duchess	Express	4	In. & out.	16½ × 28	3 0	6 9	3 9	14 6	37 0
2-6-0	4	Freight	2	Out.	17½ × 26	3 0	5 3	—	15 4	24 1
EASTERN & NORTH EASTERN REGIONS.										
2-6-0	K^3	Mxd. traf.	3	In. & Out.	18½ × 26	3 2	5 8	—	16 3	25 2
2-8-0	O^2	Mineral	3	In. & out.	18½ × 26	2 8	4 8	—	18 6	27 2
2-8-2	P^1	Freight	3	In & out.	20 × 26	3 2	5 2	3 8	18 6	36 2
4-6-2	A^3	Hvy. pass.	3	In. & out.	19 × 26	3 2	6 8	3 8	14 6	35 9
4-4-0	D^{11}	Pass.	2	In.	20 × 26	3 6	6 9	—	10 0	25 3
4-6-0	B^{12}	Hvy. pass.	2	In.	20 × 28	3 3	6 6	—	14 0	28 6
4-4-0	D^{16}	Pass.	2	In.	19 × 26	3 9	7 0	—	9 0	23 6
0-8-0	Q^7	Mineral	3	In. & out.	18½ × 26	—	4 7½	—	18 6	18 6
0-6-0	J^{39}	Goods	2	In.	20 × 26	—	5 2	—	17 0	17 0
4-4-0	Shire	Pass.	3	In. & out.	17 × 26	3 1½	6 8	—	10 0	24 11
4-6-0	Sandringham	Hvy. pass.	3	In. & out.	17½ × 26	3 2	6 8	—	16 3	27 9
4-6-0	B^1	Mxd. traf.	2	Out.	20 × 26	3 2	6 2	—	16·3	28 0
2-6-0	K^4	Mxd. traf.	3	In. & out.	18½ × 26	3 2	5 2	—	16 3	25 2
2-6-2	V^2	Fast freight	3	In. & out.	18½ × 28	3 2	6 2	3 8	15 6	33 8
4-6-2	A^4	Str'nl'd ex.	3	In. & out.	18½ × 26	3 2	6 8	3 8	14 6	35 9
4-6-2	A^1	Express	3	In. & out.	19 × 26	3 2	6 8	3 8	14 6	36 3

BRITISH RAILWAYS.

Working Pressure.	Grate Area.	Heating Surface. Tubes.	Heating Surface. Fire-box.	Heating Surface. Super-heater	Weight of Engine. On Coupled Wheels.	Weight of Engine. Total.	Tender. Coal	Tender. Water.	Engine and Tender. Total Weight.	Engine and Tender. Total Wheel Base.	Tractive Effort at 85 per cent. of Boiler Pressure.
lb. sq. in.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	T. C.	T. C.	Tons.	Galls.	T. C.	′ ″	lb.
180	21·1	1,034	124	246	34 19	54 1	4	3,500	95 5	44 9¾	17,729
200	28·4	1,170	147	272	39 4	61 14	5½	3,500	104 8	47 1½	22,649 (Simple)
175	21·1	1,034	124	216	48 15	48 15	4	3,500	89 19	38 9½	24,555
180	31·5	1,560	158	416	73 12	73 12	4	2,050	105 2	46 3¾	43,313
225	27·8	1,479	155	232	59 10	69 2	5	3,500	111 6	49 6	26,288
250	31·25	1,667	195	318	61 0	83 0	9	4,000	137 13	54 5½	33,150
200	23·6	1,402	150	312	60 15	60 15	4	3,500	101 19	41 10½	29,747
225	28·6	1,479	171	230	63 2	72 2	9	4,000	125 15	52 7¾	32,438
225	28 6	1,479	171	318	51 5	72 2	9	4,000	125 15	53 2¾	25,455
250	31·25	1,667	195	318	61 10	82 0	9	4,000	135 13	54 4¾	29,590
250	45·0	2,299	217	598	67 10	104 10	10	4,000	159 3	63 10	40,300
250	50	2,577	230	830	66 19	105 5	10	4.000	161 12	62 11	40,000
225		1,221	131		49 12	59 2	4	3,500	99 8	46 11¾	24,172
180	28·0	1,719	182	407	60 17	73 12	7½	4,200	124 12	49 1	30,031
180	27·5	1,868	163	430	69 7	78 13	7½	4,200	130 13	53 3	36,470
180	41·2	1,880 835	215	525	71 10	100 0	7	4,700	151 8	59 8	38,500 47,000 with Booster
220	41·2	2,477	215	703	66 3	96 5	8	5,000	154 3	60 10⅝	32,909
180	26·6	1,388	155	209	39 16	61 3	6	4,000	109 9	48 8½	19,644
180	31·0	1,405	154	315	48 2	69 10	4	3,670	108 16	48 3	21,970
180	21·0	1,303	126	204	36 9	55 18	5	3,450	95 3	43 8	17,096
180	27·0	1,663	166	392	71 12	71 12	5½	4,125	115 14	44 3⅞	36,910
180	26·0	1,226	171	272	57 17	57 17	5½	3,500	102 1	40 5⅛	25,664
180	26·0	1,226	171	272	42 0	66 0	6	4,000	114 6	47 10⅛	21,556
180	27·5	1,508	168	344	54 7	77 5	7½	4,200	129 5	51 0	22,812
225	27·9	1,508	168	344	52 10	71 3	7½	4,200	123 3	51 2⅜	26,878
200	27·5	1,254	168	310	57 18	68 8	5½	3,500	112 12	48 7	36,598
220	41·25	2,216	215	679	65 12	93 2	7½	4,200	145 2	56 2½	33,730
250	41 25	2,345	231	749	66 0	102 19	9	5,000	167 18	60 10⅝	35,455
250	50·0	2,216	245	680	66 0	101 2	9	5,000	164 9	62 5½	37,397

TENDER LOCOMOTIVES—

Type.	Class.	Service.	Cylinders. No.	Cylinders. Position.	Cylinders. Size.	Wheels. Bogie.	Wheels. Coupled.	Wheels. Trailing.	Wheel Base. Coupled.	Wheel Base. Total.
WESTERN REGION.										
					″ ″	′ ″	″	′ ″	′ ″	′ ″
4-6-0	King	Hvy. pass.	4	In. & out.	16¼ × 28	3 0	6 6	—	16 3	29 5
4-6-0	Castle	”	4	In. & out.	16 × 26	3 2	6 8½	—	14 9	27 3
4-6-0	Star	”	4	In. & out.	15 × 26	3 2	6 8½	—	14 9	27 3
4-6-0	Hall	”	2	Out.	18½ × 30	3 0	6 0	—	14 9	27 3
4-6-0	County	”	2	Out.	18½ × 30	3 0	6 3	—	14 9	27 3
2-8-0	28XX	Hvy. goods	2	Out.	18½ × 30	3 2	4 7½	—	16 10	25 7
2-8-0	4,700	Mxd. traf.	2	Out.	19 × 30	3 2	5 8	—	20 0	29 3
4-6-0	68XX	Mxd. traf.	2	Out.	18½ × 30	3 0	5 8	—	14 9	27 1
SOUTHERN REGION.										
4-6-0	'Nelson'	Hvy. pass.	4	In. & out.	16½ × 26	3 1	6 7	—	15 0	29 6
4-6-0	'K. Arthur'	”	2	Out.	20½ × 28	3 7	6 7	—	14 6	27 6
4-6-0	S^{15}	Mxd. traf.	2	Out.	20½ × 28	3 7	5 7	—	13 9	26 7½
4-4-0	Schools	Express	3	In. & out.	16½ × 26	3 1	6 7	—	10 0	25 6
0-6-0	Q	Goods	2	In.	19 × 26	—	5 1	—	16 6	16 6
0-6-0	Q^1	Goods	2	In.	19 × 26	—	5 1	—	16 6	16 6
2-6-0	N	Mxd. traf.	2	Out.	19 × 28	3 1	5 6	—	15 6	24 4
2-6-0	U	Pass	2	Out	19 × 28	3 1	6 0	—	15 0	23 10
4-4-0	L^1	”	2	In.	19½ × 26	3 7	6 8	—	10 0	24 3½
4-6-2	Merchant Navy	Hvy. pass.	3	In. & out.	18 × 24	3 1	6 2	3 7	15 0	36 9
4-6-2	West Country	”	3	In. & out.	16⅜ × 24	3 1	6 2	3 1	14 9	35 6

TANK LOCOMOTIVES—

Type.	Class.	Service.	Cylinders. No.	Cylinders. Position.	Cylinders. Size.	Wheels. Bogie.	Wheels. Coupled.	Wheels. Trailing.	Wheel Base. Coupled.	Wheel Base. Total.
LONDON MIDLAND REGION.										
					″ ″	″	″	′ ″	″	′ ″
2-6-0 0-6-2	Garratt	Mineral	4	Out.	18½ × 26	3 3½	5 3	—	16 6	79 0
2-6-4	4P	Pass.	2	Out.	19⅝ × 26	3 3½	5 9	3 3½	16 6	38 6
2-6-4	4P	Pass.	3	In. & out.	16 × 26	3 3½	5 9	3 3½	16 6	38 6
2-6-2	3P	Pass.	2	Out.	17½ × 26	3 3½	5 3	3 3½	16 6	33 3
EASTERN & NORTH EASTERN REGIONS.										
2-8-0 0-8-2	2,395 (Garratt)	Banking	6	In. & out.	18½ × 26	2 8	4 8	2 8	17 10½	79 1
2-6-2	V^3	Pass.	3	In. & out.	16 × 26	3 2	5 8	3 8	16 3	32 3
2-6-4	L^1	Mxd. traf.	2	Out.	20 × 26	3 2	5 2	3 2	13 6	34 6

BRITISH RAILWAYS—(*continued*).

Working Pressure.	Grate Area.	Heating Surface. Tubes.	Heating Surface. Fire-box.	Heating Surface. Super-heater.	Weight of Engine. On Coupled Wheels.		Weight of Engine. Total.		Tender. Coal.	Tender. Water.	Engine and Tender. Total Weight.		Engine and Tender. Total Wheel Base.		Tractive Effort at 85 per cent. of Boiler Pressure.
lb. sq. in	sq. ft.	sq. ft.	sq. ft.	sq ft.	T.	C.	T.	C.	Tons.	Galls.	T.	C.	′	″	lb.
250	34·3	2,007	193	313	67	10	89	0	6	4,000	135	14	57	5½	40,300
225	29·4	1,854	163	263	58	17	79	17	6	4,000	126	11	54	6½	31,625
225	27·0	1,686	155	263	55	8	75	12	6	4,000	122	6	53	6¼	27,800
225	27·0	1,583	155	315	57	10	75	16	6	4,000	122	10	53	6¼	27,275
280	28·8	1,545	169	265	59	2	76	17	7	4,000	125	17	53	6½	32,580
225	27·0	1,686	155	263	67	0	76	5	6	3,500	116	5	53	7¼	35,380
225	30·3	2,062	170	288	73	8	82	0	6	4,000	128	14	56	10½	30,460
225	27·0	1,686	155	263	55	4	74	0	6	3,500	114	0	53	4¼	28,875
220	33·0	1,795	194	376	61	19	83	10	5	5,000	141	9	60	9	33,490
200	30·0	1,716	162	337	60	0	80	19	5	5,000	138	10	58	0	25,320
200	28·0	1,716	162	337	59	5	79	5	5	5,000	135	13	57	1½	29,860
220	28·3	1,604	162	283	42	0	67	2	5	4,000	109	10	48	7¼	25,130
200	21·9	1,125	122	185	49	10	49	10	5	3,500	90	0	38	11¾	26,157
230	27·0	1,302	170	218	51	5	51	5	5	3,700	89	5	40	7½	30,000
200	25·0	1,391	135	285	52	4	61	4	5	4,000	103	12	47	9¼	26,000
200	25·0	1,391	135	285	53	10	62	6	5	4,000	104	14	47	9¼	23,900
180	22·5	1,252	155	235	37	12	57	16	5	3,500	98	6	46	2¾	18,907
280	48·5	2,176	275	822	63	0	94	15	5	5,000	144	2	59	6	37,500
280	38·2	1,869	253	545	56	5	86	0	5	4,500	128	12	57	6	31,000

BRITISH RAILWAYS.

Working Pressure.	Grate Area.	Heating Surface. Tubes.	Heating Surface. Fire-box.	Heating Surface. Super-heater.	Weight of Engine. On Coupled Wheels.		Weight of Engine. Total.		Tanks. Coal.	Tanks. Water.	Tractive Effort at 85 per cent. of Boiler Pressure.
lb. sq. in.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	T.	C.	T.	C.	Tons.	Galls.	lb.
190	44·5	1,954	183	466	122	2	155	10	9	4,500	45,620
200	26·7	1,223	143	230	51	14	87	17	3½	2,000	24,670
200	26·7	1,126	139	198	57	0	92	5	3½	2,000	24,600
200	19·2	9 7	111	138	47	7	72	10	3	1,500	21,486
180	56·4	2,757	237		143	1	178	0	7	5,000	72,940
200	22·0	1,198	127	284	59	2	86	16	4½	2,000	24,960
225	24·7	1,198	138	284	58	19	89	9	4½	2,630	32,080

TANK LOCOMOTIVES—

Type.	Class.	Service.	Cylinders. No.	Cylinders. Position.	Cylinders. Size.	Wheels. Bogie.	Wheels. Coupled.	Wheels. Trailing.	Wheel Base. Coupled.	Wheel Base. Total.
					" "	' "	' "	' "	' "	' "
WESTERN REGION.										
2-8-0	5205	Mineral	2	Out.	19 × 30	3 2	4 7½	—	20 0	28 9
2-6-2	61XX	Mxd. traf.	2	Out.	18 × 30	3 2	5 8	3 8	14 9	31 9
0-6-2	5600	Mineral	2	In.	18 × 26	—	4 7½	3 8	15 3	21 9
2-8-2	72XX	Goods	2	Out.	19 × 30	3 2	4 7½	3 8	20 0	35 3
0-6-0	9400	Shunting	2	In.	17½ × 24	—	4 7½	—	15 6	15 6
SOUTHERN REGION.										
4-6-2	H^{16}*	Goods	2	Out.	21 × 28	3 7	5 7	3 7	15 0	36 6
4-8-0	G^{16}*	Shunting	2	Out.	22 × 28	3 7	5 1	—	18 0	32 0
4-6-2	J	Pass.	2	Out.	21 × 26	3 6	6 7½	4 0	14 6	35 3
0-8-0	Z	Shunting	3	In. & out.	16 × 28	—	4 8	—	17 6	17 6
2-6-4	W	Goods	3	In. & out.	16½ × 28	3 1	5 6	3 1	15 0	36 4

AMERICAN TENDER

Railroad.	Type.	Class.	Service.	Cylinders. No.	Cylinders. Position.	Cylinders. Size.	Wheels. Bogie.	Wheels. Coupled.	Wheels. Trailing.	Wheel Base. Coupled.	Wheel Base. Total.
						" "	' "	' "	' "	' "	' "
Pennsylvania	2-10-0	115	Freight	2	Out.	30½ × 32	2 9	5 2	—		
"	2-8-2	—	"	2	Out.	27 × 30	2 9	5 2	2 9		
"	4-6-2	—	Pass.	2	Out.	27 × 28	3 0	6 8	3 0		
New York Central	4-6-4	5,200	Pass.	2	Out.	25 × 28	3 0	6 7	3 0 & 4 3	14 0	40 4
Southn. Pacific	2-10-2	—	Freight	2	Out.	29½ × 32	2 9	5 3½	4 3½	22 10	42 4
Virginia	2-10-0 / 0-10-2	800	Freight	4	Out.	Comp'd. 30 × 32 / 48 × 32		4 8	—		
Northern Pacific	4-8-4	2,600	Pass.	2	Out.	28 × 30	3 0	6 3	3 0 & 3 9¾	20 3	47 2
Pennsylvania	4-4-4-4	T^1	Express	4	Out.	19¾ × 26	3 0	6 8	3 6	6 11	51 11
C. M. St. P. & P.	4-4-2	Hiawatha	Str'ml'd ex	2	Out.	19 × 28	—	7 0	—	8 6	37 7
Baltimore & Ohio	4-6-4	Lord Baltimore	Express	2	Out.	19 × 28	3 0	7 0	3 0	14 10	42 10½
Baltimore & Ohio	4-4-4	Lady Baltimore	Express	2	Out.	17½ × 28	3 0	7 0	3 0	7 5	35 5½
Philadelphia & Reading	4-8-4	T^1	Fast Freight	2	Out.	27 × 32	3 0	5 10	3 8	19 3	45 10
Richmond, Fredericksburg & Potomac	4-8-4	Statesman	Pass.	2	Out.	27 × 30	3 0	6 5	3 6	20 0	46 1

* Maximum cut-off limited to 50 per cent.

BRITISH RAILWAYS—(*continued*).

Working Pressure.	Grate Area.	Heating Surface. Tubes.	Fire-box.	Super-heater.	Weight of Engine. On Coupled Wheels.	Total.	Tanks. Coal.	Water.	Tractive Effort at 85 per cent. of Boiler Pressure.
lb. sq. in.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	T. C.	Tons.	Tons.	Galls.	lb.
200	20·5	1,350	129	192	72 10	82 2	4	1,800	33,170
225	20·3	1,145	122	82	52 13	78 9	4	2,000	27,340
200	20·3	1,145	122	82	55 12	68 12	3¾	1,900	25,800
200	20·6	1,349	129	192	72 15	92 12	6	2,500	33,170
200		1,069	102	74	55 7	55 7	3½	1,300	22,515
180	27·0	1,267	139	252	59 0	96 8	3½	2,000	28,200
180	27·0	1,267	139	252	72 18	95 2	3½	2,000	34,000
170	25·0	1,462	124	357	54 10	89 0	3	2,000	20,800
180	18·6	1,173	106	—	71 12	71 12	3	1,500	29,376
200	25·0	1,391	135	285	57 5	90 14	3½	2,000	29,452

LOCOMOTIVES.

Working Pressure.	Grate Area.	Heating Surface. Tubes.	Fire-box.	Super-heater.	Weight of Engine. On Coupled Wheels.	Total.	Tender. Coal.	Water.	Engine and Tender. Total Weight.	Total Wheel Base.	Tractive Effort at 85 per cent. of Boiler Pressure.
lb. sq. in.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	T. C.	T. C.	Tons.	U.S.A. Galls.	T. C.	′ ″	lb.
250	70·0	{ 2,731 / 1,313	290	1,418	152 14	166 0	17½	9,000	247 4		90,000*
205	70·0				117 0	151 10					57,850
205	70·0				89 7	140 0					41,845
225	81·5	4,203	288	1,965	81 5	153 2	18	10,000	246 9	76 1½	42,000
200	82·5	4,722	399	1,208	132 14	172 6	Oil Gals. 4,000	12,000	271 8	82 7½	75,150
215	108·2	{ 5,592 / 2,511 }	532	2,120	275 10	305 8	Tons. 12	13,000	401 2		{ 176,600 / 147,200
210	115	4,115	485	1,992	116 1	190 4	24	15,000	332 2	90 0	75,500
300	92·0	3,719	499	1,680	119 14	221 19	36½	19,500	415 5		65,000
300	69	2,951	294	1,029	62 10	125	Oil. Galls. 4,000	13,000	235 10	78 10½	30,700
350	61·75	2,727	612	720	69 13	131 10	T. C. 14 6	10,000	220 9	81 6	{ 34,000 / 41,000 with Booster
350	61·75	1,257	523	351	44 10	95 10	12 10	8,000	165 0	71 4½	{ 28,000 / 35,000 with Booster
240	94·5	4,455	465	1,214	124 4	197 0	23	19,000	361 0	96 6½	{ 68,000 / 79,100 with Booster
260	86·5	3,765	500	1,325	119 0	185 0	20	20,000		96 11	62,800

* T.E. taken at 75 per cent. of boiler pressure.

DEVELOPMENTS ABROAD.

Immediately before the war the Germans put a 4–8–4 type express locomotive into service. Two turbine-driven locomotives on hand were destroyed by bombing. During the war years locomotives were of the 'utility' pattern, of the plainest and simplest construction. In Germany a large number of Class 52, 2–10–0 type locomotives were built in which hand and machine work was cut down to a minimum. Flame-cut plates, brake work, etc., were left unmachined. Coupling and connecting rods were made from drop-forged ends welded to a length of rolled bar. Later engines of the class had boilers with cylindrical corrugated fireboxes. Some of these locomotives were built as condensing engines.

Prior to the war the most striking developments in locomotives were, perhaps, those made on the French railways. A programme of modernisation of existing engines was undertaken and special attention given to the design of steam pipes and passages, valves and valve-gear, exhaust passages, in order to minimise the drop in pressure between boiler and cylinders and to reduce back pressure. Boiler pressure was increased to 290 lb. per sq. in. and a thermic syphon added to the firebox, while a new form of superheater, feed-water heating and improved smokebox arrangement increased the capacity. The horse-power output of the engines was raised by 50 to 100 per cent., and remarkable results in the way of hauling very heavy passenger trains at high speed were realised.

Since the war the French railways have introduced a number of new types experimentally, of both high-pressure and compound working. They include a 2–12–0, six-cylinder compound heavy freight engine, a 4–6–4 type, one of which is three-cylinder simple and the other four-cylinder compound; and a 4–8–4 type three-cylinder compound.

Elsewhere on the Continent the three-cylinder simple engine continues in favour. In Czecho-Slovakia express engines of the 4–8–2 three-cylinder type have been built with all-welded boilers, bar frames and roller bearings to all axleboxes. A three-cylinder 4–8–0 type in Sweden also has bar frames and roller bearings to all axles. In Holland a 4–6–0 three-cylinder passenger locomotive and a three-cylinder 0–8–0 freight engine have roller bearings throughout. A 4–8–2 type has been supplied to one of the Spanish railways.

Large numbers of American-built 2–8–2 and 2–8–0 'Liberation' locomotives of the two-cylinder type have been brought over for the rehabilitation of Continental railways.

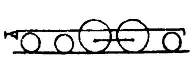
In Australia, the use of three-cylinder locomotives is being extended.

American locomotives are of larger size than those in use elsewhere, and fireboxes have grown to such an extent that a four-wheeled truck is required at the trailing end, hence such types as the 4–6–4, 4–8–4 and 2–10–4. The speed of freight services is increasing and freight engines are built with larger diameter wheels in consequence. Two-cylinder engines are preferred and the cylinders can be of large size on account of the wide loading gauge. Boiler pressures tend to increase. New construction has, however, been considerably slowed down by the increasing use of diesel-electric traction.

As a result of the competition of lightweight high-speed trains propelled by oil engines several steam locomotives have been designed in Germany, France, Belgium, and the United States for operating lightweight trains. These locomotives are partly or fully stream-lined and are capable of speeds in excess of 100 miles per hour, but otherwise follow conventional design in their construction.

Locomotive Types.

The classification of locomotives is by wheel arrangement. Abbreviations are used, the Anglo-American system denoting in their order leading carrying wheels, coupled wheels, and trailing carrying wheels by number of wheels to each group. Where there are no carrying wheels at either end a cipher is added. The letter T denotes a tank engine. Articulated engines are denoted by the wheel arrangement of their separate parts. The Continental system merely states number of axles coupled and total axles of engine, and gives the proportion of adhesive weight without specifying the actual arrangement of wheels. In addition to identity by wheel grouping, the various types receive code names in America. Example:

Diagram.	Code Name.	Anglo-American Notation.	Continental Notation.
	**Atlantic**	**4–4–2**	**2–5**
	**Mogul**	**2–6–0**	**3–4**

In France it is usual to classify a locomotive by groups of axles in their order, instead of by wheels: thus a 4–4–2 engine is known there as type 2–2–1 ; and a 2–6–0 engine is expressed as type 1–3–0.

Another system coming into general use is similar to the French, but groups of coupled axles are represented by letters—A for one driving axle, B for two, C for three, etc. : so that a 4–4–2 engine becomes type 2–B–1. A 2–6–0 engine is denoted by 1–C only, a cipher not being necessary to indicate absence of carrying wheels with this system. Uncoupled driving axles are denoted by the suffix ' O,' thus 2–D$_o$–1 represents a locomotive with a four-wheeled bogie, four uncoupled driving axles and a pair of trailing wheels. This system is used in the case of electric or Diesel-electric locomotives.

Types of Locomotives most generally used on British Railways.

Tender Engines.

Type.	Principal Use.	Type.	Principal Use.
0–6–0	Ordinary goods traffic	4–4–0	Fast passenger trains
0–8–0	Mineral traffic	4–4–2	Main line express
2–8–0	Heavy goods traffic	4–6–0	Heavy express and fast freight
2–6–0	Mixed traffic	4–6–2	Exceptionally heavy express

Tank Engines.

Type.	Principal Use.	Type.	Principal Use.
0–4–0	Private yard shunting	2–6–2	
0–6–0	Ordinary shunting	4–6–0	Main line goods trains
		0–6–2	
		0–4–2	Light and Suburban passenger trains
0–8–0		0–4–4	
0–8–2		0–6–4	
0–8–4	Heavy shunting, banking and mineral trains	2–6–4	
2–8–4		4–4–2	
4–8–0		4–6–2	Main line passenger trains
2–8–0		4–6–4	

The particular uses of the various types of tank engines are not so well defined as with tender engines. Several of the engines specified for goods traffic are often used on passenger trains, and *vice-versa*. Generally speaking, the total adhesion type (i.e. engines without carrying wheels) are mostly used for shunting and local work. Carrying wheels are added as the water and coal capacity are enlarged to permit of greater radius of action. The tank engines with two or more pairs of carrying wheels are capable of travelling considerable distances without replenishing fuel and water.

Articulated Locomotives.

On railways having a limited axle loading and tracks abounding in sharp curves and often of narrow gauge, a powerful locomotive can be provided by a division of the driving wheels into two independent groups, one or both being separated from the main framing of the engine and attached only by flexible connections. In the Fairlie and Kitson-Meyer designs the boiler, tanks, and coal-bunker are carried on a main frame supported by two steam-driven bogies, the centres of articulation being near the centre of the wheel base of each bogie. The ' Mallet ' type, first used in France, has been developed to an enormous size on American railways as a tender engine. High-pressure cylinders drive the axles on the main frame which carries the boiler, while a steam bogie driven by high- or low-pressure cylinders supports the smoke-box end of the boiler. The ' Garratt ' type is now extensively used on British colonial railways, in South America, and elsewhere. The boiler alone is carried on the main frame, the water-tanks and coal-bunker being supported by two steam-driven bogies. By this arrangement the centres of articulation are brought nearer together, the pivots being towards the inner ends of the bogies. Very powerful locomotives of this type have been constructed for the 3 ft. 6 ins. gauge in South Africa, Western Australia, and New Zealand, both for passenger and freight services.

The 4–6–2 + 2–6–4 Garratt type has recently been used for high speed passenger trains in Algeria and has been tested in Northern France for speed and haulage on heavy expresses ; the driving wheels are 5 ft. 11 ins. diameter and speeds up to 72 miles per hour have been run with safety.

The leading particulars of a recent 'Garratt' engine for the South African Railways are as follows:—Type 4–8–2 + 2–8–4, class GL, freight service, cylinders 4, size 22 ins. by 26 ins., bogie wheels 2 ft. 4½ ins. diameter, coupled wheels 4 ft. 0 ins., trailing wheels 2 ft. 9 ins. Coupled wheel base 13 ft. 3 ins., total wheel base of each unit 27 ft. 8 ins., total wheel base 83 ft. 7 ins. Centres of articulation 41 ft. 6 ins., working pressure 200 lbs. per sq. in. Grate area 74·5 sq. ft. Heating surface of tubes 3,036 sq. ft., of firebox 340 sq. ft., superheater surface 809 sq. ft. Adhesive weight 147 tons 8 cwt. Total weight 214 tons 2 cwt. Coal 12 tons, water in tanks 7,000 galls. Tractive effort at 75 per cent. of boiler pressure 78,650 lbs. The track on which these engines run is of 3 ft. 6 ins. gauge laid with 80-lb. rails, for which the maximum axle load is 18½ tons. The maximum gradient is 1 in 30, and the curves 300 ft. radius.

FIRELESS LOCOMOTIVES.

The fireless steam locomotive is adopted in industrial areas where fire prevention is essential. It is capable of performing shunting work and operating short distance traffic. Simplicity and low cost of maintenance are factors in its favour. A good deal of development has taken place in Austria in recent years. At one works 400-ton trains are drawn a distance of 4½ miles over gradients of 1 in 50 at a speed of 20 m.p.h., involving a cylinder horse power of 1,000. The storage pressure of 1,200 lbs. per sq. in. is reduced to 215 lbs. for the cylinders. Recharging from power station boilers takes 15 mins.

In Italy it is proposed to charge the steam accumulators electrically and so obviate need for stationary plant.

General Design of Locomotives.

The principal limitations governing locomotive design are:—

(1) *Loading gauge*, giving extreme dimensions as to heights and widths.

(2) *Rail gauge*, affecting stability, etc.

(3) *The track*, which limits the maximum axle load.

(4) *Bridges*, the strength of which restricts the weight per ft. run over total wheel base.

(5) *Curves*, which limit the length of engine and determine rigid wheel base.

Other factors to be taken into consideration are:—

(1) *Loads*, as to weight and type of rolling stock.

(2) *Gradients*, both ruling and maximum.

(3) *Resistance of curves.*

(4) *Speed.*

(5) *Prevailing atmospheric conditions.*

These are inter-related, load becoming a maximum for both directions of travel when there are no gradients or curves and the speed is low. An increase in speed or the presence of gradients, curves, and side winds reduces loads.

Some further points to be looked into when designing new engines are:—

(1) *Length and strength of turn-tables* and clearances round them.

(2) *Engine sheds*, as to clearances, length of pits, etc.

(3) *Coal stages* height of platform, method of coaling, etc.

(4) *Position of water columns* at stations and in yards in relation to signals and points.

(5) *Distance between available coal and water supplies.*

(6) *Capacity of workshops*, in regard to load on cranes, length of pits and traversers, necessity for special tools, etc.

The bulk of the work on British railways is done by tender engines of the 0–6–0 type for goods and 4–4–0 type for passenger trains. Many of these engines are of fairly recent construction. Some of the older engines have been brought up to date by being rebuilt with larger boilers fitted with superheaters and with new cylinders provided with piston valves. The goods engines usually have cylinders 19 ins. × 26 ins. and wheels 5 ft. 0 ins. diameter; the passenger engines 20 ins. × 26 ins. cylinders and 6 ft. 6 ins. wheels. Boiler pressure is about 180 lbs. per sq. in.

For heavy passenger work the 4–6–0 type is resorted to. The 4–6–2 type hauls exceptionally heavy trains at high speeds. For mineral traffic and long goods trains an eight-wheel coupled engine is used. The 2–6–0 type with wheels about 5 ft. 6 ins. is extensively used as a mixed traffic engine, and a similar engine of the 2–8–0 type has been introduced for heavier trains of this description. They are especially suitable for fast goods, excursion trains, etc. In the large passenger and goods engines outside cylinders predominate. The advantages claimed for the outside position are shorter wheel base, long connecting rods, no crank axle, direct drive through rods on all wheels, horizontal cylinders. The distance between centres of cylinders is about 6 ft. 8 ins. and cannot be reduced very appreciably on account of the necessity for finding sufficient bearing surface for coupling and connecting rods. Thus, the diameter of cylinders is limited by the remaining space within the load gauge, which varies on the different railways. Since the load gauge is at its narrowest near rail level, advantage can sometimes be gained by inclining outside cylinders so that they are brought up into a wider part of the gauge.

The demand for larger cylinder capacity can be met by increased diameter or stroke of cylinders, or both. Another way is to multiply the number of cylinders, hence three- and four-cylinder high-pressure engines. Multi-cylinders are also being adopted for mechanical reasons, as they give better balancing and turning movement, less disturbance in running, subdivision of driving forces, increased bearing areas and other advantages. Multi-cylinder engines are necessarily more costly and complicated than two-cylinder. It is therefore desirable to reduce their working parts to a minimum. This can be done in the case of four-cylinder engines by providing two sets of valve gear only and operating the additional valves through rocking levers attached to the valve gears.

Three-cylinder engines can be arranged in a similar manner by adopting a conjugated gear for the inside cylinder. In this case the valves of the outside cylinders are directly operated by their respective valve gears which also indirectly operate the inside valve through a simple system of levers attached to them. Some designers prefer an independent valve gear for the middle valve.

Boiler pressures tend to increase; 200 to 250 lbs. per sq. in. is now common, and 280 lbs. has been reached.

'Poppet' or double-beat lift valves, which have been used to a limited extent on the Continent for some years, have undergone trials under British railway conditions and a small number of engines have been so equipped. Indian and Colonial railways are adopting them more extensively.

Current American practice includes bedplates of cast steel with integral cylinders, roller bearing axleboxes, boosters, fireboxes with arch tubes and syphons, and feed water heaters.

Some recent American locomotives are of enormous size. For the heaviest freight traffic Mallet articulated compounds are in use on the Virginian R.R. weighing, with tender, nearly 400 tons, a pair of cylinders being added to the tender to make use of its adhesive weight, the total tractive effort being about 80 tons. The latest engines on the New York Central and other lines have a trailing bogie below the firebox, being of the 4–6–4 and 4–8–4 types. The axle loads on the Pennsylvania R.R. reach 30 tons. In order to reduce dynamic loading of wheels the reciprocating parts are of alloy steel. With small wheels the revolving masses must likewise be reduced, since it is not always possible to make the balance weights in the wheel large enough. Outside valve gears such as the Walschaert, Baker, or Southern are general, and piston valves are of large diameter with inside admission. Tank engines are not common, even switching engines having tenders.

New locomotives for passenger service are of the 4–6–4 and 4–8–4 types. For freight traffic the 2–8–2 or 2–8–4 type is mostly favoured, but the 2–10–4 type and the Mallet locomotives are also being built. Construction has been slowed down by extensive adoption of diesel traction, but a good deal of reconstruction and modernising of existing locomotives is taking place.

For operating over long distances the Union Pacific R.R. has adopted tenders having a capacity of 23,500 gallons of water and 25 tons of coal. They are carried on fourteen wheels, the first four of which are arranged as a bogie and the remainder are rigid.

The Canadian Pacific R.R. has constructed a 2–10–4 type having a tractive effort of 76,900 lbs. The twelve-wheeled tender carries 12,000 gallons of water and 4,100 gallons of oil.

The Union Pacific R.R. are trying a steam-electric locomotive rated at 5,000 h.p. consisting of two units of 2,500 h.p. coupled together, each equipped with oil-fired boiler, H.P. and L.P. turbines and condenser. Generators are driven through helical reduction gears and these supply current to traction motors suspended on the driving axles.

The Chesapeake & Ohio R.R. have three 6,000 h.p. coal-burning steam-turbo-electric locomotives. Steam at 290 lbs. per sq. in. is supplied to an atmospheric exhaust steam turbine driving two generators which supply current to eight traction motors.

Another new type is a 6–4–4–6 express passenger engine with four outside cylinders, 22 in. × 26 in., driving two independent groups of coupled wheels of 7 ft. in diameter. One pair of cylinders is located immediately to the rear of the leading six-wheeled bogie and the other pair in rear of the first set of coupled wheels. This engine will develop 6,500 h.p. at a speed of 100 m.p.h. It has a sixteen-wheeled tender carrying 23½ tons of coal and 20,200 gallons of water.

A similar engine of the 4–4–4–4 type has been put to work on the Pennsylvania R.R. for passenger service and one of the 4–4–6–4 type for freight.

In the largest American locomotives a bogie is required under the firebox, in addition to a pony truck or bogie at the leading end, so that a large percentage of the weight of the engine is not available for adhesion. The addition of a booster partly overcomes this drawback, and enables the engine to start with a heavier train than it could do in the normal manner.

The 'booster' is a small steam engine geared to the carrying wheels, and which can be thrown in or out of action as required. It is used to assist the main engine when starting a heavy train from rest or in climbing grades. The device has also been applied to the tender wheels.

The Pennsylvania R.R. have in service a turbine-driven locomotive of 6,500 h.p. It exhausts to atmosphere through a blast pipe in the smoke-box, as in the case of the L.M.S.R. engine in England. It is of the 6–8–6 type and has a conventional boiler with pressure of 310 lbs. per sq. in.

Another feature peculiar to American practice is the 'limited cut-off' locomotive, in which the maximum cut-off is fixed at about 60 per cent. in full gear. Under the operating conditions prevailing engines are at times worked 'all-out' at the full gear cut-off of 80 to 90 per cent. of the stroke for long periods, which is most wasteful of steam. By enlarging the cylinder diameter and reducing the cut-off, and consequently the mean effective pressure, an equally powerful but more economical engine can be constructed. Positive starting is ensured by means of notched ports in the valve liners which allow a small supply of steam to fill the cylinders should the engine be on a 'dead centre.'

The Norfolk & Western R.R. has some 4-8-0 type shunting engines, or 'switchers,' in which exhaust steam is discharged to atmosphere through a silencer. A mechanical stoker meters coal and controls an induced draft fan in the smokebox so that fuel and air are automatically supplied as required to maintain steam supply.

A locomotive fitted **with a 'Velox' boiler has been constructed for one of the French railways.**

Consideration is being given on the Continent and in the U.S.A. to the design of steam locomotives with individual axle drives, each axle being driven by one or more small enclosed high-speed engines. In France a $2\text{-}C_0\text{-}2$ type has two reciprocating engines per axle connected through reduction gears. In Germany a $1\text{-}D_0\text{-}1$ type was completed during the war. Each axle is driven by one enclosed steam engine. Another French $2\text{-}C_0\text{-}2$ type has individual turbines with double reduction gear to each axle, totalling 2,000 h.p.

TRAIN RESISTANCE.

There are numerous train resistance formulæ, and they differ considerably. It is only safe to apply them to similar rolling stock to that from which the experimental results were obtained, as conditions vary. For instance, American formulæ are unsuitable for British or Continental rolling stock, and *vice versa*. In estimating the resistance of a train, it should be observed that the results can only be taken as approximate, due to various small factors which may influence them, such as condition of track, play between flange and rail; also empty rolling stock has a relatively higher resistance than when fully or partially loaded, etc.

The *starting* resistance of a train is very variable, and usually ranges from 12 to 18 lbs. per ton where plain bearings are used. It is dependent on how long the vehicles have been standing and on the temperature of the atmosphere, this variation being due to a deficiency in the lubrication and to the viscosity of the lubricant. In warm weather oil boxes offer greater resistance than grease boxes, as the oil is more quickly squeezed out of the bearing while standing, while the reverse occurs in cold weather when the grease is solidified and so causes a high starting resistance. Once movement of the vehicle takes place, efficient lubrication is restarted and the resistance falls to about 5 lbs. per ton at slow speed.

Trains fitted with roller bearings have a lower starting resistance than those fitted with plain bearings lubricated by oil, but when running speed has been attained there is very little difference between them.

Resistance of Air to Train.

(1) The area of resistance presented by the first carriage is its front superficies; that of the following carriages, about 10 square feet, but depending upon the distance between them.

Q = resistance of air, in lbs. A = surface of resistance, in sq. ft. = about 70 ft. + 10(*n* − 1) for railway waggons, where *n* = number of waggons in train. (54 ft. according to M. Vuillemin, &c., 50 ft. Unwin.) *v* = velocity, in feet per second. *e* = 1·43 for a thin surface = 1·17 for a cube, = 1·07 for a train of 5 waggons, = 1·04 for a train of 25 waggons.

$$Q = \cdot 0011896 eAv^2 \ (\textit{Pambour}).$$

(2) If P = pressure on the front ends in pounds per square foot; V = velocity in miles per hour:—

$$P = \cdot 00265 \ V^2.$$

Assuming the superficial area of the front end to be about 75 sq. ft., the following table shows the total pressure and the power required to overcome that pressure at various speeds:—

Speed in miles per hour.	Total pressure, lbs.	H.P.
20	79·5	4·24
40	318·0	33·92
60	715·5	115·28
80	1272·0	271·36
100	1987·5	530·0

It will be seen that at speeds below 40 miles an hour the effect of head end resistance is practically negligible; but at speeds of 60 miles an hour or more the power required to overcome head end resistance is by far the most important element in train resistance. As the number of cars in a train is increased, the relative effect of head end resistance is decreased.

Experiments made on models of trains in a wind tunnel at the National Physical Laboratory indicate that about 400 horse-power is expended on overcoming air resistance at a speed of 60 miles per hour with a 10-coach train made up as follows:—

	Per cent. of total.	Horse-power.
Engine and tender	29	116
First coach	$8\frac{1}{2}$	34
Second to ninth ($6\frac{1}{2}$ each)	52	208
Tenth coach	9	38

For reduction of Air Resistance see 'Wind Tunnel Experiments in Canadian National Laboratory.' *The Engineer*, April 14th, 1933.

An engineer of the firm which furnishes electric locomotives for the Simplon Tunnel says that at a speed of 50 miles an hour in the tunnel the air-resistance absorbs 400 horse-power, or 40 per cent. of the capacity of the locomotive, against 95 horse-power in the open air.

In a paper read before the Institution of Mechanical Engineers in 1936 by Mr. F. O. Johansen, M.Sc. (Eng.), the results were given of tests in a wind tunnel of various rakes of trains, standard and modified, of L.M.S.R. and L.N.E.R. designs.

The models were in most cases made to $\frac{1}{40}$th scale and were tested at various wind velocities and various angles of yaw.

The following conclusions were among those reached:—

(1) The air resistance of a train of conventional British type is equivalent to about $0{\cdot}0016\,T^2$ lbs. per ton of train weight, where T is the speed in still air in miles per hour. This represents upwards of half the total train resistance at speeds above 80 m.p.h.

(2) The air resistance can be reduced by 50 per cent. without drastic departure from conventional design, and by 75 per cent. by ideal streamlining.

(3) Air resistance is augmented by side winds, the increase being mainly due to frontal pressure on exposed surfaces. The lateral wind force, perpendicular to the train, is large, but the consequent increase of forward resistance due to flange and bearing friction is relatively small, except for highly streamlined trains.

(4) The worst natural wind is not directly ahead but ranges from 30° to 60° on either side of the ahead direction.

(5) The gaps between the coach bodies of an ordinary train account for relatively little air resistance, and much of it can be obviated by abutting coaches as close together as practicable and covering the remaining gap to the general contour of the train. The resistance is roughly proportional to the width of the gaps and arises far more from frontal pressure than from suction on the trailing end.

(6) A surprisingly large proportion of the air resistance of a coach, especially under the action of oblique winds, is contributed by the bogies and undercarriage structure. The air resistance is less if the undercarriage is totally enclosed than if only side valances are fitted.

(7) A fair shape at the tail end of a train reduces air resistance to an extent which is more marked the more complete the streamlining, but greater advantage can be gained by fairing the front than by fairing the rear end.

(8) The air resistance of a conventional locomotive, amounting to 30–40 per cent. of that of a complete six-coach train, can be reduced 25 per cent. by rounding the smokebox front and covering the tender to the general contour of the train. An important advantage of the covered tender is in reducing the air resistance of the first coach.

(9) The ideal streamlined train is a continuous cylindrical body with well-rounded ends, having a polished surface free from external fittings and irregularities.

In the first resistance formula the weight of a coach has been taken at about 30 tons.

The resistance of the last coach can be reduced only by thorough streamlining.

The gap between coaches is of considerable importance for ideally streamlined trains only.

In comparing air resistance with mechanical resistance, the formulæ used for the latter were:—

$$\text{Resistance of locomotive} = 8{\cdot}8 + 0{\cdot}126T \text{ lbs. per ton}$$
$$\text{Resistance of coach} = 4{\cdot}0 + 0{\cdot}025T \quad ,, \quad ,,$$

where T = velocity in m.p.h.

The tractive resistance due to lateral force arising from wind pressure is 1·45 per cent. of that force.

Lateral air resistance is practically constant for a standard L.M. & S.R. train and for an ideal train. Therefore, in the case of a streamlined train the lateral air resistance is a greater percentage of the total longitudinal and lateral resistance than in the case of an ordinary train.

The ratio $\frac{\text{Streamlined}}{\text{Standard}}$ Total Air Resistance for a locomotive and six coaches at 100 m.p.h. is about ⅓ at 0° and 10° yaw, dropping to about ½ at 30° yaw.

The direction of the wind for the greatest resistance depends on the relative speeds of the train and the wind, and on the angle ϕ between them, and on the form of the rolling stock with respect to streamlining.

As the ideal shape of the train is approached, so the angle ϕ approaches 180°, the nearest for practical conditions being about 160° and 200°.

Resistance to Traction on a Level Railroad.

FORMULÆ FOR TRAIN RESISTANCE.

Authority.	Value of R.	Conditions.	Reference.
Aspinall	$2{\cdot}5+\frac{V^{\frac{5}{3}}}{50{\cdot}8+0{\cdot}0278L}$	General formula, bogie-coaches, oil axle boxes	**R = tractive resistance in lbs. per ton (2240 lbs.). V = velocity in miles per hour. w_1 = average weight of each vehicle in tons (2240 lbs.). L = length of train in feet.** w_a = average weight per axle in tons. n = number of axles per vehicle.
Great Indian Peninsular Railway	$3{\cdot}20+\frac{4000-41w_1}{2500w_1}V+\frac{1000+7w_1}{2000w_1}V^2$	Passenger coaches on 5 ft. 6 in. gauge	*The Engineer*, June 21, 1935, p. 640.
	$\frac{1190-13w_1}{250}+\frac{(7w_1)}{10{,}000}V+\frac{(331)}{10{,}000w_1}V^2$	Covered goods wagons, 4-wheeled, 11 ft. 6 in. wheel base, 5 ft. 6 in. gauge.	
	$\frac{53-w_1}{10}+\frac{7w_1}{10{,}000}V+\frac{(369)}{10{,}000w_1}V^2$	Open goods wagons 4 wheeled, 11 ft. 6 in. wheel base, 5 ft. 6 in. gauge	
L.M.S.R. (W. A. Stanier)	$\frac{5{\cdot}5(V+15)}{V+3{\cdot}5}+\frac{V^{1{\cdot}18}}{520}$	**Locomotives**	***Engineering*, September 18, 1936.**
”	$\frac{4(V+12)}{V+4{\cdot}5}+\frac{V^{1{\cdot}87}}{322}$	**Carriages**	” ”
Davis	$\left(1{\cdot}45+\frac{29}{w_a}+0{\cdot}034V+\frac{0{\cdot}041V^2}{w_a n}\right)$	**Carriages.**	U.S.A. Formula, translated for English tons.

From tests carried out on the former L. & N.E.R. with four-wheeled goods wagons of 6 tons tare and 10 tons capacity, and with bogie wagons of 17 tons tare and 50 tons capacity, the following results were obtained :—

Speed—m.p.h.	**10**	**20**	**30**	**40**	**50**
Resistance in lbs. per ton—10-ton wagons	**4½**	**4¾**	**5½**	**8**	**12½**
Resistance in lbs. per ton—50-ton wagons	**3½**	**3½**	**3¾**	**4¾**	**6¾**

All wagons were fitted with oil axle boxes, and were fully loaded.

Resistance of Curves.

Resistance due to curvature of track varies inversely as the radius, and directly as the rail-gauge and length of rigid wheel base. It also depends to some extent on the state of the surfaces of wheels and rails. The practice of watering the rails or lubricating the flanges of the wheels lessens the resistance, whilst the occasional necessity of sanding the rails for the locomotive greatly increases it. On main lines the curve resistance is usually very small, and no allowance is made for it. In special cases, where the curves are sharper, such as in sidings, station yards, or at junctions, it is sometimes necessary to form some estimate of the resistance. The American Master Mechanics' Association gives the resistance per ton (of 2,240 lbs.) as 0·77 lbs. for each degree of curvature of track in the case of cars (with bogies), and double this amount—1·54 lbs. —for locomotives.

In British practice, curvature is expressed by the radius in feet or chains. As 1° of curvature corresponds to a radius of 5,730 feet, the radius of any curve in degrees is given in feet by $\frac{5{,}730}{x^\circ}$ **or in chains by** $\frac{87}{x^\circ}$.

$$\text{Resistance in lbs. per ton} = \frac{4{,}412}{\text{radius in ft.}} \text{ or } \frac{67}{\text{radius in chains}} \text{ for cars.}$$

$$\text{,,} \qquad \text{,,} \qquad = \frac{8{,}824}{\text{radius in ft.}}\,;\text{ or } \frac{134}{\text{radius in chains}} \text{ for locomotives.}$$

To find the equivalent up-gradient, 1 in n, **that offers same resistance as curve :—**

$$n = \text{radius in ft.} \times 0{\cdot}5.$$
$$= \text{radius in chains} \times 33.$$
$$= \frac{2{,}871}{\text{curvature in degrees.}}$$

Resistance due to Gradients on Railways.

The force, F, required to move 1 ton up a slope of 1 in n, **neglecting friction and resistance of air** $= \frac{1 \text{ ton}}{n} = \frac{2{,}240 \text{ lbs.}}{n}$, **or resistance in lbs. due to gravity on any incline** $= \frac{2{,}240}{\text{rate of gradient}}$ **per ton of train.**

On a gradient of 1 in 300 the total resistance becomes about twice that on a level at a speed of 30 m.p.h., and therefore only half the utmost load can be carried, or the speed must be diminished. Gradients above 1 in 330 are considered first class ; 1 in 330 to 1 in 150, fair working gradients.

In American and Continental practice, gradient is given as rise per cent. or per thousand, so that grade of 1 in $n = \frac{100}{n}$ **per cent., or** $\frac{1000}{n}$ **per thousand. Thus 1 in 300 = 0·33 per cent. or 3·3 per mille.**

Rise in feet per mile $= \frac{5280}{n}$ **or 17·6 ft. for 1 in 300.**

Effect of Gradients on Speed of Passenger Trains.

The work done by the locomotive in a given time is proportional to total resistance × speed. For speeds over 25 m.p.h. on the level, the resistance of an average engine and train may be fairly represented in lbs. per ton by the simple formula $R = 5 + \cdot 003v^2$. The work done on the level is therefore proportional to $v\,(5 + \cdot 003v^2)$. On a gradient of 1 in n this becomes

$v\left\{(5 + \cdot 003\,v^2) + \frac{2240}{n}\right\}$. If the engine is capable of maintaining a speed of 60 m.p.h. on the level, and the horse-power output is constant, the effect of the gradients on the speed is as follows:—

Inclination.	Level.	1 in 2,500	1 in 1,000	1 in 750	1 in 500	1 in 250	1 in 150	1 in 120	1 in 100	1 in 90	1 in 80	1 in 70
Speed { Miles per hour	60	59	57	55	53	47	40	36	33	31	28	26

Resistance of Locomotives.

The resistance of a locomotive is a very uncertain quantity, and there is no formula which can take account of all the variables. There are, however, several formulæ which give results approximating to the average resistances in ordinary working.

The resistance may be divided into three components:—

(1) Internal resistance of engine machinery, including driving wheels.
(2) Resistance of locomotive as a vehicle, including head-on air resistance.
(3) Resistance of tender as a vehicle.

According to Professor W. F. Goss, the internal resistance is fairly constant for any given cut-off for a wide range of speed, but that the later the cut-off, and consequently more even the turning moment, the less the friction.

As a vehicle the locomotive not only has the usual rolling resistance, journal friction, etc., but it has to encounter the atmospheric pressure on the surfaces perpendicular to the direction of travel, and it shelters to some extent the train behind.

The reaction of the steam pressure on the cylinder covers, applied alternatively to either side of the engine, causes it to move in a sinuous path and sets up further flange friction.

The tender has a resistance very similar to any other vehicle of the train, but it is modified a little by the movement of the engine to which it is closely coupled.

A method in use in the U.S.A. for estimating locomotive resistance is to calculate the resistance of the carrying and tender wheels by means of the Davis formulae (see page 528), add to this 25 lb. per ton (of 2,000 lb.) of weight on the coupled wheels and $0 \cdot 36V^2$ pounds as frontal air resistance on an area of 15 ft. × 10 ft. Converted to English tons and a frontal area of 13 ft. × 9 ft. total resistance can be expressed as follows:—

$$WR = nw_a\left(1 \cdot 45 + \frac{29}{w_a} + 0 \cdot 031V\right) + 0 \cdot 315V^2 + 28w_d$$

where

w_a = average weight per carrying axle, n = number of carrying axles, V = speed in m.p.h., w_d = total weight on coupled wheels, W = total weight of engine and tender in working order and R = resistance in lbs./ton.

Mr. Lawford H. Fry gives separate formulæ for engines with 4, 6, and 8 coupled wheels. which in lbs. per ton (2,240 lbs.) of engine and tender can be expressed as:—

$$R = 8 \cdot 5 + 0 \cdot 09V + 0 \cdot 004V^2 \text{ for 4 wheels coupled.}$$
$$= 10 \cdot 0 + 0 \cdot 12V + 0 \cdot 004V^2 \text{ for 6 wheels coupled.}$$
$$= 13 \cdot 4 + 0 \cdot 48V + 0 \cdot 004V^2 \text{ for 8 wheels coupled.}$$

The equations are derived from the mean of the results obtained by numerous European experimenters. (*The Engineer.*)

According to L. H. Fry a general expression for the machine friction of a steam locomotive is given by the formula

$$F = \frac{0 \cdot 33nW}{D}.$$

Where n = number of driving axles coupled in a group, W = the weight on all driving wheel in pounds, D = driving-wheel diameter in inches and F = the machine friction in pounds of draw-bar pull.

About 22 lbs. per ton at 50 m.p.h. is the usual resistance of a British 4-4-0 type engine; and at 60 m.p.h. some 4 h.p. per ton is expended in moving the engine and tender.

In order to see how much power the locomotive absorbed as compared with the train, a certain number of experiments were tried on the former Lancashire and Yorkshire Railway, and it was found that the ten-wheeled engine No. 1392 absorbed 34 per cent. of the total horse-power. Mr. W. M. Smith (the Institution of Mechanical Engineers) gave the result of his experiments as about 36 per cent. of the total horse-power; but the actual figure depends on the load behind the engine, the percentage being higher with light loads.

Train Acceleration.

The resistances to be overcome in moving a train are frictional and gravitational. Any draw-bar pull in excess of them is available for accelerating the train; or

$$F = T - W(R_1 + R_2 \pm R_3),$$

where

F = accelerating force in lbs.; T = tractive effort in lbs.; W = weight of train in tons; R_1 R_2 and R_3 = opposition to traction in lbs. per ton due to train resistance, curves, and gradients respectively. A negative result indicates retardation.

For the actual acceleration or retardation produced, allowing 10 per cent. for rotary acceleration of wheels and axles,

$$a = \frac{F \times 32}{1{\cdot}10W \times 2240} = \frac{F}{77W} \text{ ft./sec.}^2$$

In American practice about 5 per cent. is allowed for rotary acceleration, as the car wheels are smaller in diameter and also bear a lesser ratio to the weight of car.

To produce a speed of V miles per hour in distance S feet,

$$F = \frac{82V^2W}{S} \text{ lbs.; or } F = \frac{82(V^2 - v^2)W}{S} \text{ lbs., } v \text{ being initial speed.}$$

To produce a speed of V miles per hour in t seconds,

$$F = \frac{112VW}{t} \text{ lbs.; or } F = \frac{112(V - v)W}{t} \text{ lbs.}$$

Power, Tractive Force, and Adhesion of Locomotives.

The pressure exerted on the piston is to the force exerted in the traction of the train inversely s the velocities of the piston and train, i.e., as twice the stroke of the piston to the circumference of the wheel.

Piston Speed.—Ordinary piston speed is about 900 feet per minute.

If d = diameter of piston, in inches; a = area of piston, in sq. inches; l = length of stroke, in inches; p = indicated pressure per square inch, in lbs.; D = diameter of driving-wheel, in inches; V = velocity, in miles per hour; n = number of revolutions per minute = $336\frac{V}{D}$; then

$$\text{Tractive force at rails} = \frac{d^2\,lp}{D} \text{ lbs.}$$

$$\text{Indicated horse-power} = \frac{d^2plV}{375D} = \frac{4an\,pl}{33{,}000} \times \tfrac{1}{12}.$$

The calculated tractive (or starting) effort

$$T = \frac{cd^2Pl}{D} \text{ lbs.}$$

where

P = boiler pressure, and c is a coefficient varying from 0·8 to 0·9, but usually taken at 0·85 both in British and American practice.

The above relates to the ordinary two-cylinder high-pressure engines. Multiply results by 1·5 for three-cylinder, and by 2 for four-cylinder locomotives. (For compound locomotives, see p. 536.)

Cylinder Horse Power.—According to F. J. Cole, horse-power is at a maximum at a piston speed of 1,000 ft. per minute. Based on this

$$\text{I.H.P.} = 0{\cdot}0212 \times P \times A \text{ for saturated steam;}$$
$$\text{I.H.P.} = 0{\cdot}0229 \times P \times A \text{ for superheated steam}$$

where

P = boiler pressure in lbs. per sq. in. and A = area of one cylinder in sq. ins. (in the case of two-cylinder engines).

The horse-power given out at the draw-bar is less than the indicated horse-power by the amount expended in moving the engine and tender. This is approximately 400 h.p. in modern locomotives running at 60 m.p.h., representing about 35 per cent. of the maximum indicated horse-power. (See Resistance of Locomotives, p. 530.)

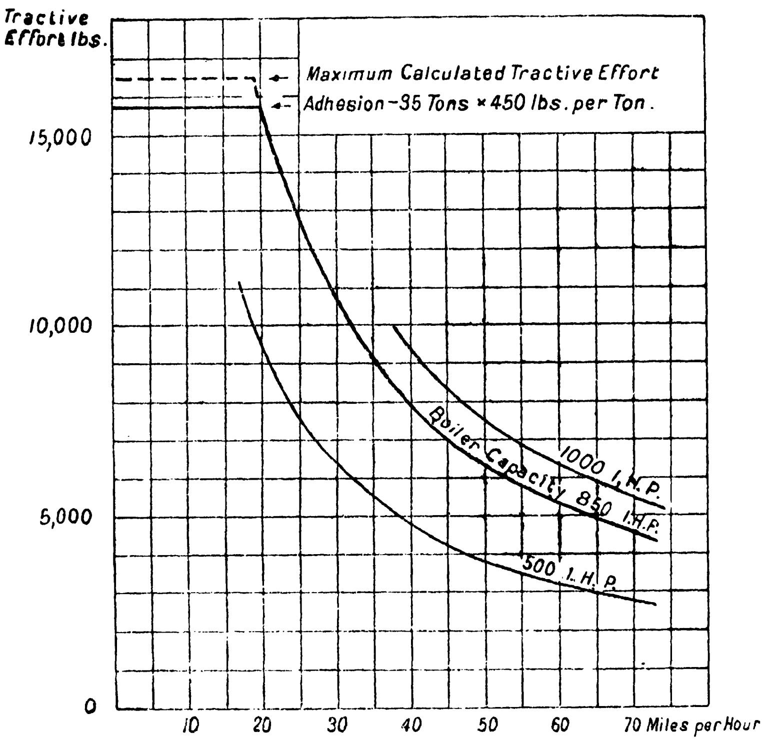


FIG. 1.

Draw-bar horse-power = 5·97 × pull (in tons) × speed (m.p.h.) = 0·00267 × pull (in lbs.) × speed m.p.h.).

ADHESION.

The proportion of the weight on the driving wheels giving *adhesion* and available as tractive force varies considerably, from about 200 lbs. per ton in frosty and greasy weather to 600 lbs. in dry weather. It must, of course, exceed the tractive power, and is generally taken at about 350 lbs. per ton, this being quite feasible, as all locomotives are fitted with a sanding arrangement.

In average British practice, the factor of adhesion, that is, the ratio between adhesive weight and calculated tractive effort, is between 4 and 5 for four and eight-wheel coupled engines, and between 5 and 6 for six wheels coupled. In tank engines the factor varies with the amount of water and coal present. Multi-cylinder engines may have a lower factor on account of their more even turning moment.

The permissible axle load in relation to the maximum tractive effort determines the number of axles to be coupled, thus :—

$$\text{Max. T.E.} = \text{adhesion, } i.e. = \frac{\text{Adhesive weight}}{\text{Factor of Adhesion}}.$$

If n = number of axles to be coupled, W = permissible weight on each axle, and K = factor of adhesion

$$\text{Max. T.E.} = \frac{nW}{K} \text{ or } n = \frac{\text{T.E.} \times K}{W}.$$

The relation between horse-power, tractive effort, and adhesion, is shown by the characteristic curve of a locomotive in fig. 1, where tractive effort is plotted against speed. Horse-power appears on the diagram as a hyperbolic curve, since for any given horse-power pull × speed = a constant. In the example given the maximum calculated tractive effort is 16,500 lbs. The actual pull is limited by adhesion to 15,750 lbs., adhesive weight being 35 tons and the adhesion 450 lbs. per ton. The boiler capacity is taken as 850 i.h.p. and is plotted as a hyperbolic curve. The intersection of the curve with the maximum available tractive effort gives the lowest speed at which the boiler is fully used. Until this point is reached horse-power is limited by cylinder capacity or adhesion, as the case may be. Beyond it, the horse-power is approximately constant and the pull falls off as the speed rises. A deduction must be made for engine and tender resistance to obtain actual draw-bar pull.

Relation of Power to Weight of Locomotives.

Ordinary locomotives are capable of developing from 9 to 12 horse-power for each ton of their weight (inclusive of tender).

Weight Restriction in Locomotives.

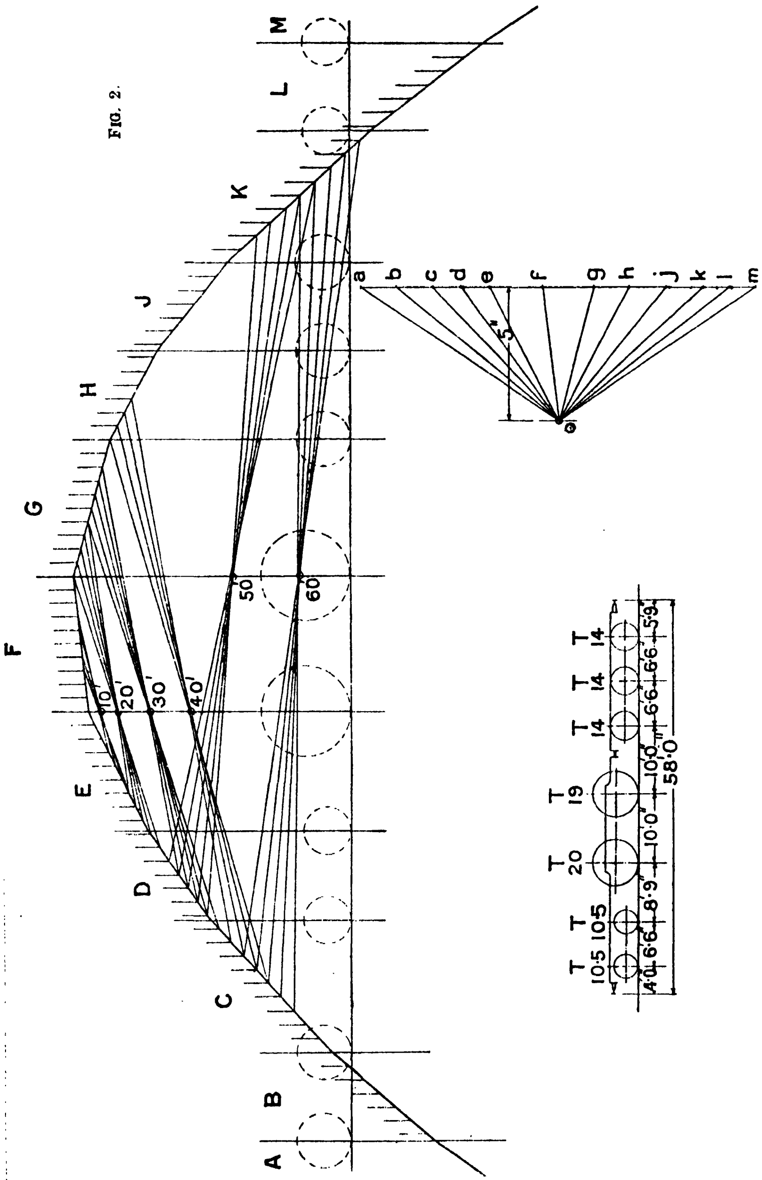
On English railways the heaviest load is usually 20 tons, but in the latest 3- and 4-cylinder engines a load of about 22½ tons has been permitted on account of the lighter hammer blow.

The axle loads carried with the goods engines are about 15 tons in Germany and Italy, 17 tons in England, Belgium, and France, and 23 to 30 tons in the United States. (In tons of 2,240 lb.)

The following is a formula given by F. J. E. Spring for locomotive driving-wheels :—

Safe load = 200 × diameter of wheel in inches × width of rail head on which wheel bears in inches.

In addition to restriction of individual loads on wheels, the total weight of a locomotive is limited by the strength of permanent-way structures. In designing a new locomotive, or rebuilding an existing one, it is essential to study the resulting bending moment imposed on bridges. This varies with the total length of engine, wheel base and spacing of wheels, and the weights on individual pairs of wheels. A short engine with the wheels close together may stress a bridge more than a heavier engine in which the weight is spaced over a longer base. The locomotive engineer, however, is not concerned in the actual stresses produced, but his problem is to keep within the boundaries of a limiting bending moment curve. The permanent-way engineer sets out the strength of the weakest bridges of various spans, and from them deduces a curve showing the maximum equivalent uniformly distributed load that the different spans can sustain. The curve is either set out giving the total load on the spans, or the tons per foot run of span. In fig. 3 the upper curve shows a limit determined by the permanent-way engineer in lb. per ft. run. No locomotive may run that sets up on any bridge a greater bending moment than this. If the excess is small, a slight modification of the wheel spacing or distribution of loads may rectify matters. As the locomotive constitutes a rolling load, any direct calculation is very tedious. The loads must be advanced across the span a foot or so at a time, and the bending moment ascertained for each step. It is obvious that maximum bending will occur under one of the loads, but it is not always certain which load it is. The maximum also changes from under one wheel to another at a certain point. The most satisfactory way is to first of all solve the problem graphically : if care is taken the results are very reliable. Once a solution is obtained the bending moment can be exactly calculated, and this acts as a check on the drawing and reading of scales. For the loading of the span it is assumed that there are several locomotives of the type under consideration, coupled together. As an example of the method, a 4–4–0 type engine is taken, the loads and spacing of wheels being shown on the line diagram at the foot of fig. 2. The first step is to set up a vertical line and mark it off to a convenient scale, say 1 in. = 10 ft., to represent the wheel loads to scale. Thus, *c–d* is the weight on the leading bogie wheels, *d–e* on the next pair, *e–f* that on the first pair of coupled wheels, and *f–g* for the trailing wheels. The loads on tender wheels are represented by *g–h*, *h–j*, *j–k*. The lengths *a–b*, *b–c*, are tender wheels of the preceding engine; while *k–l*, *l–m*, are for bogie wheels of the following engine. If the span is a long one more loads must be added in their proper order. Having done this a pole O is taken, approximately in line with the centre of gravity of the loads, and at a distance of, say, 5 ins. from the vertical line. The various points on the line are then joined to the pole. The next step is to draw a horizontal base line and mark on it the wheel base of the engine to a scale of, say, ½ in. = 1 ft. Verticals are projected through the points, and the spaces are lettered A, B, C, etc., to correspond with the letters *a*, *b*, *c*, etc., on the vertical load line.


A
B
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T 20
T 19
T 14
T 14
T 14
4'0"
6'6"
8'9"
10'0"
10'0"
6'6"
6'6"
5'9"
58'0"

FIG. 2.

Commencing at any convenient place, a line is drawn through space A parallel to the polar line o–a. This is followed by line through space B parallel to o–b, and so on until complete. For convenience a number of vertical lines are spaced at intervals representing each foot of the span. In order to ascertain the bending moment for a 10-ft. span, the intersection of a vertical with the curve is joined to another ten spaces away and scaling 10 ft. when measured horizontally. The next and succeeding points are treated in the same way, and it will soon be evident where the greatest bending moment is enclosed. This is equivalent to moving the bridge span along under the engine. Having ascertained the greatest bending for a 10-ft. span, that for a 20-ft.

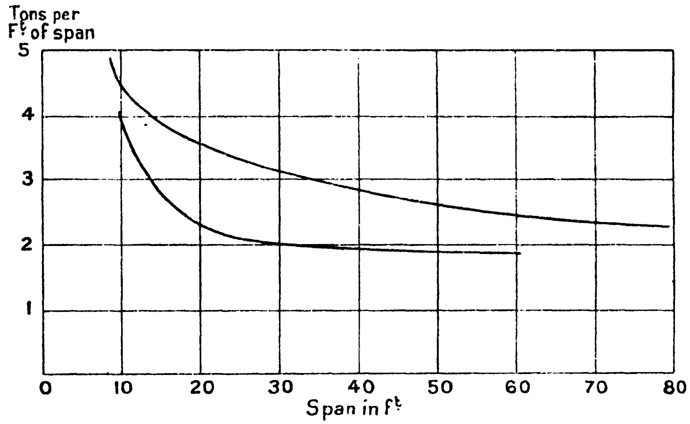


FIG. 3.

is obtained in the same way by joining verticals 20 ft. apart. The 30, 40, 50 ft., etc., spans are treated in like manner. It will be observed that the maximum B.M. moves from under the driving to the trailing wheels between 40- and 50-ft. spans. The maximum B.M. can be scaled by measuring vertically, and the results tabulated. If 1 in. = 10 tons of weight, and 1 in. = 2 ft. of span, and the polar distance is 5 in., the scale of bending will be 1 in. = 10 × 2 × 5 = 100 ton-ft. The final step is to obtain the equivalent uniformly distributed load by equating each result to it—i.e., B.M. = $\frac{WL}{8}$, so that W = $\frac{8 \times \text{B.M.}}{L}$, where W = total distributed load on span. For load per ft. run of span, B.M. = $\frac{wl^2}{8}$, and $w = \frac{8 \times \text{B.M.}}{l^2}$.

The results obtained can be plotted on the bridge curve supplied by the permanent-way engineer (as the lower curve in fig. 3), and unless all the plotted points fall below the upper curve, the design of the engine will have to be modified in some way, either by reduction of weight, or possibly by an alteration to the wheel base, or the addition of another pair of wheels.

When the position of maximum bending and the loads producing it are known, calculation is a simple matter, and it acts as a check on the graphical working. The position of load system giving maximum bending is such that the centre of span bisects the distance between point of maximum bending (which occurs under one of the loads) and centre of gravity of loads on span.

British Standard Unit Loadings for Railway Girder Bridges are given in B.S.S. No. 153, Parts 3, 4 and 5. The unit of axle load is taken as 1 ton. For British railways, the loading recommended in the Requirements for Passenger Lines and Recommendations for Goods Lines of the Minister of Transport in regard to Railway Construction and Operation (1925) is 20 units for heavy main lines, with lower multiples for lines constructed for lighter loading.

Thus the bending moments given in the Specification must be multiplied by 20 to obtain a 20 units loading, or by 16 in the case of a 16 units loading, etc.

Compound Locomotives.

The general claim for compound locomotives, by most authorities, is a saving of about 20 per cent. in the case of saturated steam. With superheated steam the saving is very much less. Expansive working, and consequently exhausting at a low temperature, is the main object.

Many of the larger locomotives recently constructed on the Continent are four-cylinder compounds, while the two-cylinder cross-compounds are favoured in the Argentine. A number of three-cylinder compounds has been built in England for the L.M.R. and in Ireland for the G.N.R. Some of the latest French locomotives are three-cylinder compounds, but four-cylinder compounds have also been built. Some experimental engines have been put into service in the U.S.A., including one for triple expansion working.

Ratios of the Areas of the High-pressure Cylinders to the Low-pressure.

Wordsell, Von Borries, and others, 1 to 2–2½ ; Webb, 1 to 2·3 ; Vauclain and America, 1 to 3. In modern compounds ratio is 1 to 2·4 on L.M.R., 1 to 2·5 Mallet compounds in U.S.A., and about 1 to 2·2 on Continent. Latest French practice 1 to 2·7.

Tractive Force.

Von Borries, who recommends a ratio of 1 : 2·2 or 2·3, gives tractive power of two-cylinder compounds as,

$$\text{T. F.} = \frac{psd^2}{4\,\text{D}},$$

where,

p = boiler pressure ; s = stroke in inches ; D = diameter of wheel in inches ; d = diameter of low-pressure piston.

For four-cylinder compounds, Baldwin gives

$$\text{T.F.} = \frac{2\,psc^2}{3\text{D}} + \frac{psd^2}{4\text{D}},$$

c being diameter of high-pressure piston.

Another formula is given as

$$\text{T.F.} = \frac{0\cdot 8\,psd^2}{(r+1)\text{D}} \text{ for two-cylinder compounds} = \frac{1\cdot 6\,psd^2}{(r+1)\text{D}} \text{ for four-cylinder compounds,}$$

r being ratio of low to high-pressure piston area.

Cylinders of Simple and Compound Locomotives of Equivalent Power.

For proportioning the diameters of the cylinders of compound locomotives when designing them to generate a power equivalent to that of a given size of simple cylinder and of the same piston stroke, the average ratio of area of the simple cylinder and the high-pressure cylinder area of compound locomotives is 1 to 1·25 for the 2-cylinder and 1·10 for the 4-cylinder. In the case of the 4-cylinder engines the area is that of the sum of the two high-pressure cylinders. (*Railroad Gazette.*)

Parts of Locomotives.

CYLINDERS.

The size of cylinders is governed by the maximum tractive effort required for starting the train, boiler pressure, diameter of driving wheels and adhesive weight available. $d^2l = \dfrac{\text{T.E.} \times \text{D}}{0\cdot 85\text{P}}$

where d = diameter of piston in inches, l = length of stroke in inches, P = boiler pressure, D = diameter of driving wheels in inches, and T.E. = tractive effort in lb.

The above formula refers to two-cylinder engines.

For two-cylinder engines, the diameter in inches is approximately $\dfrac{120}{\text{P.}}\sqrt{\text{I.H.P.}}$ **where I.H.P. = maximum H.P. that engine will be required to develop for any length of time, and P. = working pressure.**

For three- and four-cylinder engines the equivalent cylinder area can be calculated.

Stroke and diameter of driving wheels should be so proportional that the piston speed is about 900 ft. per minute, as this is found to be the most economical speed.

For steam pressures from 150 to 250 lb. per sq. in. and cylinder diameters from 15 in. to 22 in. thickness of barrel in inches may be taken as $t = \dfrac{pd}{4000} + 0\cdot 3$.

Thickness of metal of steam passages may by ·7 t and of the flanges 1·25 t, where t = thickness of barrel in inches.

The studs for the cylinder cover should be calculated for a stress of 4,000 lb. per sq. inch to allow a large margin of safety in case of priming.

For Flat Slide Valves.—Area of steam port $= d^2 \div 15$. Length of ports should not be less than $\cdot 85\,d$. Area of exhaust port should be two and a half times that of the steam port.

Clearance between cylinder covers and piston is usually made from ½ in. to ⅝ in. The *clearance volume*, including the passages, varies from 6 to 8 per cent. of the cylinder volume. If the clearance is too small the compression line at high speeds rises above the steam chest pressure, and the engine does not run well. *(Wolff.)*

Cylinders for Highly-superheated Steam.—Owing to the very high temperatures, it is necessary that the steam chest walls should be separate from the walls of the cylinders. There is a very great variation in the temperatures of superheated steam, on the one side, and of the exhaust steam on the other. If a common wall is used between the two the result is that the cylinders are liable to fracture. *(H. N. Gresley, 'Proceedings, Inst. Loco. Engineers.')*

Pistons.—One railway company makes all pistons 0·0625 in. smaller in diameter than the bore of the cylinder. Another railway allows 0·002 in. per inch of diameter for clearance.

VALVES.

For Flat Slide Valves.—*Outside lap* usually $= 0\cdot 8a$, where a is the width of the steam port. As a rule there is no inside lap, but in some cases a small inside clearance not exceeding $\frac{1}{16}$ in. is allowed. The *lead* is about ⅛ in., and the *maximum travel* is about $2\cdot 7a$. Valves of tank engines and others that make frequent stops should be of soft grey cast iron. For long-run engines, 5 parts copper, 1 part tin. *(Wolff.)*

PISTON VALVES.

Cylindrical or piston valves are extensively used in connection with superheated steam. A large proportion have internal admission of steam. Various packings are in use; some consist of spring rings, of which there are several kinds. The advantages over flat slide valves are reduced friction, larger port opening, and wear is confined to the easily renewable packing rings.

The diameter of piston valves is about half that of the cylinders, so that circumferential length of ports is approximately $1\cdot 5d$, where d equals diameter of cylinders.

The lap of the valves on the admission side is usually ¾ in. or 1 in., but this has been increased in recent years to 1¼ ins. or 1⅝ ins., and the travel in full gear working has increased correspondingly from about 4½ ins. to 6½ ins. The advantages gained are sharper cut-off and freer exhaust, leading to greater mean effective pressure at high speed and greater economy in steam consumption.

Piston valves are of two kinds, inside admission and outside admission. In the former, steam is admitted between the heads, and exhaust takes place on the outside. The advantages are reduced area for radiation of heat and packing of spindle is only subjected to exhaust steam pressure. Outside admission valves are arranged like flat slide valves, with admission of steam on outside and exhaust inside, thus giving a shorter and more direct passage to the blast pipe.

The cast-iron valve liners are turned 0·003 in. to 0·004 in. larger than the steam chest, and are drawn into position by means of screw tackle. At the W.R. works, Swindon, liners are shrunk in diameter by subjecting them to intense cold through contact with solid carbon dioxide. They are then pushed into position and allowed to expand.

LIMITS AND FITS FOR LOCOMOTIVE WORK.

The Locomotive Manufacturers Association of Great Britain has set up a code of standard practice in respect of limits and fits recommended for use in locomotives built by them. Through this code a greater measure of interchangeability can be effected by enabling spare parts to be designed in detail as well as manufactured in an identical manner at individual factories.

The code departs from British Standard Specification 164 for 'Limits and Fits for Engineering' in that it is based partly on the 'hole basis' and partly on the 'shaft basis'; it is termed by the Association a 'locomotive basis.' In addition to dealing with the sizes and tolerances for steam locomotive details of all descriptions, recommendations are made as to the forms and dimensions of rivets, mild steel sections and bars to be used.

MOTION WORK, ETC.

ALLOWANCES USED IN FITTING THE PARTS.

		In.			In.
Gland and neck rings	+ piston rod	0·03125	Slipper flanges	+ width of bar	0·03125
Eccentric straps	+ sheaves	0·015	Axle-box flanges	+ horn	0·03125
Side play of do.		0·03125	Brake gear holes	+ pin dia.	0·0156
Slide blocks	− bar opening	0·006	Valve gear pins	− holes	0·004

(F. Thompson, 'Journal of Inst. of Loco. Engineers.')

Coupling and Connecting Rods.

In the case of outside cylinder engines the throw of coupling rods is usually the same as that of the connecting rods, namely, 12 ins. to 15 ins., but in a few cases the crank pins are turned eccentrically on their outer bearing, so that the throw of the coupling rod may be shortened a little. For inside cylinder engines the throws are from 9 ins. to 11 ins. Small throws require rods to be stiff laterally, in order to resist crippling loads, while large throws need a deep rod on account of the bending stresses set up by centrifugal force.

Where a fire-box comes between two coupled axles, some railways allow a little for expansion by heat of the frames by shortening the centres between axles by $\frac{1}{32}$ to $\frac{3}{64}$ in. When steam is raised the frames lengthen, so that the distance between centres of the axles is equal to the length of the coupling rods.

Most coupling rods have plain bushed ends, as have the solid-ended connecting rods of outside cylinder engines. The gun-metal bushes are turned slightly larger than the hole by an allowance of 0·002 inch per inch of diameter. This requires a tonnage of 12 tons to press a 7-inch bush into place. A running clearance on the crank pin of 0·002 inch per inch of diameter is specified in one case, $\frac{1}{64}$ inch being a common figure. Another railway presses the bushes in with a load of 20 to 25 tons, and allows $\frac{1}{32}$ in. to $\frac{3}{64}$ in. clearance on the pin.

Valve Gear.

With inside cylinders Stephenson valve gear (see also Sec. XXVII, Part III) is general. Outside cylinder engines, with the valves inside the frames, also have the Stephenson gear; but, when valves are outside, the Walschaerts valve gear is now preferred.

The outside cylinder engines on the former Great Western Railway have 10-in. piston valves placed above the cylinders worked by Stephenson gear from the inside of the frames. The maximum valve travel is $6\frac{1}{4}$ ins., the lap being $1\frac{5}{8}$ ins., and the port opening $1\frac{1}{2}$ ins., although the steam ports are $1\frac{3}{4}$ ins. wide. The eccentrics are set to give $\frac{1}{8}$ in. lead at 25 per cent. cut off, resulting in the valves being about $\frac{1}{8}$ in. blind in full gear. This arrangement gives a sharp cut off, prolonged expansion and a free exhaust, resulting in large mean effective pressure at high speeds.

Walschaerts Valve Gear.

The Walschaerts valve gear, figs. 4 and 5, is mainly used on outside cylinder engines. The travel of the valve is derived from two movements, the crosshead of the engine and a return

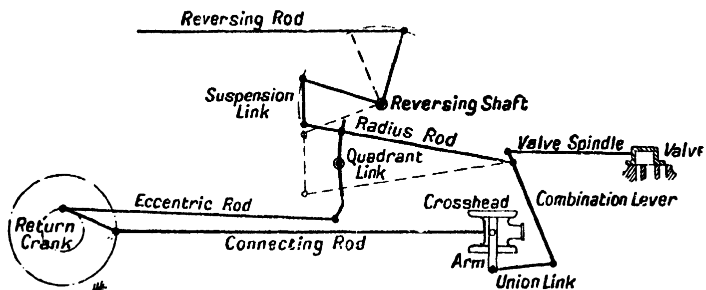


FIG. 4.

crank attached to the main crank pin, or, in the case of inside cylinder engines, from an eccentric on the crank shaft. The first movement is a reproduction of the travel of the crosshead to a reduced scale, the reduction being effected by a combination lever connected at its lower end to the crosshead through a link. In order that the angular movement of the lever shall not be too great by any restriction of length an arm is rigidly attached to the crosshead so that the lever may be suitably lengthened. The combination lever is also attached to the valve spindle and to the radius rod which forms its fulcrum. With an outside admission valve the valve spindle is attached above the radius rod and with inside admission of steam the positions are reversed (see figs. 4 and 5). The function of the combination lever is to give the mid-gear travel, equal to twice 'lap + lead,' and the lever is so proportioned that the stroke is reduced to this amount. The lead is constant for any length of travel.

The second movement, derived from the return crank, is transmitted to a quadrant link rocking on fixed bearings. The maximum angle through which this link swings should be kept

within 45°. The radius rod is attached to the link through a die block and communicates its motion to the valve by way of the combination lever. This second movement is at a phase of 90° with the first, and its magnitude can be varied and its direction reversed by altering the position of the radius rod on the quadrant link. In mid-gear the second movement is nil, only the first movement being communicated to the valve. The position of the radius rod in the link is regulated by the reversing shaft, the attachment being either a vibrating suspension link (see fig. 4) or a slider consisting of a die block working in a long slot, as in fig. 5. In the latter case it is

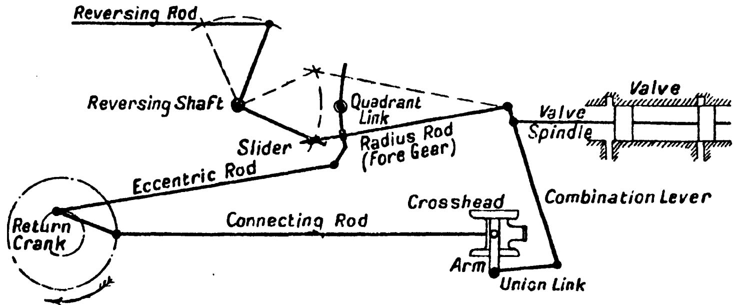


FIG. 5.

more convenient to use the bottom of the link for fore-gear running, while in the former either top or bottom can be used, since it is only necessary to alter the position of the return crank, which is nominally 90° ahead or behind the main crank. For convenience of design the quadrant link is usually raised so that the eccentric rod is inclined, and this angle of inclination should be taken into account when setting the return crank, being added to or subtracted from the main angle of 90°. The setting of the return crank is as follows :—

Valve.	Fore Gear.	Return Crank.
Outside admission	At top of link	90° behind main crank
" "	At bottom of link	90° in advance of main crank
Inside "	At top "	90° in advance of "
" "	At bottom "	90° behind "

BAKER VALVE GEAR.

The Baker valve gear is extensively used in America, and has been successfully adopted in New Zealand and elsewhere. It is similar to the Walschaerts gear as far as that lap and lead are obtained from a crosshead-driven combination lever, but in place of the quadrant link and its die block, a form of Marshall gear is used. An eccentric rod operates a swing link whose point of suspension can be moved to vary travel and to reverse the motion. Longer valve travels are possible than given by Walschaerts gear.

LENTZ POPPET VALVE GEAR.

Each cylinder is fitted with 4 poppet valves. These are disposed in pairs, one for admission and one for exhaust, at each end of the cylinder casting. A central annular chamber is provided in the cylinder casting, through which a camshaft runs at right angles to the valve spindles, which camshaft has mounted on it 2 cams, one of which operates the steam valves and one the exhaust. The cams may come into direct contact with the valve spindles, which in such cases are fitted with a roller and pin, the former bearing upon the cam profile. Alternatively intermediate levers are fitted which are arranged on a suitable fulcrum support.

The fulcrum support is a special feature of the design in question; it carries the fulcrum pins and may be disposed above the cams or below them in the cam space, as may be most convenient. This fulcrum bar fits in the end of the cam space and also in the cam chamber cover, being securely held in order to prevent it from any possible movement. This arrangement permits of the withdrawal of the fulcrum bar, together with intermediate levers for inspection purposes, and also greatly facilitates the withdrawal of the cams and camshafts for a like purpose.

The intermediate levers are fulcrummed at one end; at the opposite end they bear against the valve spindles, and in the centre a special roller and pin is fixed, the roller bearing against the cams. The advantage that the intermediate lever system has over what is known as the direct

drive lies in the fact that better port openings are secured, as for any given cam profile the valve lift is doubled.

The valves are securely held on the spindles by an improved form of combined nut and inner spring cap, and the spindles, which are hardened and ground, work in cast-iron bushes pressed into the cylinder casting.

Each valve and spindle is fitted with a control spring and a control spring adjusting device fixed in the valve covers. The function of the springs is to keep the valves following the cams when the locomotive is running with steam shut off.

The poppet valve arrangement described is operated by one or other of the usual type of valve motion common in locomotive practice, and to this end the valve connecting-rod joins the upper part of the valve motion with a rocker arm securely keyed to the outer end of the camshafts. With the motion in full gear the camshaft has imparted to it a maximum angular movement corresponding to the greatest valve opening or travel. As the engine is linked up, the amount of motion imparted to the rocker arm becomes less, and likewise the valve opening, and in this manner the cut-off is made earlier or later, as may be desired.

With poppet valves more direct steam passages are obtainable than is possible with piston valves, and this is especially so for the exhaust. This together with the improved port openings possible with poppet valves, as against piston valves for any given degree of cut-off, are amongst the important advantages offered by a poppet valve gear as described as against piston valve gear.

One of the chief principles underlying the design of poppet valves as arranged in the above manner is a slow opening followed by a quick movement of the valve, a rapid closing followed by a slow movement at the actual time of closure, and also the fact that up to the actual seating of the valve on its face the valve spindle either directly or indirectly (through the intermediate system) is in contact with the cam profile. It is in this way that absolutely noiseless operation is assured, there being no drop action whatsoever.

Steam-tightness of the valve is at all times assured through the special design of the inner and outer valve face.

The Lentz valves have also been constructed to work with a rotary cam drive instead of the usual valve gear.

Caprotti Valve Gear.

This is somewhat similar to the Lentz, but the valves are vertical instead of working in a horizontal direction. The valves are lifted by cams on a rotating shaft, which derives its motion from bevel wheels on one of the main axles. Variation in cut-off and reverse is arranged by altering the angular advance of the admission cams on their shafts by an ingenious mechanism. The exhaust cams are operatated independently and give a fixed point of exhaust.

In the U.S.A. the Franklin poppet valve gear has been fitted to a few engines.

CRANK AXLES.

If W = weight of engine in tons, when in working order; D = diameter of driving wheels in inches; then

$$d = K\sqrt[3]{WD}.$$

For the best modern practice, K = ·522. (*Wolff.*)

Dimensions taken from the crank axles of a number of modern British locomotives, with cylinders ranging from 17 to 21 ins. diameter, give the following rule:—

$$d = 0{\cdot}375C + 1,$$

d being the diameter of main bearings and C the diameter of cylinders.

The crank-pin journals are usually the same diameter as the main bearings, the barrel or centre portion about ½ in. less and the wheel seat 1 in. more. Thickness of webs varies from 4 to 4½ ins.

Built-up crank axles are common. The advantages over a forged axle are cheapness, more effective bearing surfaces due to absence of radii, fractured parts may be replaced without scrapping the whole axle. By extending the webs a balanced axle is obtained. Special steels may be used.

The parts are assembled by shrinkage fits, the webs being heated in a gas furnace and shrunk on to the shaft and pins in special fixtures to set the throws in their correct positions. Holes are drilled between web and shaft, into which screwed dowel pins are fitted as keys.

The shrinkage allowance, or the amount by which the hole is smaller than the pin, depends on the tensile strength of the crank webs. For steel to B.S.S. 8c this is 0·025 in. in Horwich practice for axles about 8 ins. diameter.

AXLE-BOXES AND BEARING SURFACES.

In modern practice the straight axles of coupled wheels have no collars. Lateral thrust on axle-boxes is taken on wheel hubs only.

The simplest axle-box is one of solid bronze or gun-metal, with a keep of the same material. The crown is recessed to form pockets for a white-metal lining over the bearing surface. The sides of the box slide against steel or cast-iron horns.

Boxes forged from steel are fitted with a bronze or gun-metal bush, white metal lined, and the sides lined with the same metal as the bush.

Cast-steel boxes have a bronze or gun-metal crown pressed in with a load of 8 to 9 tons. The bearing surfaces are covered with a thin layer of white metal. Latest practice is to face the horn surfaces with thin plates of manganese steel (11 to 14 per cent. Mn.). The plates are bedded to the sides of the axle-box and welded to it. The horns have a compound assembly consisting of a manganese steel plate riveted and welded to a mild steel backing plate. The final machining of the mild steel surface affords an easy method of adjusting the horn gap.

Angle included at centre of journal by arc of contact is usually for coupled wheels, 160° to 170°, for bogie, carrying, or trailing wheels, 60° to 80°. The maximum bearing pressures are approximately 250 lbs. per sq. in. of projected area for coupled wheels, 350 lbs. for bogie, and 450 lbs. for tenders. Bearing brasses are commonly of gun-metal to Admiralty specification and the white-metal lining of Babbitt. In some cases alloys having a lead base are used. Maximum crown wear of brasses or axle-boxes is ⅛ in.

Lubrication is introduced either at a transverse slot at top centre, or at two slots at 30° each side of the vertical centre line.

Lubrication of bearings is by top feed from crown of box or from auxiliary oil boxes, and underpad of wool on spring frame. In case of bogie and tender wheels lubrication is often by underpad only.

The 'Isothermos' axle-box is used on the Continent for tenders. A disc or thrower dipping into a reservoir of oil raises the lubricant by centrifugal force to an upper chamber, whence it flows to the bearing and returns.

The lubricating oil is a medium mineral oil mixed with about 10 per cent. of rape oil. An average oil consumption is 6 to 7 oz. per bearing per 100 miles.

In U.S.A. cakes of hard grease are used in place of oil: these are placed in the keeps and pressed under strong spring action through a perforated metal plate against the journals. The cakes last for some time and obviate the frequent attention necessary in the case of oil lubrication. In some cases grease lubrication is extended to coupling and connecting rods, and even to valve motion.

Roller bearings are being adopted to an increasing extent for axles of carrying wheels and for tenders. In the case of inside bearings of coupled wheels it is usual to house the two bearings in a cannon box embracing the axle. A roller-bearing box has, however, been produced for bearings of crank axles, where the cannon box cannot be adopted.

For crank-pin journals and coupling-rod pins, the pressure may be as much as 1,500 lb. per sq. in., the load on connecting rod being taken as full boiler pressure on piston area.

For crosshead pins 4,000 lb. per sq. in. is allowable on account of the small angular movement. Crosshead slippers should be about 60 lb. per sq. in., a low pressure being advisable on account of the exposure of slide bars to sand and grit.

WHEELS AND TYRES.

Cast-steel wheels are general. The method of securing tyres by means of set screws through the rim of the wheel is the most common, but this practice is not favoured to-day. The alternative is to depend on the shrinkage of the tyre to grip the wheel, a retaining ring being inserted to lock the tyre in place laterally. This is a cheap and satisfactory fastening, but there is a tendency for tyres to become loose as they wear thin.

The shrinkage allowance for tyres is usually between 1/750th and 1/1000th of the diameter of wheel centre.

In W.R. practice the total shrinkage is given as $\frac{1\cdot0625\,D + 10}{1{,}000}$**, D being the diameter of wheel in inches.**

Driving wheels have tyres with flanges about ⅛ in. thinner than the other wheels. This eases the engine on curves and protects the crank axle to some extent. Blind or flangeless tyres are uncommon in England, although often resorted to in America.

Of modern types the 4–4–0 engine is the most economical in tyre wear. The flanges of the leading bogie wheels which guide the engine wear much faster than the second pair. Bogie wheels wear less than radial truck wheels, as a bogie sets itself tangentially to any curve. In engines of the 0–6–0 and 0–8–0 types mileage is determined by the flange wear of the leading coupled wheels.

Tyres with treads worn hollow are detrimental to points and crossings. A flange worn sharp is liable to cause derailment. Tyres are turned down to 1⅝ ins. on tread and allowed to wear to 1½ ins. before they are scrapped, except in case of large driving wheels, where 1¾ ins. or 2 ins. is the limit. The thickness of new tyres is usually 3 ins.

Wheels are forced on axles by hydraulic pressure. A minimum of 10 tons per inch diameter of wheel seat should be allowed when rape oil is used as a lubricant, 100 tons being a usual amount for large axles; or, if D is the diameter of wheel seat in inches, the load = 10D tons.

The wheel is bored slightly less in diameter than the axle, the difference expressed as a fraction of an inch being from $\frac{2D + 3 \cdot 5}{1,000}$ to $\frac{4D + 2}{1,000}$, depending on rigidity of the boss, quality of steel, etc.

In some cases the wheel and wheel seat are slightly tapered, the difference in diameter in the length of the wheel seat being $\frac{1}{32}$ in. The W.R. practice is to bore the hub parallel to and smaller than the axle by an allowance varying from 0·0015 to 0·002 per inch of diameter. Tonnage for pressing on the wheels by hydraulic press is from 8 to 12 tons per inch of diameter, the intensity of the grip depending to a large extent on the thickness of the hub. The best results are obtained when this is not less than half the diameter of the axle in thickness.

If the tyre is on the wheel, it takes 25 per cent. more pressure to force the wheel on the axle for the same allowance.

Keys should be tapered $\frac{3}{64}$ in. per foot and fitted to within $\frac{3}{4}$ in. before being driven home.

Crank pins should be forced in with a minimum pressure of 13 tons for each inch of diameter and the ends of the pins riveted over inside the wheels.

BRITISH STANDARD CONTOURS FOR LOCOMOTIVE TYRES.*

(FOR BRITISH RAILWAYS 4 FT. 8½ IN. GAUGE.)

(No. 276—1927.) (*Abstract.*)

Contour A recommended for the leading and trailing wheels of bogies, the wheels of pony trucks and radial axles, or any pair of wheels which constitute the leading or trailing wheels of a locomotive, whether tender or tank engine.

Contour A or G recommended for the leading wheels of a set of coupled wheels, when led by a bogie, pony truck, radial axle, or any pair of carrying wheels.

Contour A or G recommended for the trailing wheels of a set of coupled wheels, when followed by a bogie, pony truck, radial axle, or any pair of carrying wheels.

Contour G or B recommended for coupled wheels, other than the leading and trailing coupled wheels of groups 6, 8, and 10 coupled.

(In practice, the use of the G Contour is being given up in favour of A or B.)

FLEXIBLE WHEEL BASE.

Main-line engines should be capable of traversing a curve of 5 chains radius, and sufficient side-play is given to bogies and radial wheels for this. The theoretical amount required is not attained in practice, as engine frames and track spring a little on sharp curves.

There is a small amount of side-play allowed between flanges of wheel and rail. With new tyres on new rails, this is about $\frac{5}{16}$ in. each side, or a total of $\frac{5}{8}$ in. The flanges of driving wheels are thinned down to give about $\frac{3}{4}$ in. to $\frac{7}{8}$ in. total. Additional side-play is sometimes given by allowing a little clearance between axle-boxes and guides. It is very exceptional to give any side-play between journal and bearings.

Baldry's rule for the radius of pony trucks and radial axleboxes is

$$R = \tfrac{1}{2}\left(B - \frac{A^2}{B}\right),$$

where B is horizontal distance from truck wheels to centre of rigid wheel base and A is half rigid wheel base.

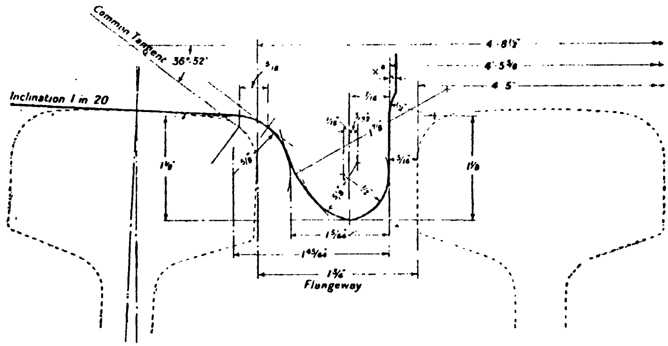
Bogies, in addition to turning on the pin, are allowed a certain amount of transverse movement. This is controlled by springs, suspension links, or inclined slides. Some bogies have curved slides giving radial instead of transverse movement. Fixed centre bogies are now but little used.

Counterbalancing Locomotive Driving Wheels.

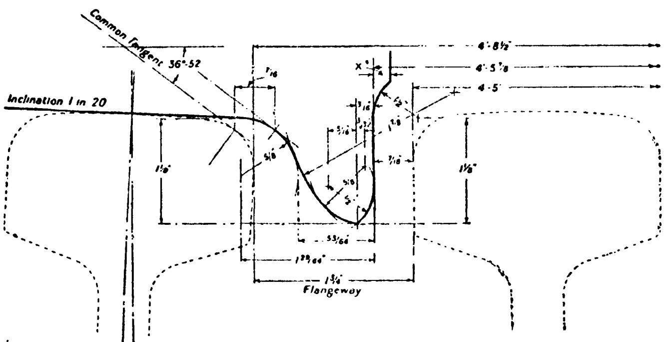
The revolving and about 60 per cent. (this figure varies considerably) of the reciprocating parts should be counterbalanced. The tendency at the present time is to reduce this figure, and in three- and four-cylinder engines it has been as low as 40 per cent. In some recent three-cylinder engines the counterbalance for reciprocating masses has been omitted altogether, so that no 'hammer blow' is exerted on the rails.

* By permission of the British Standards Institution.

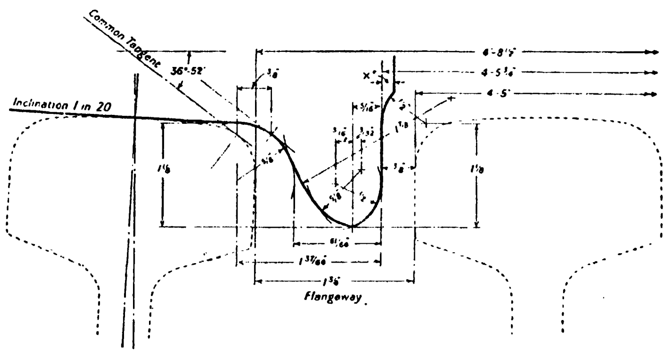
British Standard Contours for Locomotive Tyres for British Railways.



B.S. CONTOUR A.



B S CONTOUR E.



B.S. CONTOUR G.

The revolving parts develop centrifugal force, the horizontal component of which disturbs the engine longitudinally, to which must be added that due to the momentum of the reciprocating parts: also, these two forces produce a sinuous motion by acting first upon one side of the engine and then on the other. The vertical component of the centrifugal force generates a rolling motion by acting alternately upon each side of the engine. A peculiar effect is also produced by the obliquity of the connecting-rod to the slide bars, a turning upon a horizontal axis, together with an up-and-down movement at the crank axle. The horizontal forces are the most injurious, although some American engineers consider the vertical to be so; but English practice is to take a medium course between excessive horizontal and vertical disturbing influences. It has been clearly demonstrated that high centres of gravity are advantageous.

The balance weights in the wheels do not revolve in the same vertical plane as the parts to be balanced, and it is therefore necessary to balance moments as well as forces. The weights for each part must be split into two, and their proportions will vary with the distance of the part away from the balance weight plane. If this distance is a and the distance between weights in left and right wheels is b, a primary weight of $(a + b)/b$ times the weight of the part must be placed in the near wheel, diametrically opposite to the crank pin, and a secondary one of a/b times in the far wheel, on the same side as the crank pin, if the part revolves outside the wheels. If the part is between the wheels a primary weight of $(b - a)/b$ is placed in the near wheel and a secondary of a/b in the further one, both opposite to the crank pin. There will, therefore, be as many primary and secondary weights as there are vertical planes of revolving and reciprocating parts. These are summed up and the weights combined in each wheel to form a single weight at an angle to the crank pin. This is calculated at crank pin radius and reduced afterwards in accordance with the distance of the centre of gravity of the weight when placed at the rim of the wheel. This weight is usually made of crescent shape.

American practice has been to allow 1/400th part of the engine weight to remain unbalanced in the reciprocating parts on each side. The remainder is then divided amongst the coupled wheels;

$$r = \left(R - \frac{W}{400}\right)/N,$$

where,

r = reciprocating weight in each wheel at crank pin radius; R = total weight of reciprocating parts; W = weight of engine in working order; and N the number of coupled axles.

The Association of American Railroads has recently investigated the effects of balancing in locomotives upon the track and upon the riding of locomotives. Tests were carried out within the limits of 56 per cent. to zero reciprocating compensation, that is, from 4·6 to 12·8 lb. per ton (of 2,000 lb.) reciprocating unbalance. The disturbing effect of reciprocating compensation was judged to be unduly high when the reciprocating unbalance was more than 8 lb. per side per ton of total locomotive weight in working order. An overbalance of 100 lb. per wheel for all drivers showed a low hammer-blow effect on the track, and this was increased to 200 lb. per wheel without causing excessive hammer blow. Over 200 lb. it became undesirably high. Because of the disturbing effect of the vertical component of the connecting rod thrust some advantage is to be obtained by placing 50 lb. less overbalance in the main driving wheels than in the coupled wheels.

The English method of cross balancing is being taken up in America.

Cast steel wheels have the balance weights cast solid with them; occasionally the weights have pockets cored out for the reception of lead. Another practice is to cast the wheels without weights and to rivet steel plates to the spokes. Molten lead is then run into the pockets formed between the spokes.

In order to obtain uniformity of results, some railways make use of a wheel-balancing machine. The pair of wheels to be tested is mounted on bearings supported by springs set radially round them. Weights are attached to the crank pins to represent the revolving and a proportion of the reciprocating parts. The wheels are then spun round at high speed and the counterbalance adjusted until steady running is obtained.

BOILERS.

Locomotive boilers should have a factor of safety of 5: considerations of weight make a larger factor undesirable. The copper fire-box stays should not be stressed greatly in excess of 4,000 lbs. per sq. inch, as copper loses much of its tensile strength at the steam temperature, while steel is not greatly affected.

Coned boiler barrels, although more costly, are adopted by some railways in order to obtain the most powerful boiler for a given weight, by the reduction in diameter at the front end, while allowing ample space by reason of the larger diameter in the region of the fire-box tube plate, where steam generation is at a maximum.

Some railways prefer the 'Belpaire' type of fire-box, while others adopt the round top with direct crown stays. In British practice the long narrow grate is almost universal for lengths up to 10 ft. 6 ins. and areas up to 33 sq. ft., but as this is insufficient for engines working the heaviest main line trains to the North, the wide type of fire-box has been introduced with grate areas up to 50 sq. ft.

This practice necessitates a carrying axle under the fire-box.

In America the wide fire-box is almost universal due to the large grates required, up to about 110 sq. ft., and to the use of bar frames which allow less width of fire-box than with the plate frames employed in British practice.

Most American fire-boxes of over 50 sq. ft. area are fitted with a mechanical stoker or are oil-fired.

Steel or Monel metal fire-box stays are now being extensively used in place of copper.

Steel fire-boxes are being more extensively used owing to the ease with which they can be constructed and repaired by arc welding. Tube ends can be seal welded to them after being expanded by rolling.

The all-welded boiler is under trial in many countries. The barrel and shell is stress relieved and all seams subjected to X-ray examination. Caution is being observed until more experience is gained with the boilers in service.

Boiler Capacity.—The maximum evaporation per sq. ft. of firebox (including combustion chamber, thermo-syphons or water-tubes) is about 55 lb. per hour. Evaporation from fire tubes is approximately 10 lb. per sq. ft. per hour, depending on length, diameter and spacing, and from superheater flue tubes about 11½ lb.

The maximum continuous rate which it is desirable to maintain in service is 12 to 13 lb. per sq. ft. of total heating surface per hour.

Grate Area and Heating Surface.

$$\text{Grate area in sq. ft.} = \frac{\text{average I.H.P.}}{25 \text{ to } 45}$$ the smaller denominator being used for small, low-pressure engines, and the larger one for the latest type of high-power, high-pressure engines.

The ratios giving the best results are as follows: Grate to total heating surface, 1 : 55 to 65; grate to fire-box heating surface, 1 : 5 to 5·5; and fire-box heating surface to total heating surface, 1 : 11 to 12.

Bituminous coals require space in the fire-box for the proper combustion of their gases before entering tubes. The fire-box volume should be 6 or 7 cub. ft. for every square foot of grate. Under these conditions 15 lb. of air are needed for each pound of coal burnt. The air spaces through fire-bars should not be less than 33 per cent. of grate area, and the damper opening not less than 15 per cent. Air spaces of 50 per cent. and damper opening 20 per cent. are not too large. In coupled engines, where an axle comes close under the grate, the slope of pan over it should be not less than 45°; and the restricted space for air between pan and fire-bars must have an area of at least 15 per cent. of the part of the grate which it supplies.

Fire-bars are of wrought iron or cast iron, and usually ⅞ inch wide at top. Gaps of ⅜ inch to ½ inch between these give an air space of 30 per cent. to 40 per cent. of the grate area.

Boiler Tubes.

Diameter and Length of Tubes.—The diameter of tubes varies with the length of barrel. Short barrels require 1¼ in. to 1⅝ in. outside diameter tubes; those over 10 ft. long, 1¾ in. to 1⅞ in. tubes. Long barrels, 13 ft. and upwards, require tubes 2 ins. to 2¼ ins. diameter. The internal diameter of a tube should be approximately 1/100th of its length.

Usual Thickness of locomotive boiler tubes, 5-in. superheater flue tubes, No. 8 W.G.; 1¾-in. tubes, No. 11 W.G.

British Standard.—For 1⅝ in. outside diameter, 13, 12 or 11 s.w.g.; for 1¾ in., 1⅞ in., and 2 ins., 12, 11 or 10 s.w.g.; for 2¼ ins., 11 or 10 s.w.g.

Water space between tubes should not be less than ⅝ in. for 1¾ in. diameter tubes, and ¾ in. for 2¼ in. tubes at the fire-box end, but may be pitched 1/16 in. closer at the smoke-box end.

Life of Boiler Tubes.

The life of copper and steel tubes in locomotive boilers, taken over a period of eight years is as follows:—

	Miles.
Copper 1st period (new)	110,000
„ 2nd period (drawn)	80,000
„ 3rd period (pieced)	50,000
Total	240,000

Subsequently 30 to 40 per cent. of those pieced are treated so again.

	Miles.
Steel 1st period (new)	70,000 to 80,000
„ 2nd period (pieced)	30,000 to 40,000

(*G. Hughes.*)

After withdrawal from the boiler steel tubes are de-scaled and the ends cut off. The tubes are safe-ended by butt welding on a short piece of new tube, the joint being ground flush. They are then swaged and belled in readiness for a further period of service.

Alternatively, the shortened tubes are used in boilers having shorter barrels than those from which they were withdrawn.

Calculation of the Heating Surface of Boilers for Locomotives.

The Association of Railway Locomotive Engineers' method of calculating the heating surfaces of locomotives is as follows:—

Fire-box.—The outside (wetted surface) of inner fire-box plates to be taken to top of foundation ring. Areas of tube-holes and sectional area to wetted side of fire-hole ring, or flanged plate, as the case may be, to be deducted; section of fire-hole being taken close up to the inner fire-box plate.

No deductions to be made for copper stays or roof stays. If the boiler has fire-box water tubes the surface to be taken on the inside (wet side) of tube.

Tubes.—The outside surface of tubes (wet side) to be taken, calculated on uniform diameters for both ordinary and superheater tubes; the length of tube being taken between tube-plates.

Elements.—The inside surface (steam side) of elements to be taken, the effective length of tube being from and to smoke-box end of large flue tube.

Smoke-box tube-plate.—No allowance to be made.

Equivalent heating surface of superheaters.—In order to compare the steaming capacities of superheater and non-superheater boilers, an equivalent heating surface is sometimes given for the superheater boiler, as owing to the lesser evaporative surface of the tubes, it appears less powerful. If 1·5 times the superheater surface is added to the evaporative heating surface, an equivalent heating surface is formed, which may be compared with that of a non-superheater boiler. Thus, a non-superheater boiler has 2,455 sq. ft. of heating surface, but when converted to superheater has 2,031 sq. ft. of evaporative heating surface and 460 sq. ft. of superheating surface. The addition of 1·5 times the superheating surface to the evaporative gives a total of 2,721 sq. ft., and indicates that the boiler is 11 per cent. more powerful when superheated.

Fire-Boxes.

Many attempts have been made in the past to substitute steel for copper plates in fire-boxes of British locomotives. Electric arc welding now offers improved facilities for the construction and repair of steel boxes, and their number is increasing. Steel fire-boxes are universal in America, where conditions favour their adoption. Greater success had attended the introduction of steel side stays for copper fire-boxes. Owing to the higher tensile strength the diameter over threads is about ⅝ in. for a pitch of 3½ ins. The stay projects through the fire-box casing plate about ¼ in., but the projection through the copper plate is furnished with a hexagon nut of reduced thickness. This forms a renewable head for the stay and protects it from injury by the fire. Similar stays are also made in Monel metal.

In locomotive practice a pitch of about 4 ins. is almost invariably adopted for the screwed stud stays in the fire-box. Theoretically this is much closer than is necessary, but the practice is good, as it provides a substantial margin of wear in the copper plates.

Life of Fire-Boxes.

The life of fire-boxes depends more upon the tons hauled than the actual miles run. Copper fire-boxes on the Lancashire and Yorkshire section of the L.M.R. have a life of from 10 to 16½ years. The older engines with 140 lbs. pressure are found to run their fire-boxes up to as much as 500,000 miles: an average figure for 70 engines being 310,000 before replacement by new ones. In standard engines, such as the radial tank and 6-wheeled coupled goods classes, working at 160 lbs. pressure, the average life of a copper fire-box is eleven years, with average mileages of 272,000 and 235,000 respectively. These figures are based upon records of 120 locomotives. Individual copper tube-plates in the fire-boxes above named require renewing during the life of the box. In the radial tank class, records of 53 tube-plates show an average life of seven and three-quarter years with 205,000 miles before renewing, whilst the 6-wheeled goods average 135,000 miles for 144 tube-plates. The remaining plates, viz., door and wrapper plates, practically do not require replacing during the life of the fire-box, but certain repairs and patches are fitted if necessary.

Smoke-Boxes.

In superheater engines the blast pipe is set some 30 ins. forward of the tube-plate, to allow ample room for the superheater header, and the orifice is about 7 ins. below the centre line of the boiler, but this figure varies considerably. The blast pipe should not be too low or there is risk of cinders being drawn into it. The diameter of the orifice should be as large as possible consistent with good steaming. For cylinders 16 ins. to 22 ins. diameter, the orifice varies from

4½ ins. to 5½ ins., 5 ins. being usual with a 19-in. cylinder, but a good deal depends on class of coal, work performed, and other conditions.

Chimneys are made as high as load gauge permits, and are extended downwards into the smoke-box. The main consideration as to dimensions is that a cone of steam diverging from the blast pipe orifice with a total angle of 1 in 6, should just pass clear through the top of the chimney, leaving a small annulus for the gases. The chimney should be slightly tapered and terminate in a bell, so that gases may be gradually entrained and increased in velocity.

In American practice a diaphragm is fitted across the smoke-box above the top row of tubes and carried down just below the blast-pipe orifice, terminating a little in front of it. By this means the draught on the tubes is equalised and the emission of sparks prevented. Some British engines are also fitted with this device.

The smoke-box vacuum obtained in modern superheater engines varies with the horse-power developed, and is about 5 in. of water, with a maximum of 7 in., in heavy passenger work, and about half these amounts for superheater goods engines.

Superheaters.

The superheater of a locomotive boiler performs two functions. In the first place, it acts as a steam drier by evaporating the moisture carried over with the steam to the cylinders. Secondly, the dried steam is heated some 200°–400° F. above its normal temperature at the working pressure, and a volumetric increase occurs. By superheating the steam, not only is the loss due to moisture avoided, but initial condensation in the cylinders is eliminated or greatly reduced. Losses due to leakage are also less. The expansion line of indicator diagrams taken from cylinders using superheated steam show a more rapid fall of pressure than with saturated steam. On the other hand, the compression line is also lower, showing less back pressure, so that there is little difference in the mean effective pressure to that of cylinders using saturated steam at the same initial pressure and cut-off. The orifice of the blast pipe with a superheater engine needs to be about ¼ in. smaller as the weight of steam discharged in a given time is less than in saturated steam engines. The chief gain due to superheating the steam to a final temperature of about 650° F. results in a saving of from 15 to 25 per cent. in fuel and 20 to 30 per cent. in water. The steam consumption of saturated engines is approximately 27 lb. and coal consumption 4 lb. per i.h.p. The corresponding consumptions of engines superheated about 200° F. are reduced to 19 lb. of steam and 3·0 lb. of coal per i.h.p.

Boiler repairs are also less, due to smaller demand for steam and lower rate of combustion of fuel. Engines fitted with superheaters can run longer distances without replenishing water and fuel supply. This is particularly advantageous in the case of tank engines, where the supplies of both are limited. The superheater adds little or nothing to the starting power of an engine, but it enables a larger horse-power output to be maintained, and consequently the hauling power is increased when running. Engines which had become obsolete on account of the growth of train loads have been given a new lease of useful life by the addition of a superheater. In some cases it has been successfully applied to engines with flat unbalanced slide valves between the cylinders, the only alteration being the provision of new tube plates for the superheater flues. A moderate degree of superheat does not appear to affect bronze valves provided great care is taken with the lubrication. Either a mechanical force-feed or a sight-feed displacement lubricator can be used, but the lubricant must be of first-class quality. With piston valve engines, the chief trouble is the formation of carbon deposit. This clogs the packing rings, chokes the ports and coats pistons, covers valve heads and spindles, etc. When steam is shut off, the pumping action of the main pistons, when running, draws smoke-box gases, dust, and ashes down the blast pipe, and solids adhere to the oily surfaces. Some decomposition of the oil also occurs. This action can be considerably lessened by (1) putting the reversing lever into full gear when steam is shut off; (2) by the provision of air valves to automatically admit air to the steam chests and destroy any vacuum created; (3) by automatic by-pass valves which put the opposite ends of the cylinder in communication when steam is shut off. Another practice is to admit a very small quantity of steam, either through the regulator or by means of a special valve. By filling the spaces with steam instead of air, decomposition of the oil is prevented.

Fire-tube Superheaters are principally employed. Several rows of large fire tubes are fitted in the boiler, and into these superheating tubes extend from a header or headers in the smoke-box towards the fire-box and back again. The construction of the header varies, as well as the arrangement of tubes, but the object in all designs is to direct steam through small tubes exposed to hot gases direct from the furnace and supplying an extended steam path, so that while travelling at a high velocity a considerable degree of heat is taken up by the steam.

The 'Swindon' (used only on the Western Region) design has its own constructional features, being designed for a moderate degree of superheat. The Schmidt design aims at a high superheat – from 250° to 530° F. above the temperature of saturated steam. It consists of a suitable header or steam collector, to which are attached four-fold elements. This provides a long steam path, and in virtue of the high steam velocity it is able to take up the required heat.

To obtain superheated steam of from 625° to 675°, the net gas area through flue tubes should be approximately 51 per cent. of the whole.

Most locomotives designed in recent years have relatively large superheating surface, as it has been found that a high degree of superheat is more economical as regards consumption of steam than a low degree, despite the fact that the exhaust steam from the blast-pipe still retains some degrees of superheat.

Some railways fit snifting or anti-vacuum valves on the top of the smoke-box. When an engine is drifting with steam shut off, air is drawn from the valves through the superheater elements, and prevents overheating. At the same time the air is heated before coming into contact with the valves, and thus reduces the risk of the valve liners becoming loose by contraction from sudden cooling.

Feed-water Heating.

For many years attempts have been made to recover some of the losses that occur in the locomotive by returning waste heat to the boiler through the medium of the feed water. To devise a method—or methods—however, of feeding continuously with water preheated to a degree high enough to render the means worth while, has taken years of development, research and experiment.

Locomotive feed heaters may be classified as:

1. Surface or closed type, incorporating in their construction water tubes deriving heat from either exhaust steam or from flue and smoke-box gases.
2. Direct contact, mixture or open type embodying either steam-jet pumps (injectors), steam-driven, direct-acting pumps and jet condenser chambers, or mechanically-driven pumps, comprising jet condensers and surface economisers in series.

It is generally recognised that if the temperature of the feed water be raised by heat from waste sources, a saving in fuel of 1 per cent. for every 11° F. rise will be obtained.

This may be expressed simply by:

$$S = \frac{100\,(t_2 - t_1)}{H + 32 - t_1}$$

where S = percentage of fuel saved:
t_1 = inlet feed temperature before heating in F°;
t_2 = outlet or final feed temperature in F°;
H = total heat of steam.

The principle of the tubular surface equipment has been in use at sea for many years, but one of the first successful applications to locomotives was made by Trevithick on the Egyptian State Railways, and the exhaustive experiments made are fully described in *Proc. Inst. Mech. Eng.*, 1913. This class of heater is probably the simplest in existence, and consists essentially of a nest or group of tubes through which passes the feed water from the pump delivery on its way to the boiler and around which tubes a quantity of exhaust steam from the blast-pipe is condensed. This heater is, in effect, a surface condenser open to atmosphere by means of the drain through which passes to waste or back to the tender the water of condensation.

Modifications of this arrangement involve the placing of the heater, usually of cylindrical form, across the smokebox forward of the chimney, and either attached by brackets to the shell or set in a recess formed by cutting away or shaping the smoke-box plating to suit. Both dispositions are widely used in the U.S.A. and in Germany, and this positioning of the heater has the advantages of an extremely short exhaust lead, and of providing a ready and simple means of returning by gravity the condensed exhaust steam to the tender tank, which is reached after the condensate has passed through an oil filter or separator. The tender capacity is thus increased by the amount of exhaust condensed.

An ingenious form of water raiser, for use on closed heaters situated below tank-top level, comprises a drain receiver fitted with a ball float and trip valve. The upper part of the tank is connected with the air-brake reservoir, and the system is so arranged that when a sufficient quantity of condensate is received from the heater drain the ball lifts, opens the trip valve, which admits high-pressure air to the tank, and the water is thus forced back into the tender.

The surface or closed type heater, while being undoubtedly simple, has the inherent disadvantage of being quickly rendered inoperative when used on engines working in districts with 'bad' water.

A large French railway determined the relative economies rendered by three classes of feed heaters as follows:

(*a*) *Fuel.*

Mixture heater	.	10 to 12 per cent. saving.
Surface heater	.	7 per cent. saving.
Exhaust Injector	.	4 to 5 per cent. saving.

(*b*) *Water.*

Mixture heater	.	15 per cent. saving.
Surface heater	.	Nil.
Exhaust injector	.	5 to 6 per cent. saving.

In the early days of locomotive engineering, before sufficient experience had been obtained with the effects of high temperatures on tubes containing impure water, much costly experimentation was made with tubular feed-water heaters fitted in the smoke-box. The high temperature obtaining in this compartment formed a very tempting bait to designers, and there is no doubt that given sufficient surface and the right materials the effect of the smoke-box gases can be a useful one. The difficulty is that, although a high temperature of 600° or 650° F. is normal to a smoke-box in full operation, its heat is what may be termed comparatively static, and it is difficult to impart this heat through a tube wall to the water.

One of the first contact or mixture heaters to be applied to locomotives was the exhaust steam injector which, by reason of its low first cost, light weight and small bulk, has enjoyed considerable favour. The exhaust injector is, perhaps, one of the best examples of finesse in modern steam locomotive practice, and while its returns in the way of fuel economies are not so high as those rendered by some of the more modern pump and heater systems, there would seem to be circumstances where it is considered of value, especially where weight is a vital question. It is much used on British railways. It appears to be the consensus of opinion of most locomotive engineers that an average fuel economy of 5 to 6 per cent. is as high as can reasonably be expected.

An early form of direct-contact heater was the tender tank itself, to which was led a branch from the engine blast-pipe. Owing to a number of reasons, the chief being the long lead of exhaust pipe, the unwieldy connections between engine and tender, undue heating of the water causing trouble with the injector, the arrangement did not find universal favour; it has, therefore, not become one of the recognised feed-water heaters. It may be further added that the maintaining of water at a temperature at which it would be acceptable to the injector in cases of pump failure, viz., 120° F., scarcely justified the means adopted.

Mixture or open type heaters are further subdivided into those using a mechanically-operated pump and those incorporating the steam-driven auxiliary.

An example of the former system is the Dabeg.

The Worthington arrangement, used in the U.S.A., employs a small turbine-driven centrifugal pump situated as near as possible to the tender, and which delivers water at constant speed to the heater. A practically constant, low-delivery pressure is thus maintained, and the pump, therefore, supplies as much water as is permitted by the opening of the control valve in the heater.

The feed having been pre-heated in the exhaust chamber, passes then to the feed pump proper, which is an ordinary direct-acting reciprocating pump.

Undoubtedly, one of the simplest mixture systems so far evolved is the A.C.F.I., employing a pair of cylindrical chambers or drums placed in parallel across the top of the boiler and combined with a small direct-acting steam-operated feed pump of the horizontal pattern. It is much used on French railways, and the L.N.E.R. have a number of engines fitted with it.

Water from the tender is drawn through a fine mesh strainer contained in the feed well by a cold water pump, which discharges it through a spray pipe into the mixing chamber. Here exhaust steam from the blast-pipe is met with, having just previously traversed an oil separator, and the mixture passes over through an inter-connecting pipe to the settling chamber, where dissolved oxygen and CO_2 are liberated, which gases are discharged through a vent in the top of the chamber. The feed, now heated practically to exhaust temperature, reaches the suction of the hot pump, and is delivered to the boiler. Any excess hot water collecting in the settling compartment is returned to the feed well.

(Abstracts of paper read before Inst. Loco. Eng.)

Injectors.

TABLE SHOWING MAXIMUM DELIVERY OF COMBINATION INJECTORS.

(Gallons per hour working on dry steam, with feed water at 60° F., lifting 3 ft. and delivering against pressure equal to steam at back of nozzle.)

Size.	Boiler Pressure in lbs. per Square Inch.							Size of Pipes.			
	60	90	120	150	180	210	225	Steam.	Water.	Out-flow.	Delivery.
								Ins.	Ins.	Ins.	Ins.
6	550	670	780	870	840	800	790	1	1¼	1¼	1¼
7	750	920	1,070	1,190	1,140	1,090	1,080	1	1⅜	1⅜	1⅜
8	980	1,200	1,400	1,550	1,490	1,420	1,410	1¼	1½	1½	1½
9	1,240	1,520	1,770	1,970	1,880	1,800	1,790	1¼	1¾	1¾	1¾
10	1,530	1,880	2,180	2,430	2.330	2,220	2,200	1½	1⅞	1⅞	1⅞
11	1,850	2,270	2,650	2,940	2,820	2,700	2,670	1½	1⅞	1⅞	1⅞
12	2,200	2,700	3,150	3,500	3,350	3,200	3,150	2	2	2¼	2¼

(J. N. Gresham, 'Jour. of Inst. of Loco-Engineers.')

BRITISH STANDARD SPECIFICATIONS FOR MATERIAL USED IN THE CONSTRUCTION OF RAILWAY ROLLING STOCK.

(Report No. 24—Parts I.–VI.). (*Abstracts.**)

LOCOMOTIVE CRANK AXLES—B.S.S. 1—1941 (of Report No. 24, Part I.).

The crank axles shall be manufactured from high quality steel made by the acid open-hearth or electric process, and shall not show on analysis more than 0·05 per cent. of sulphur or of phosphorus.

A standard test-piece C or a subsidiary test-piece 0·798 in. diameter shall show an ultimate tensile stress of not less than 28 tons nor more than 33 tons (62,720 to 73,920 lb.) per sq. in. (44·10 to 52 kg. per mm.²), with an elongation of not less than 25 per cent. The yield stress in either case shall be not less than 50 per cent. of the ultimate tensile stress.

A test-piece 9 ins. (230 mm.) long, 1¼ in. (32 mm.) square, with $\frac{1}{16}$ in. (1·6 mm.) radius at the edges, machined from the crank axles, shall withstand being bent cold by direct pressure from a tool 2 in. (51 mm.) wide, the end having a radius of 1 in. (25 mm.), until the sides of the test-piece are parallel. The ends of the test-piece are then to be brought together, without fracture, by direct pressure.

As an alternative to the cold bend test, the test-piece may be placed upon bearings having a clear span of 6 in. (152 mm.), resting on a solid foundation, and shall withstand, without fracture, 20 blows from a weight of 1,120 lb. (508 kg.), having a rounded end of 1¼ in. (32 mm.) radius, falling 6 in. (152 mm.). The test-piece shall be reversed after the first and every alternate blow. The fall shall then be increased to 12 in. (305 mm.), and the test continued until fracture occurs.

All crank axles shall be oil-hardened, and tempered.

LOCOMOTIVE STRAIGHT AXLES—B.S.S. 2—1941 (of Report No. 24, Part I.).

Quality of material same as for locomotive crank axles. Use of basic open-hearth stee permitted.

The axle shall be placed upon bearings resting on a block of metal of not less than 5 tons (11,200 lb. = 5,080 kg.) weight supported on a concrete or other solid foundation, and shall withstand, without fracture, five blows from a weight of 1 ton (2,240 lb. = 1,016 kg.) falling 16 to 35 ft., according as the diameter is under 4 or over 6 in., the distance apart of the bearings being from 3 to 5 ft., as shown by table.

The axle shall be turned after the first and third blow, and shall be broken after testing, both in the centre and at the journals.

A standard test-piece C or a subsidiary test-piece 0·798 in. diameter shall show an ultimate tensile stress of not less than 35 nor more than 40 tons (78,400 to 89,600 lb.) per sq. in. (55·1 to 63 kg. per mm.²), with an elongation of not less than 25 per cent. with 35 tons and 20 per cent. with 40 tons; the sums of the intermediate tensile breaking strengths and corresponding elongations being not less than 60. The yield stress shall be not less than 50 per cent. of the ultimate tensile stress.

A test-piece 9 in. (230 mm.) long and 1¼ in. (32 mm.) sq., with $\frac{1}{16}$ in. (1·6 mm.) radius at the edges, shall withstand being bent cold by direct pressure from a tool 2 in. (51 mm.) wide, the end having a radius of 1 in. (25 mm.), until the sides of the test-piece are parallel. The ends of the test-piece shall then be brought together, without fracture, by direct pressure.

N.B.—This test shall be taken only when a less number than 15 axles is ordered and the falling weight test has not been carried out.

All axles shall be either normalised or oil-treated.

LOCOMOTIVE STRAIGHT AXLES—B.S.S. 2A—1941 (of Report No. 24, Part I.).

The axles shall be manufactured from high quality steel made by the acid or basic open-hearth, acid Bessemer or electric process. In the event of the basic open-hearth process being employed, the maximum percentage of sulphur or of phosphorus shall not exceed 0·06 per cent.

Tests and treatment as for B.S.S. 2.

CARRIAGE AND WAGON AXLES—B.S.S.3—1941 (of Report No. 24, Part I.).

Same as for locomotive straight axles, B.S.S. 2, except that sulphur or phosphorus shall not exceed 0·06 per cent. and that range of falling weight test is from 15 to 35 ft. for axles under 3¾ in. to over 6 ins.; distance between bearings, 3 to 5 ft.

CARRIAGE AND WAGON AXLES—B.S.S. 3A—1941 (of Report No. 24, Part I.).

Same as for locomotive straight axles, B.S.S. 2A, except that range of falling weight test, etc., is as specified in B.S.S. 3.

LOCOMOTIVE TYRES—B.S.S 4—1942 (of Report No. 24, Part II.).

Quality of material same as for locomotive axles—B.S.S. 2.

* **By permission of the British Standards Institution.**

The tyre shall be placed in a running position with the tread resting on a block of metal of not less than 5 tons weight supported on a concrete or other solid foundation, and shall withstand, without fracture, blows from a falling weight of 1 ton. The weight shall be allowed to fall freely on to the tread from heights of 10 ft., 15 ft., 20 ft., and upwards, until the deflection of the tyre corresponds to that given in the following formula, in which d is the internal diameter of the tyre as rolled, in inches, and t is the thickness of the centre of the tread, as rolled, in inches, and s is the depth of the snip in inches.

Class C.—Tensile Breaking Strength 50 to 55 tons per square inch.

$$\frac{(d-3s)^2}{50t^2}$$

Class D.—Tensile Breaking Strength 56 to 62 tons per square inch.

$$\frac{(d-3s)^2}{55t^2}$$

Class E.—Tensile Breaking Strength 63 to 69 tons per square inch.

$$\frac{(d-3s)^2}{60t^2}$$

A standard test-piece C taken from the position indicated shall show the ultimate tensile stress and minimum elongations given in the table, the intermediate elongations being in proportion.

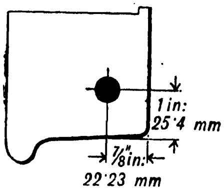


FIG. 6.

Class.	Tensile Strength. Tons per sq. in.	Minimum Elongation. Per cent
C	50–56	13 to 11
D	56–62	11 to 9
E	63–69	10 to 8

When so specified in the inquiry, each tyre shall be allowed to drop freely in a running position, from the height specified below, upon a rail fastened to an iron block of not less than 2 tons in weight. Each tyre shall then be turned round through an angle of 90 degrees, and dropped a second time.

Internal Diameter of Tyre.	Up to 3½ ft.	Over 3½ ft. and up to 4½ ft.	Over 4½ ft. and up to 5½ ft.	Over 5½ ft. and up to 6½ ft.	Over 6½ ft.
	ft.	ft.	ft.	ft.	ft.
Height of fall	5	4	3½	3	2½

CARRIAGE AND WAGON TYRES—B.S.S. 5—1942 (of Report No. 24, Part II.).

The specification is the same as for locomotive tyres, with the addition of Class B tyres, tensile breaking strength 42 to 48 tons per sq. in. and minimum elongation 18 to 15 per cent. Basic open-hearth process permitted, and limit of sulphur and phosphorus raised to 0·06 per cent for Classes B, C and D. For Class E tyres it is 0·05 per cent. Formula for deflection, Class B— $\frac{(d-3s)^2}{45t^2}$. Height of fall for diameters over 3½ ft. is 4 ft., and up to 3½ ft. it is 5 ft.

CARRIAGE AND WAGON TYRES—B.S.S. 5A—1942 (of Report No. 24, Part I.).

Material as for axles B.S.S. 2A. Class E omitted.

CARBON STEEL LAMINATED SPRINGS—B.S.S. 6X—1942 (of Report No. 24, Part III.).
With Analysis.

The spring plates shall be manufactured from high quality steel made from the best selected material by the acid or basic open-hearth or electric process, and shall show on analysis not more than 0·05 per cent. of sulphur or of phosphorus, nor more than 0·9 per cent. nor less than 0·65 per cent. of carbon in the case of plates which are to be oil hardened, nor more than 0·65 per cent. nor less than 0·45 per cent. of carbon in the case of plates which are to be water hardened. The engineer (or purchaser) may, however, specify the range of carbon which he requires, within the limits given above, when placing the order.

All springs shall be tested by being deflected by a quick-acting scrag before the buckle is put on, to an amount equal to $\frac{L^2}{900t}$, where L is the length of the top plate in inches measured along the arc as shown in fig. 7, and t is the thickness of the thickest plate in inches. They must then stand being deflected again three times in quick succession without showing any permanent set. The required deflections are shown in the table and must not be exceeded at any time during manufacture by more than 15 per cent.

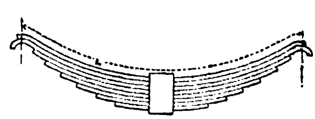


FIG. 7.

All plates shall be carefully hardened and tempered.

The buckles shall be made of good fibrous iron having a tensile breaking strength of from 20 to 24 tons per sq. in. or of suitable steel having a tensile breaking strength of not less than 24 nor more than 30 tons per sq. in. The elongation in each case shall not be less

L	Required Deflection under Scrag, in Inches.				
In.	$\frac{3}{8}$ in. plate.	$\frac{7}{16}$ in. plate.	$\frac{1}{2}$ in. plate.	$\frac{9}{16}$ in. plate.	$\frac{5}{8}$ in. plate.
90	24	$20\frac{5}{8}$	18	16	$14\frac{3}{8}$
84	$20\frac{7}{8}$	$17\frac{7}{8}$	$15\frac{5}{8}$	14	$12\frac{1}{2}$
78	18	$15\frac{1}{2}$	$13\frac{1}{2}$	12	$10\frac{7}{8}$
72	$15\frac{3}{8}$	$13\frac{1}{8}$	$11\frac{1}{2}$	$10\frac{1}{4}$	$9\frac{1}{4}$
66	$12\frac{7}{8}$	11	$9\frac{5}{8}$	$8\frac{5}{8}$	$7\frac{3}{4}$
60	$10\frac{5}{8}$	$9\frac{1}{8}$	8	$7\frac{1}{8}$	$6\frac{3}{8}$
54	$8\frac{3}{8}$	$7\frac{3}{8}$	$6\frac{1}{2}$	$5\frac{3}{4}$	$5\frac{1}{8}$
48	$6\frac{7}{8}$	$5\frac{7}{8}$	$5\frac{1}{8}$	$4\frac{1}{2}$	$4\frac{1}{8}$
42	$5\frac{1}{4}$	$4\frac{1}{2}$	$3\frac{7}{8}$	$3\frac{1}{2}$	$3\frac{1}{8}$
36	$3\frac{7}{8}$	$3\frac{1}{4}$	$2\frac{7}{8}$	$2\frac{1}{2}$	$2\frac{1}{4}$
30	$2\frac{5}{8}$	$2\frac{1}{4}$	2	$1\frac{3}{4}$	$1\frac{5}{8}$

than 20 per cent. (standard test-piece A), and in the case of buckles made from the solid, the elongation (standard test-piece C or subsidiary test-piece 0·798 in. diameter) shall not be less than 25 per cent., and the material shall admit of bending, when cold without showing crack or flaw, as follows :—

For iron, $\frac{3}{4}$-inch thick through an angle of 120 degrees.
" $\frac{5}{8}$-inch " " 130 "
" $\frac{1}{2}$-inch " " 140 "
For steel, all thicknesses, through an angle of 180 degrees.

In all cases the inside radius of the bend shall be not greater than $1\frac{1}{2}$ times the thickness of he test-piece.

SILICO-MANGANESE STEEL LAMINATED SPRINGS—B.S.S. 6Y—1942 (of Report No. 24, Part III.).

The spring plates shall be manufactured from high quality steel made by the acid open-hearth or the electric process and shall show on analysis the following composition :—

Quality.	Carbon. Per cent.	Silicon. Per cent.	Manganese. Per cent.	Sulphur. Per cent.	Phosphorus. Per cent.
Oil-hardening . .	0·50 to 0·60	1·50 to 2·0	0·60 to 1·0	0·05 max.	0·05 max.
Water-hardening .	0·33 to 0·50	1·50 to 2·0	0·60 to 1·0	0·05 max.	0·05 max.

The engineer (or purchaser) must specify whether he requires oil- or water-hardening.
Scragging test as for B.S.S. 6X.
All plates shall be carefully hardened and tempered.
Buckles as for B.S.S. 6X.

CARBON STEEL LAMINATED SPRINGS—B.S.S. 6A—1942 (of Report No. 24, Part III.).

Without Analysis.

Same as specification B.S.S. 6X, except that spring plates shall be manufactured from steel made from selected material by the acid or basic open-hearth, the acid Bessemer, or electric process. In the event of the basic open-hearth process being employed, the maximum percentage of sulphur and phosphorus must not exceed 0·05 per cent. The manufacturer shall supply a certificate of the carbon determination when required to do so.

CARBON SPRING STEEL FOR LAMINATED SPRINGS—B.S.S. 6B—1942 (of Report No. 24, Part III.).

With Analysis.

The spring bars shall be manufactured from high quality steel made from the best selected material by the acid open-hearth or electric process, and shall show on analysis not more than 0·05 per cent. of sulphur or of phosphorus, nor more than 0·9 per cent. nor less than 0·65 per cent. of carbon in the case of material which is to be oil hardened, nor more than 0·65 per cent. nor less than 0·45 per cent. of carbon in the case of material which is to be water hardened. The

engineer (or purchaser) may, however, specify the range of carbon which he requires, within the limits given above, when placing the order.

The test-piece shall be 60 T long, where T is the thickness of the steel and shall be cambered to a radius of 80 T, hardened and tempered. The required camber should then be 5·5 T. Any adjustment of camber shall be made at tempering heat. The test-piece shall be pressed straight between straight parallel surfaces and the camber noted after release which shall not be less than 5 T. The test-piece shall then be pressed straight again six times in quick succession without showing any permanent set.

	In.	In.	In.	In.	In.	In.	In.
Thickness of bar T . . . . Length of test-piece 60 T . .	$\frac{1}{4}$ 15	$\frac{5}{16}$ $18\frac{3}{4}$	$\frac{3}{8}$ $22\frac{1}{2}$	$\frac{7}{16}$ $26\frac{1}{4}$	$\frac{1}{2}$ 30	$\frac{9}{16}$ $33\frac{3}{4}$	$\frac{5}{8}$ $37\frac{1}{2}$
Initial camber 5·5 T . . .	$1\frac{3}{8}$	$1\frac{3}{4}$	$2\frac{1}{16}$	$2\frac{7}{16}$	$2\frac{3}{4}$	$3\frac{1}{8}$	$3\frac{7}{16}$
Minimum camber after first blow and next six consecutive blows, 5 T	$1\frac{1}{4}$	$1\frac{9}{16}$	$1\frac{7}{8}$	$2\frac{3}{16}$	$2\frac{1}{2}$	$2\frac{13}{16}$	$3\frac{1}{8}$

CARBON SPRING STEEL FOR LAMINATED SPRINGS—B.S.S. 6C—1942 (or Report No. 24, Part III.). Without Analysis.

Material for spring bars same as B.S.S. 6A. Tests same as B.S.S. 6B.

SILICO-MANGANESE SPRING STEEL FOR LAMINATED SPRINGS—
B.S.S. 6D—1942 (of Report No. 24, Part III.).

Material for spring bars same as B.S.S. 6Y. Tests same as for B.S.S. 6B.

CARBON STEEL VOLUTE AND HELICAL SPRINGS—B.S.S. 7X—1942 (of Report No. 24, Part III.).

The springs shall be manufactured from high quality steel made by the acid open-hearth or the electric process and shall show on analysis the following composition:—

	Not less than	Not more than
Carbon . . .	0·9 per cent.	1·2 per cent.
Manganese . .	0·45 „	0·70 „
Silicon . . .	—	0·30 „
Sulphur . . .	—	0·05 „
Phosphorus . .	—	0·05 „

Each spring shall be tested by being pressed home three times by a quick-acting scrag, the free height shall then not exceed that specified by more than $\frac{1}{4}$-inch or $1\frac{1}{2}$ per cent., whichever is the greater, and it shall not be less than that specified by more than one half of this tolerance.

Up to 5 per cent. of the springs may also be tested under varying loads to determine the range and deflection per ton.

All springs shall be hardened in oil and suitably tempered.

CARBON STEEL VOLUTE AND HELICAL SPRINGS—B.S.S. 7A—1942 (of Report No. 24, Part III.).

The springs shall be manufactured from the highest quality of steel made from the best selected material by the acid open-hearth, the acid Bessemer, or the electric process, and shall not contain more than 1·2 per cent. nor less than 0·9 per cent. of carbon.

Test and treatment same as B.S.S. 7X.

SILICO-MANGANESE STEEL VOLUTE AND HELICAL SPRINGS—
B.S.S. 7Y—1942 (of Report No. 24, Part III.).

The springs shall be manufactured from high quality steel made by the acid open-hearth or the electric process and shall show on analysis the following composition:—

	Not less than	Not more than
Carbon . . .	0·50 per cent.	0·60 per cent.
Silicon . . .	1·50 „	2·00 „
Manganese . .	0·60 „	1·00 „
Sulphur . . .	—	0·05 „
Phosphorus . .	—	0·05 „

Test and treatment same as for B.S.S. 7X.

CARBON SPRING STEEL FOR VOLUTE AND HELICAL SPRINGS—
B.S.S. 7B—1942 (of Report No. 24, Part III.).

Material same as B.S.S. 7X above.

SILICO-MANGANESE SPRING STEEL FOR VOLUTE AND HELICAL SPRINGS—B.S.S. 7C—1942 (of Report No. 24, Part III.).

Material for spring steel same as B.S.S. 7Y.

STEEL FORGINGS AND CLASS 'C' AND CLASS 'D' ROLLED BARS FOR LOCOMOTIVES—B.S.S. 8—1941 (of Report No. 24, Part IV.).

The forgings and Class 'C' and Class 'D' rolled bars shall be manufactured from steel made as given below.

The tensile breaking strength and minimum elongation on standard test-piece C or subsidiary test-piece 0·798 in. diameter to be as follows :—

Class.	Description of Forging.	Process of Manufacture.	Maximum Manganese Content per cent.	Maximum Sulphur or Phosphorus Content per cent.	Tensile Breaking Strength in tons per sq. in.	Elongation Minimum per cent.
A	Special forgings which will be case-hardened	Acid or basic open-hearth, Acid Bessemer or electric	0·70	0·05	24 to 28	29 to 25
B	Ordinary and boiler forgings	Do.	—	0·05	26 to 32	28 to 22
C	Special forgings without wearing surfaces, and bars	Acid open-hearth or electric	—	0·05	23* to 38	26 to 20
D	Special forgings with wearing surfaces and bars	Do.	—	0·05	40* to 45	20 to 15

The sums of the tensile breaking strengths and corresponding elongations must be not less than 53 for Classes A, 54 for Class B, 58 for Class C, 60 for Class D. In the case of Classes C and D the yield stress shall be not less than 50 per cent. of the ultimate tensile strength.

A test-piece, 9 inches long and 1¼ inch square, or as near this size as the forging or forgings selected will permit, with 1/16 inch radius at the edges, machined cold, must, without any reheating, withstand being bent cold by direct pressure from a tool 1½ in. wide in the case of Classes A, and B, 1¾ in. wide in the case of Class C, and 2½ in. wide in the case of Class D, the ends having a radius respectively of ¾ in., ⅞ in., and 1¼ in., until the sides of the test-pieces are parallel. The ends of the test-pieces of all classes of forgings, except Class D, shall then be brought together without fracture by direct pressure.

Forgings in Classes A and B may be, and those in Classes C and D shall be, either normalised or otherwise heat treated.

STEEL BLOOMS FOR LOCOMOTIVE FORGINGS—B.S.S. 9—1941 (of Report No. 24, Part IV.).

Specification and tests for Classes A, B, C, and D same as for B.S.S. 8.

One bloom selected by the representative of the engineer (or of the purchaser) from each cast shall be tested at the works of the manufacturer, as follows :—

Blooms for Classes A and B forgings shall comply with the tests without any forging down or heat treatment.

For blooms Classes C and D forgings a portion of full section shall be cut and normalised or a portion may be forged down to not less than 5 in. diameter or thickness and then normalised before testing.

STEEL CASTINGS—B.S.S. 10—1941 (of Report No. 24, Part IV.).

The castings shall be manufactured from steel produced by such process as may be agreed between the manufacturer and the customer, and shall show on analysis not more than 0·06 per cent. of sulphur or of phosphorus.

All castings shall be heat-treated by heating to a uniform temperature not less than the normalising temperature and allowing to cool slowly from the maximum temperature in a practically uniform manner, or alternatively, normalised by heating in a similar manner and allowing to cool in air.

* Yield stress shall not be less than 50 per cent. of the ultimate tensile stress.

The minimum tensile breaking strength and elongation on standard test-piece C or subsidiary test-piece 0·798 in. diameter to be as follows:—

Grade.	Description of Casting	Minimum Tensile Breaking Strength per sq. in.	Minimum Elongation per cent.
1	Castings with wearing surfaces	35	15
2	Other General Castings and Wheel Centres . .	26	20

In the case of wheel centres the yield stress shall be determined and shall be not less than 13 tons per sq. in.

Cold bend tests shall be made upon machined test-pieces 9 in. long and having either a diameter of 1 in. or a rectangular section of 1 in. wide by ¾ in. thick. In the case of rectangular test-pieces the edges shall be rounded to a radius of $\frac{1}{16}$ in., and the test shall be made by bending the test-piece over the thinnest section.

Bend tests may be made by pressure or by blows and the test-piece shall without fracture withstand being bent cold round a former having a radius of 1 in. through an angle not less than 60° for castings of Grade 1 and 90° in the case of Grade 2.

Each wheel centre shall be suspended in a suitable position and struck with a 4 lb. hammer on the rim and spokes (where such exist) and an examination made for any signs of blowholes or other defects.

When wheel centres are tested to destruction, the test shall be made by dropping the wheel centre on to a block of metal of not less than 5 tons weight, supported upon a rigid concrete or other solid foundation. The wheel centre shall be raised and allowed to fall freely in a running position through distances of 5 ft. and 10 ft. on the end of a spoke (where such exists), and then turned through 90 degrees and again raised and allowed to fall freely in a running position through distances of 5 ft. and 10 ft. on the end of a spoke (where such exists). Coupled wheels having balance weights shall be dropped with the weights at the bottom. Any sign of failure under this test will render the wheel centres liable to rejection. The test shall then be continued, the fall being increased 5 ft. each time until fracture results or the wheel centre is doubled up.

Copper Plates for Locomotive Fire-boxes—
B.S.S. 11—1943 (of Report No. 24, Part V.).

The plates shall contain not less than 99·20 per cent. of copper, and shall contain not less than 0·30 per cent. nor more than 0·50 per cent. of arsenic. Antimony shall not exceed 0·05 per cent. Bismuth shall not exceed 0·01 per cent.

Tensile breaking strength to be not less than 14 tons per square inch, with an elongation of not less than 35 per cent. on standard test-piece A.

Pieces of the plate shall be tested both cold and at a red heat by the ends being bent through an angle of 180 degrees in opposite directions and doubled up close, without showing either crack or flaw on the outside of either bend.

Rolled Copper Rods for Locomotive Stay Bolts, Rivets, etc.—
B.S.S. 12—1943 (of Report No. 24, Part V.).

The rods shall contain not less than 99·20 per cent. of copper, and shall contain not less than 0·30 per cent. nor more than 0·50 per cent. of arsenic. Antimony shall not exceed 0·05 per cent. Bismuth shall not exceed 0·01 per cent.

Tensile strength to be not less than 14·50 tons per square inch, with an elongation of not less than 40 per cent. on standard test-piece B. For rods up to 1¼ inch diameter, the test shall be made upon the unturned rod. For rods above 1¼ inch diameter, the rod may be turned down, when a subsidiary standard test-piece 0·798 in. or 0·977 in. in diameter shall be used, in which case the elongation shall be not less than 45 per cent.

The rod shall, both when cold and at a red heat, withstand, without machining, being doubled up close without cracking.

A piece of rod 1 inch long shall be placed on end and hammered or crushed down when cold to a thickness of ⅜ inch without showing either crack or flaw on the circumference of the resulting disc.

Extruded Copper Rods for Locomotive Stay Bolts, Rivets, etc.—
B.S.S. 12a—1943 (of Report No. 24, Part V.).

Tests the same as for B.S.S. 12.

Copper Tubes for Locomotive Boilers—
B.S.S. 13—1943 (of Report No. 24, Part V.).

The tubes shall contain not less than 99·20 per cent. of copper, and shall contain not less than 0·30 per cent. nor more than 0·50 per cent. of arsenic. Antimony shall not exceed 0·05 per cent. Bismuth shall not exceed 0·01 per cent.

Tensile tests shall be made on pieces of tube or strips cut from the tube. If the test is made on a piece of tube, the tensile strength shall be not less than 14·50 tons per square inch, with an elongation of not less than 50 per cent. on 2 inches. If the test is made on a strip, the tensile strength shall be not less than 14 tons per square inch, with an elongation of not less than 40 per cent. on a test piece having a gauge length of four times the square root of the area.

The test-piece shall stand bulging or drifting cold without showing either crack or flaw until the outside diameter of the bulged or drifted end measures not less than 25 per cent. more than the original diameter of the tube.

The test-piece shall stand flanging cold without showing either crack or flaw until the diameter of the flange measures not less than 40 per cent. more than the original diameter of the tube.

The test-piece shall stand the following test, both cold and at a red heat, without showing either crack or flaw. The test-piece shall be flattened down until the interior surfaces meet, and be doubled on itself, that is, bent through an angle of 180 degrees, the bend being at right angles to the direction of the length of the tube.

Each tube shall be tested by an internal hydraulic pressure of 750 lb. per square inch, or by such internal pressure as may have been specified.

BRASS TUBES FOR LOCOMOTIVE BOILERS—
B.S.S. 14—1943 (of Report No. 24, Part V.).

Brass for locomotive tubes may be of either 70/30 alloy or 2/1 alloy of copper and zinc.

The tubes shall consist of an alloy of copper and zinc, and shall contain :—

70/30 Alloy.—Not less than 70 per cent. of metallic copper and not more than a total of 0·75 per cent. of materials other than copper and zinc.

2/1 Alloy.—Not less than 66·70 per cent. of metallic copper and not more than a total of 0·75 per cent. of materials other than copper and zinc.

The tubes shall be carefully annealed at both ends.

The test-piece shall stand bulging or drifting cold without showing either crack or flaw until the diameter of the bulged or drifted end measures not less than 25 per cent. more than the original diameter of the tube.

The test-piece shall stand flanging cold without showing crack or flaw until the diameter of the flange measures not less than 25 per cent. more than the original diameter of the tube.

The test-piece when cold shall stand the flattening and doubling-over test applied to copper tubes, B.S.S. 13. Tubes to be tested by an internal hydraulic pressure as may have been specified.

WELDLESS STEEL TUBES FOR LOCOMOTIVE BOILERS.*
(B.S.S. No. 53—1927.)

The tubes shall be cold-drawn and weldless, and shall be manufactured from steel of the best quality made by the open-hearth process. They shall not show on analysis more than 0·035 per cent. of sulphur or more than 0·030 per cent. of phosphorus.

The tubes or strips cut from the tubes shall, without further annealing, show a tensile strength of not less than 20 tons per sq. in., nor more than 26 tons per sq. in., with an elongation of not less than 28 per cent. in 8 in. for tubes, and not less than 22 per cent. in 8 in. for strips cut from the

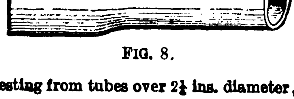


FIG. 8.

tubes. If strips are cut for testing from tubes over 2½ ins. diameter, they may be annealed before testing.

The tubes shall, when cold, stand bulging with a parallel drift (see fig. 8) without showing either crack or flaw, until the diameter of the bulged end exceeds the original diameter of the tube by not less than 15 per cent. for tubes under 11 s.w.g. in thickness and 12½ per cent for tubes from 11 s.w.g. to 6 s.w.g. inclusive.

A piece of tube 2 in. long shall be placed on end and shall, when cold, stand hammering down until it is reduced to 1¼ in. high without showing either crack or flaw.

Tubes up to 2½ in. external diameter and 10 S.W.G. shall be flattened by pressure between two parallel flat surfaces, the width of which shall be not less than three times the diameter of the tube, until the interior surfaces meet at the middle and remain in contact when the pressure is released.

Other tubes shall be flattened in a similar manner until the interior surfaces are, at the middle, brought out to a distance apart equal to the thickness of the wall of the tube.

Every tube shall be tested by an internal pressure of at least 1,000 lb. per sq. in.

*** Pending revision of this Specification B.S.S. 494 is at present quoted. Limits for sulphur and phosphorus 0·05 per cent.**

SEAMLESS COPPER PIPES FOR LOCOMOTIVES—

B.S.S. 15—1943 (of Report No. 24, Part V.).

The pipes shall contain not less than 99·20 per cent. of copper, and shall contain not less than 0·30 per cent. nor more than 0·50 per cent. of arsenic. Antimony shall not exceed 0·05 per cent. Bismuth shall not exceed 0·01 per cent.

Tensile tests shall be made on pieces of pipe or strips cut from the pipe. If the tensile test is made on a piece of pipe, the tensile strength shall be not less than 14·50 tons per square inch, with an elongation of not less than 50 per cent. on 2 inches for pipes of cross-section greater than 0·3 sq. in. and 50 per cent. on a gauge length of 1 in. for pipes of 0·3 sq. in. or less cross-sectional area.

If the tensile test is made on a strip cut from the pipe, the tensile strength shall be not less than 14 tons per square inch, with an elongation of not less than 40 per cent. on a test-piece having a gauge length of four times the square root of the area.

The test piece shall stand drifting cold without showing either crack or flaw until the diameter of the drifted end measures not less than 25 per cent. more than the original diameter of the pipe.

Flattening and doubling-over test the same as for copper tubes, B.S.S. 13.

Tubes to be tested by such internal hydraulic pressure as may have been specified.

STEEL PLATES (INCLUDING FIREBOX PLATES), SECTIONS, BARS AND RIVETS FOR LOCOMOTIVE BOILERS *—B.S.S. 16—1942 (of Report No. 24, Part VI.).

The plates, sections and bars shall be manufactured from steel made from selected material by the acid or basic open-hearth or the electric processes. The steel must not show on analysis more than 0·05 per cent. of sulphur or of phosphorus.

In the case of plates for fireboxes the steel must be made from the acid open-hearth or electric process and sulphur and phosphorus must not exceed 0·03 per cent.

When plates are required for flanging purposes it shall be so stated at the time of inquiry.

Test-pieces shall be cut or sheared crosswise from rolled material in case of plates, and lengthwise in case of sectional material. When material is annealed or otherwise treated, the test-pieces shall be similarly and simultaneously treated with the material before testing, except in the case of drawn steel wire when the test-pieces only shall always be annealed before testing. Any straightening of test-pieces shall be done cold.

Material.	Tensile Strength. Tons per sq. in.	Minimum Elongation. Per cent. On Test-Piece A.	On Test-Piece B.	On Test-Piece B.1.
Plates (other than firebox).	25–30	23	—	—
Firebox plates †	23–28	25	—	—
Sections and flats	28–33	20	—	—
Round and square bars (up to and including 1-in. diameter or thick)	28–33	—	20	—
Round and square bars (above 1-in. diameter or thick)	28–33	—	—	24
Rivet bars	25–30	—	26	30
Cold drawn material for rivets below ⅞-in. diameter	20–25	—	26	—

Test-pieces for the cold-bend test shall be cut or sheared crosswise from plates and lengthwise from sections and bars, and shall be not less than 1½ in. wide, but for small material the whole section may be used. In all bend tests the rough edges of the test-pieces may be removed by filing, grinding or machining. The edges of the test-piece may be ground or draw filed to remove sharpness and roughness.

* This Specification also applies to drawn steel wire under ⅞ in. diameter for the manufacture of cold forged rivets.

† Firebox Plates shall have a yield stress of not less than 55 per cent. of the ultimate tensile stress.

The test-pieces shall not be annealed or otherwise heat-treated unless the material from which it is cut is similarly annealed or heat treated, in which case the test pieces shall be similarly and simultaneously heat-treated before testing, except in the case of drawn steel wire, when the test-pieces only shall be annealed before testing.

The test-pieces shall withstand without fracture the bend test described in the following table.

The bend tests may be made either by pressure or by blows.

Material.	Max. Internal Radius of Bend.	Description of Bend Test.
Firebox plates	The thickness of test-piece	Doubled over until sides are parallel
Plates (other than firebox plates)	Twice the thickness of the test-piece	Do.
Sections and bars	Twice the thickness of the test-piece	Do.
Rivet bars	—	Doubled over and flattened on itself.

Tests for Manufactured Rivets.—The rivet shanks shall be bent cold, and hammered until the two parts of the shank touch, without fracture, on the outside of the bend. The rivet heads shall be flattened while hot without cracking at the edges until the diameter of the head is 2½ times the diameter of the shank.

Dump Test for Rivet Bars.—Short lengths equal to twice the diameter cut from the rivet bars shall, when cold, withstand without fracture being compressed to half their length.

Reverse Torsion Test for Rivet Bars.—Alternatively for material under ⅜-in. diameter the reverse torsion test may, at the option of the manufacturer, be substituted for the dump test. A length of material 8 in. between grips of the machine shall withstand without showing splits or frills being axially twisted four complete turns in one direction, then from this position four complete turns in the reverse direction.

STEEL PLATES, SECTIONS, BARS, AND RIVETS FOR LOCOMOTIVES*—

B.S.S. 17—1942 (of Report No. 24, Part VI.). With Analysis.

The plates (other than locomotive frame plates), sections, and bars shall be manufactured from steel made from selected material by the acid or basic open-hearth, acid Bessemer, or the electric process. Locomotive frame plates shall be made by either the acid or basic open-hearth process only.

The material shall, on analysis, contain not more than 0·06 per cent. of sulphur or phosphorus.

The tensile test-pieces shall be cut or sheared lengthwise or crosswise from the rolled material in the case of plates, and lengthwise in the case of sections and bars. The edges of the flat test-pieces shall be ground or draw filed to remove roughness and sharpness.

The test-pieces shall not be annealed or otherwise heat treated unless the material from which they are cut is similarly annealed or heat treated. In the case of drawn steel wire the test-pieces shall always be annealed before testing. For all material, other than rivet bars, under $\frac{1}{16}$-in. thick, bend tests only are required.

Cold-Bend Tests.—Test-pieces shall be cut or sheared crosswise from plates and lengthwise from sections and bars, and shall be not less than 1½ in. wide, but for small material the whole section may be used. In all bend-tests the rough edges of the test-pieces may be removed by filing, grinding or machining. The edges of the test-pieces may be ground or draw filed to remove sharpness and roughness.

The test-pieces shall not be annealed or otherwise heat-treated unless the material from which they are cut is similarly annealed or heat treated, in which case the test-pieces shall be similarly and simultaneously heat treated before testing.

For small sections these bend tests may be made from the flattened section.

Bend tests may be made by pressure or by blows.

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* When material is required for cold flanging, cold pressing, welding or cold working, this shall be so stated at the time of the enquiry or order. This specification shall apply also to drawn steel wire under ⅜-in. diameter for the manufacture of cold forged rivets.

TABLE OF BEND TESTS.

Material.	Dimensions.	Max. Internal Radius of Bend.	Description of Bend Test.
Plates—for other than cold-flanging or cold-pressing	All thicknesses	3 times the thickness of test-piece	Doubled over until sides are parallel
Do. for cold-flanging or cold-pressing	$\frac{5}{16}$ in. thick and above	Twice the thickness of test-piece	Do. Do.
Do. Do.	Below $\frac{5}{16}$ in. thick	—	Doubled over and flattened on itself
Sections, flats, and bars—for other than welding or cold-working	All thicknesses or diameters	3 times the diameter or thickness of test-piece	Doubled over until sides are parallel
Do. for welding or cold-working	$\frac{5}{16}$ in. in diameter or thickness and above	Twice the thickness of test-piece	Do. Do.
Do. Do.	Below $\frac{5}{16}$ in. in diameter or thickness	—	Doubled over and flattened on itself
Rivet Bars	All diameters	—	Do. Do.

Tests for Manufactured Rivets.—Same as for B.S.S. 16.

TABLE OF TENSILE TESTS.

Material.	Dimensions.	Ultimate Tensile Strength. Tons per sq. in.	Minimum Elongation. Per cent. On Test-Piece A.	On Test-Piece B.	On Test-Piece B.1.
Plates—for other than cold-flanging or cold-pressing	$\frac{5}{16}$ in. thick and above	28–33	20	—	—
Plates—for cold-flanging or cold-pressing	Do.	25–30	23	—	—
Sections and flats—for other than welding or cold-working	$\frac{5}{16}$ in. thick and above	28–33	20	—	—
Do. for welding or cold-working	Do.	25–30	26	—	—
Round and square bars—for other than welding or cold-working	$\frac{5}{16}$ in. diam. or thickness and above.	28–33	—	20	24
Do. for welding or cold-working and rivet bars . .	Do.	25–30	—	26	30
Cold drawn material for rivets below ½-in. diameter	—	20–25	—	26	—

STEEL PLATES, SECTIONS, BARS, AND RIVETS FOR LOCOMOTIVES—
B.S.S. 17A—1942 (of Report No. 24, Part VI.).

Same as B.S.S. 17, but without analysis. No special provision is made in the case of locomotive frame plates.

STEEL PLATES, SECTIONS, BARS, AND RIVETS FOR CARRIAGES AND WAGONS—
B.S.S. 18, B.S.S. 18A—1942 (of Report No. 24, Part VI.).

Specifications same as for locomotives, B.S.S. 17, B.S.S. 17A.

WROUGHT IRON.

(B.S.S. No. 51—1939.)

Specifications are given for wrought iron, Grades A, B, and C. The tensile tests, etc., vary with the sizes of the different sections rolled. Manganese shall not exceed 0·10 per cent. in the case of Grade A and 0·15 per cent. in the case of Grades B and C.

(B.S.S. 858–1939.)

Specification given for best Yorkshire wrought iron. The tensile tests, etc., vary with the sizes of the different sections rolled.

Manganese shall not exceed 0·06 per cent. and phosphorus shall not exceed 0·16 per cent.

Other Materials used in Locomotive Construction.

Iron Castings.—The tensile test is the most trustworthy; 7 tons per sq. in. is the usual specification; 8 and 9 tons may be demanded in reason and expected. General foundry iron up to 11½ and 12½ tons per sq. in. and cylinder iron up to 13½ and 14 tons per sq. in. are not unknown, each having good fractures, and being good workable foundry metal. The usual commercial test is generally made transversely upon bars 3 ft. long, 2 in. deep, and 1 in. thick, without being machined, 25 to 32 cwt. being the specified central breaking load. The test bars should be cast by means of an attachment to the casting, and the specified ultimate loads should not vary more than 1 per cent. Special irons such as Meehanite have tensile strengths of 20 tons or more.

Alloy steels are now used to some extent for the motion parts of fast-running locomotives in order to minimise the weight of the revolving and reciprocating parts.

Connecting and coupling rods are the most important items.

Nickel-chrome steel has been used on the E.R., and a manganese-molybdenum steel is used to a large extent on the L.M.R.; the Southern Region has used 'Vibrac' steel.

Nickel steel has been used for boilers to effect saving in weight by its higher tensile strength.

(For further details see Section XXIII., Part I., Vol. I. 'METALLURGY.')

For *Bronze, Brass, White Metals, and other Alloys*, see Section XXIII., Part I., Vol I.

Use of Materials made to British Standard Specifications.

(Report No. 24—Parts I.–VI.)

No. 1.—Crank axles.

Nos. 2 *and* 2A.—Straight axles, locomotive and tender.

Nos. 3 *and* 3A.—Carriage and wagon axles.

No. 4.—Tyres for locomotive and tender. Class C or D is usually specified. Class E is used for motor bogies of electric stock.

Nos. 5 *and* 5*a*.—Carriage and wagon tyres. Class C for carriages and Class B for wagons are the most commonly specified.

Nos. 6X, 6Y, *and* 6A.—Laminated springs, purchased complete.

Nos. 6B, 6C *and* 6D.—Plates and bars for manufacturing laminated springs, spring wearing plates, flat springs.

Nos. 7X, 7Y *and* 7A.—Manufactured volute and helical springs.

No. 7B *and* 7C.—Spring steel for volute and helical springs.

No. 8.—Steel forgings of Classes A to D (as below).

No. 9.—Steel blooms for making forgings. *Class A* is used for motion work and pins and other parts requiring to be case-hardened; spring hangers and links, brake work, draw bar, and safety links and pins, coupling screw and shackle, regulator rod, fire hole ring, foundation ring, and other parts requiring a weld. *Class B* for ordinary forgings, valve spindles, motion work not requiring case-hardening, reversing shaft and rod, reversing screw, buffer heads, regulator handle, roof slings, safety valve, and other seatings, wash-out doors and bridges, etc. *Class C*, connecting and coupling rods, crossheads, keys and cotters. *Class D*, piston rods, slide bars, crosshead cotters.

No. 10.—Steel castings. Wheel centres form one class; castings with wearing surfaces another, and this includes axle boxes and guides, bogie centres, bogie slide, brake shaft brackets, spring brackets, buffer plungers and guides. General castings without wearing surfaces include motion plate or slide bar brackets, smoke-box saddle, etc.

No. 11.—Copper fire-box plates.

Nos. 12 *and* 12A.—Copper stays and studs.

No. 13.—Copper boiler tubes.

No. 14.—Brass boiler tubes.

No. 15.—Copper pipes for feed water and steam. Steel fire-box plates.

No. 16.—Boiler barrel and fire-box shell plates, front tube plate, rivets, stays, nuts, angles, rods, etc., for boiler work. Expansion brackets, roof stays, palm stays.

Nos. 17 *and* 17A.—Locomotive and tender frame plates and cross stays, angles, bolts, rivets, plating, ash pan and doors, fire door, smoke-box and door, cab, tender tank plates, etc.

Nos. 18 *and* 18A.—Steel plates, angles, and other sectional material for carriage and wagon underframes.

No. 53 (1927).—Weldless steel tubes for boilers. (B.S.S. 494 used in lieu.)

Contract Specification for Locomotive Construction.

The *Contract Specification* should provide for delivery and payment after running a guaranteed mileage (about 3,000 to 5,000). That all parts shall be constructed exactly to the drawings supplied, and no deviation made whatever without consulting those responsible for letting the contract, and that lack of information shall be no excuse for discrepancies in material or construction. The price quoted must be inclusive, and no responsibility shall devolve upon the purchaser for royalties and patent rights. The quotation should also include drawings (2 complete sets), photographs (12), lists of details, weights, etc., to be furnished with the (3rd or 5th) engine.

The specification should also stipulate that the workmanship should be accurate and well finished.

Liquid Fuel.

In oil-burning locomotives the ashpan is replaced by a brick lined floor to the fire-box in which are openings to admit air. Admission is governed by doors or else there is a false bottom into which air is admitted through a damper. The weir type burner about 2 in. wide permits the flow of oil fuel to fall on to a ribbon of steam issuing below from a slot $2\frac{1}{4}$ in. wide by $\frac{3}{16}$th in. deep. The fuel is pulverised and burns as a long flame directed towards a flash wall built below the fire-hole door. The lower part of the fire-box is also lined with firebrick. The burner is fixed below the front of the foundation ring and is housed in a small chamber communicating with the combustion space. Some 1,800 gal. of oil are carried in a reservoir on the tender and heating coils are arranged to facilitate the flow of fuel both there and at the burner.

On the Buenos Aires Pacific Railway the control arrangement has atomiser, auxiliary blower, oil-firing valve and damper combined in one operation.

The Laidlaw-Drew burning equipment used on the G.N.R. of Ireland has a vertical burner in centre of fire-box base. The advantages of this system include controlled steam raising, automatic supply of correct amount of air and minimum amount of brickwork. The train-heating pipe line is utilised to supply steam when starting up from cold.

Liquid fuel burning is seldom resorted to on British railways except at times of coal shortage resulting from prolonged strikes at the collieries. A temporary conversion to oil burning is readily made. A burner is inserted in the fire-hole, and is fed from the tanks placed on the tender, each of which holds from 400 to 500 gal. of oil. The flame is directed under the brick arch, no alteration being made to the fire-box except by bricking up the portion of the tube plate under the brick arch; the top portion of the fire-hole is closed by a special door, which can be opened if it is necessary to use coal.

The fire-bars are covered with a layer of broken fire-brick, and if it is at any time necessary to use a little coal, it is put at the back of the fire-box. The fire can be started with wood, and as soon as the steam pressure reaches about 40 lb. the burner can be used. In many cases no coal whatever is burnt for long periods, the engine steaming freely and the fire requiring very little attention. The best steaming results are, of course, obtained when there is a slight amount of smoke given off at the chimney. The locomotives fitted with this arrangement are able to deal easily with their accustomed loads. The advantage of the system is the ease with which it can be fitted up, and the simplicity in handling.

Oil is extensively burned in the South-Western States of America, Mexico, and the Argentine Republic, as coal has to be imported and oil is nearer at hand or is more cheaply transported.

See also Section XXV., Vol. I. 'FUELS.'

Mechanical Stokers.

Mechanical stokers are used in the United States, Canada and South Africa. They are adopted not for reasons of economy but because the locomotives had become too large for their full capacity to be maintained by hand firing. The consumption per unit of power developed is approximately the same with either hand or stoker firing. The same grade of fuel is used with mechanical stokers as is fired by hand, but in combination with a shaking grade it is possible to make use of a lower grade of fuel. The proper use of these stokers to obtain best results calls for as much skill on the part of the fireman as hand firing, but the labour is reduced.

Mechanical stokers can be justified only on locomotives too large to be fired to full capacity by hand. The minimum economical size of engine for them are those of a rated tractive effort of 50,000 lb., and then only when operating on long heavy grades. Engines having a rated tractive effort of 55,000 lb. or over justify the use of mechanical stokers on any territory in freight service, as also engines of 50,000 lb. tractive effort in passenger service.

The mechanical stoker for locomotives is a development in response to the necessity of securing greater capacity from single power units. Mechanical stoking permits more tonnage to be handled at a higher sustained speed, and gives greater average dependability than would be practicable with hand stoking. Its development has rendered it possible to design locomotives capable of burning coal continuously at a rate much in excess of the capacity of any fireman to supply it by hand. It is generally considered that a tractive capacity of 50,000 lb., or a coal consumption of 4,000 lb. per hour, will justify the use of a mechanical stoker. The present commercial production of mechanical stokers in the United States is confined to the overfeed or 'scatter' type, made by four companies. According to information supplied by those companies, 7,856 mechanical stokers were in service or on order on November 1, 1923, but their use has been considerably extended since then.

Mechanical stokers are under trial on British railways with a view to the use of coal of a lower quality. The French railways have put a number into service for this purpose.

Coal Consumption.

The average rate of combustion in British express engines on heavy service is about 100 lb. per sq. ft. of grate area per hour, and about 50 lb. of coal per mile; the maximum rate may reach 160 lb. per sq. ft. of grate area per hour.

According to the Railway Returns for 1938, the lb. of coal consumed per engine-mile for the four British railway groups were: for passenger service 49·38 lb., for freight service, including shunting, 55·69 lb.

In 1918 average consumption of coal for all purposes was approximately 60·0 lb. per mile. The increase in consumption is mainly attributed to a decline in the quality of the fuel.

The consumption of fuel in a locomotive engine at rest, as when standing in a station or at a signal, is about one-fifth of that consumed when the engine is in motion. When standing at a shed with fire banked the consumption is very much less.

In a modern engine about 3¼ cwt. of coal is actually consumed in raising steam in a boiler from cold water, and about 2¼ cwt. from hot water. Some 10 cwt. is required for making up the fire before working a heavy train.

Water Consumption.

With a piston speed of 900 ft. per minute and a boiler pressure of 175 lb. per sq. in., locomotives using saturated steam require about 24 lb. per i.h.p. hour. The evaporation is approximately 8 lb. of water per pound of coal, or 9½ lb. from and at 212°F. The coal consumption is therefore 3 lb. per i.h.p. hour. Locomotives using superheated steam require 25 per cent. less, or about 18 lb. of steam per i.h.p. hour. The coal per i.h.p. is 12½ per cent. less, or roughly 2½ lb. per i.h.p. hour, the evaporation being 7 lb. of water per pound of coal.

Lubricating Oil Consumption.

The oil consumption of different railways varies considerably, but under favourable conditions 1 lb. of cylinder oil and 4 lb. of engine oil are required per 100 miles. Superheater engines use rather more cylinder oil than non-superheater, and the quality of the oil is more expensive.

The oil consumption of the 4-6-0 on the Western Region engines averages 5 lb. per 100 miles.

According to the Railway Returns for 1938, the consumption of lubricating oil per 100 running miles for the four British railway groups was 6·41 pints for all services.

Sight feed lubricators on the hydrostatic principle, delivering oil to the steam pipes, are used on both superheater and non-superheater engines. It is more usual on superheater engines to employ a mechanical force feed lubricator driven by an attachment to some part of the engine motion.

The practice of feeding oil to the crowns of axle boxes by mechanical lubricators is also extending, more particularly in the case of heavy eight wheels coupled mineral engines.

For LUBRICATION AND LUBRICATORS, see Section XX, Part II., Vol. I.

Brakes.

In addition to hand brakes the majority of engines and tenders are fitted with power brakes, usually the Westinghouse or Automatic Vacuum apparatus (page 2113). Goods engines not fitted with a continuous brake have a simple steam brake. Many engines are fitted with a steam brake working automatically with the vacuum by means of a valve of simple construction which can either be controlled by the vacuum in the train pipe or by hand. This is found to be a convenience, and the brake cylinders being of small diameter occupy but little space under the engine. The arrangement is extensively used and gives very satisfactory results.

Instead of a small ejector for maintaining the vacuum in the train pipe, a vacuum pump about 5 in. bore directly driven from one of the crossheads of the engine can be used instead. As the pump is capable of creating a high vacuum, a relief valve is provided to limit the vacuity. On the Western Region this is set for 25 in., but other railways set the valve for the standard 20-inch vacuum.

Under average conditions as to load, leakage, and number of stops, the steam required by a vacuum ejector in maintaining a 20-in. vacuum results in the consumption of 42½ lb. of coal per hour. The power absorbed by a vacuum pump coupled to the cross-head of the engine is equivalent to a consumption of 15 lb. of coal per hour with a 20-in. vacuum, and 17½ lb. per hour with a 25-in. vacuum. (*J. N. Gresham*, '*Journal of Inst. of Loco. Engineers.*')

The percentage of brake power—that is, pressure on brake blocks in relation to load on wheels—should not exceed 75 per cent. for coupled wheels and 100 per cent. of weight when empty for tenders. If bogie wheels of engine are braked the percentage should not exceed 50 in order to minimise any chance of skidding.

To avoid frequent taking up of blocks brake leverage should not be too great. With steam and air brakes leverage is about 7 to 1; vacuum brakes are sometimes as much as 14 to 1 on account of difficulty of finding room for large cylinders. With this ratio a 7-in. piston travel allows ½-in. clearance for blocks and ½ in. wear.

Tenders.

On the British Railways each of the main line companies has evolved one or two standard designs of tenders to meet all their requirements.

The L.M.R. now build six-wheeled tenders exclusively, the larger holding 9 or 10 tons of coal and 4,000 gal. of water. Smaller tenders carry 5 tons of coal and 3,500 gal. of water.

On the E.R. there are two types in general use, one being of the six-wheeled type to carry 7½ tons of coal and 4,200 gal. of water; the other is of the eight-wheeled rigid axle type with a capacity of 8 tons and 5,000 gal.

The standard tenders on the W.R. are of the six-wheeled type and the largest carry six tons of coal and 4,000 gal. of water. Others are of 3,500 and 3,000 gal. capacity with 6 and 5 tons of coal respectively.

Each of the above companies has water troughs so that very large tank capacities are unnecessary.

The Southern Region has no troughs and uses large tenders for long distance runs. These are of the bogie type and have eight wheels; the water capacity is 5,000 gal., and there is a space for 5 tons of coal. This railway also has a six-wheeled tender of the same capacity.

The second standard tender design of the Southern Region is a six-wheeled one to carry 4,000 gal. and 5 tons of coal.

The connection between engine and tender in modern designs usually consists of a substantial eye-bolt pinned to the engine and fitted with a helical or rubber spring and nut on the tender.

It is also desirable to fit rubbing blocks between the engine and tender so that the tendency for side oscillation is resisted. Two safety links or bars are also provided in case the main drawbar should break.

Side buffers have been used to reduce oscillation, but the central block is better when curves are encountered.

In order that long non-stop runs may be made on the E.R., some of their eight-wheeled tenders have gangways through the tank on one side and which are coupled to the vestibule on the train to enable a change of enginemen to be made on the journey.

Tanks are now generally of welded construction instead of being riveted; a small saving in weight and less liability to leakage being the main advantages.

In the U.S.A. where water capacities of 9,000 to 14,000 gal. and 15 tons of coal are common, the six-wheeled bogie is used, and sometimes the tanks are cylindrical or round-bottomed.

Tender journals are almost invariably situated outside the wheels. The bogies are generally of the plain pivoted type without side play, but in the case of very long engines having a trailing truck or bogie, it is common to allow side play on the leading tender bogie to facilitate the motion of the locomotive round a curve. Roller bearings are gradually being adopted on tenders.

Water Troughs.

Effective length is from 440 to 560 yards. Theoretically a quantity of about 1,000 gallons can be picked up, the mouth of the scoop being at rail level and $1\frac{1}{2}$ in. below water level. In practice considerably more is obtained owing to the heaping up of the water in front of scoop, the maximum being obtained at 45 miles per hour with some 1,700 galls. At higher speeds there is a considerable loss by spray. The lowest speed at which water can be picked up is 15 miles per hour.

When running at 60 miles an hour over a water trough $\frac{1}{4}$ mile long, a locomotive tender will pick up about 1,000 gallons or 10,000 lb. of water.

Examination of Locomotives at Running Sheds.

A failure on the road is a serious matter, and the best way of reducing breakdowns to a minimum is by periodical examinations. The following is a list of the usual examinations of the L.N.E.R. (G.N. Section):—

Slide valves:—		Small ends	6 months
Piston valves	2 months	Vacuum regulating valves	1 month
Richardson's	2 "	Water scoops	1 "
'D' valves, $\frac{7}{8}$ in. thickness	4 "	Fire-boxes	1 week
'D' " $\frac{3}{4}$ and $\frac{11}{16}$ in. "	3 "	Axle-box pads	3 months
'D' " $\frac{5}{8}$ and $\frac{9}{16}$ ins. "	2 "	Injectors	1 month
'D' " $\frac{1}{2}$	1 month	Main steam pipes on superheated engines	1 "
Pistons	3 months	Tender wheels gauged	1 "
Tender tanks	1 month	Leading wheels of 2–4–0 type gauged	1 "
Drip valves	1 "	Engine and tender tyres gauged	1 "
Safety valves	6 weeks	Mechanical lubricators to slide valves and pistons	2 months
Ball valves in water gauge frames (dependent upon quality of water in district)	2 months	Mechanical lubricators to axle-boxes	1 month
Lead plugs	1 month		
Large ends	3 months		

Cracked frames, flaws in connecting and coupling rods, or any part of motion, loose tyres, etc., should also be watched for. (*G. A. Musgrave, 'Proceedings, Inst. Loco. Engrs.'*)

Engine and Tender Tools and Equipment.

Carried in Tool Boxes.

Flat chisel.
Pin punches (large and small).
Chipping hammer.
Wrench for union nuts.
Complete set of spanners.
Adjustable wrench.
Tommy bar.
Large can for engine oil (6 to 8 quarts).
Large can for cylinder oil (6 to 8 quarts).
Small can for paraffin (about 1 quart).
Spring valve oil feeder, long spout.
Spring valve oil feeder, short spout.
Flare lamp.
Case of fog signals.
Rod flag.
Hand sweeping brush.
Packing drawer } (if required).
Tallow pot } (if required).
Spare glasses for water gauge.
Spare glasses for sight feed lubricator (if fitted).
Spare packing rings for glasses.
Copper wire for trimmings.
Worsted for trimmings.
Spare cotter pins.
Spare nuts.

Carried on Tender.

Fire rake, Pricker, Dart, Tube scraper, Clinker shovel } (length to suit firebox).
Slice (to suit depth of firebox at back).
Coal hammer.
Coal pick.
Pinch bar.
Bucket.
Fireman's shovel.
Screw-lifting and traversing jack (if specified).

OTHER EQUIPMENT.

Full set of head and tail lamps.
Full set of head boards or discs (when used).
Water gauge lamp.
Sight feed lubricator lamp (if required).

SPECIAL EQUIPMENT AT SHED FOR CLASS OF ENGINE.

Spanner for brake screw adjusting nuts.
Box spanner for fusible plugs.
Ashpan rake.

RAIL MOTOR CARS AND TRAINS.

For certain feeder services to points on the main line or for branch-line working, it is more economical or more convenient to use either rail-motor trains or self-propelled railcars, which can be driven from either end.

The practice on the L.M.R. and S.R. is to use up obsolete locomotives and carriages by suitably converting them for motor services. A small tank engine is attached to one or more trailer cars to form a unit which can be driven from either end. When pushing the car, the locomotive is controlled by the driver from the opposite end of the train by means of levers and rods with universal joints connected to the regulator. A brake valve and whistle cord are provided and an electric bell signal enables the driver to communicate with the fireman on the footplate. In some cases control of the regulator is by means of compressed air or vacuum.

For their suburban traffic round Paris the French railways have developed this plan by building trains of nine coaches each. A powerful tank engine is attached at one end, and this can be controlled from a driving compartment at the opposite end.

The W.R. steam railcars built many years ago have been replaced by push-and-pull trains with steam locomotives of the 0–4–2 and 0–6–0 types, and partly by oil-engined cars.

During the past few years almost all new railcars have been built with internal combustion engines, the oil engine being used to a much greater extent than the petrol engine.

In England, where progress in this direction has been on a smaller scale than in Europe generally, the former L.N.E.R. put into service three oil-electric motor coaches of 250 b.h.p. each, and one oil-electric railbus of 110 b.h.p.; the former are of the heavy engine type, and the latter has an engine similar to those in road vehicles.

The former L.M. & S.R. experimented with small petrol-engined vehicles of the road-railer type **for running on the road or the rails; in France pneumatic tyred cars of very light weight and capable of high acceleration are being used. These cars with pneumatic tyres have to be built extremely light as the maximum load that can be carried under a pair of wheels is about 1½ tons; consequently a car of normal length requires two eight-wheeled bogies.**

The W.R. is the greatest user of oil-engined cars in this country. It has standardised on **A.E.C. oil engines of 130 b.h.p. and the Wilson pre-selective gear-box with hydraulic clutch. Its first car has one engine which is mounted outside the main frame and below the floor and drives on to the end of an axle. The seventeen subsequent cars have two engines each, and the latest ones have had improvements made to the engine as a result of experience with the original one which was standard with those of road vehicles. All these cars are for passenger carrying except one for parcel traffic.**

On the Continent there are railcars for all purposes, ranging from four-wheeled ones for slow branch lines to high-speed long distance cars, either running as single units or in two and three car sets. In Norway three-car sets include a compartment for diners.

The engines are generally of the high-speed type developed from road and aviation experience, and are mainly of the four-stroke airless injection design. The usual position for them is either on the bogies or underneath the floor; in the latter case the flat opposed cylinder engine is gaining ground and several manufacturers have developed this design.

Transmission systems have passed through many stages as the mechanical transmissions of road vehicles were quite unsuitable for railway working, and existing electric systems were heavy and of low efficiency.

Mechanical change-speed systems are almost entirely of the constant-mesh type where the drive is taken up gradually through slipping clutches or by hydraulic clutches, but above 150–200 h.p. the mechanical system has not been developed to any extent, therefore electric transmission is used for higher powers; its objection for small powers is its high cost and heavy weight.

Hydraulic transmission of the rotor type is being developed for all powers now, but is expensive in first cost and usually requires a change-up gear between itself and the engine. It is much lighter than electric transmission but has not the same wide range of torque variation, although this can be increased by the inclusion of a two-speed gear or two converters.

The efficiencies of the three transmission systems are all different from each other and are apt to lead to wrong conceptions of the tractive power at the rails.

A well designed mechanical transmission will give up to 92 per cent. efficiency, but it requires a variable speed engine and precludes a constant h.p. output.

The electric transmission has a comparatively low efficiency, ranging from about 65 per cent. at starting to a maximum of about 80 per cent.; it permits the engine to give almost constant maximum h.p. at all speeds if required.

The hydraulic transmission has a starting efficiency similar to that of the electric, but it soon rises and attains a figure between that of the mechanical and electric transmissions; its torque range is limited.

The electric transmission is unique in that the location of the driving axles is independent of the position of the engine, thereby permitting the motors to be situated in any convenient part of the car or train. The motors are either of the axle-suspended type or of the frame mounted type with a cardan shaft drive on to a worm or bevel gearbox.

Generally, when oil-engined cars are being built, advantage has been taken of new methods to build light-weight stock, thereby saving on the initial cost of the engine and on running costs.

In some recent designs the articulated type of car has been adopted. This has a driver's cab integral with a power bogie. The front of the car body is carried by the power bogie bolster and the other end by a trailing bogie. The passenger compartment is thus insulated from the noise and heat of the engine compartment, and the body can be readily detached from the power bogie.

In Italy an articulated set has a central power unit with a passenger coach at each end.

Super-charged engines now being manufactured have improved the power/weight ratio of railcars.

Some cars have been built to run on producer gas made from such fuels as charcoal.

In the U.S.A. the railway oil-engine has been developed mainly in large powers for trains of three to twelve cars, and electric transmission is used exclusively. These trains are usually run at timings faster than the ordinary trains and are popular for long distance as they enable journeys to be made during more convenient periods of the day than formerly.

Some of these trains are being purchased on long-term payments.

High Speed Services.

The competition of road transport, and in some cases of air transport as well, has led to the demand for increased speed in both passenger and freight services on railways all over the world. This has been met on branch lines by single cars of light weight and capable of high acceleration, propelled by oil engines. On main lines these have been developed into articulated train sets, consisting of two, three or more stream-lined cars, capable of speeds of 80 to 100 miles per hour.

As an alternative, specially built steam locomotives, partly or fully stream-lined, and capable of speeds in excess of 100 miles per hour, have appeared in Great Britain, Germany and the United States, and they draw lightweight stream-lined cars. The advantage of this arrangement over the articulated train set is that the number of cars can be varied to suit fluctuations in traffic.

OIL-ENGINED LOCOMOTIVES.

The application of the oil engine to railway locomotives has been given much consideration throughout the world.

In America most of the locomotives now being constructed are diesel-electric. It is claimed that owing to greater availability one diesel-electric can replace at least two steam locomotives.

Its adoption in any particular country depends largely upon the natural fuel resources of that country and its economics.

In England the cheapness of coal and the fact that over 90 per cent. of oil has to be imported, coupled with the cost of an oil-engined locomotive being more than twice that of a steam locomotive, has caused little progress to be made with the former except in small sizes for industrial work, and for railway shunting where the locomotive can be given a high load factor and be worked by one man.

On British railways two diesel-electric locomotives for passenger service have been put to work on the London Midland Region. Each has a 1,600 h.p. diesel engine of the 16-cylinder, Vee type, four-cycle construction running at 750 r.p.m. The locomotives are of the C_0-C_0 type with three motors per bogie. By coupling the two units together the heaviest and fastest trains can be operated.

OIL-ENGINED RAILCARS AND TRAINS (OUTSIDE U.S.A.).

Railway.	Gauge.	Type.*	Engine b.h.p.	at r.p.m.	Transmission.	Tare.	Weight Loaded.	Length over Body.	Seats.	b.h.p. per Ton Loaded.	Max. Speed m.p.h.
	ft. ins.					Tons.	Tons.	ft. ins.			
G.W.R., England . .	4 8½	B–B	2×130	2,000	Fluid flywheel and mechanical	25–29	29½–33½	62 0	68	8·8–7·8	75
Nord Ry., France . .	,,	Double bogie	2×150	1,500	Mechanical	34	—	—	62	—	62
P.O.-Midi, France . .	,,	,,	280	1,400	,,	36·7	44	76 5½	64	6·35	75
State Ry., France . .	,,	1A–A1	2×150	1,500	,,	21·8	30	—	64 or 81	10	75
Nord Ry., France . .	,,	3 cars 2–B_0,2–2,B_0–2	2×410	1,400	Electric	122	137·4	208 10½	150	5·97	90
Belgian National Ry. . .	,,	3 artic. cars 2–B_0–B_0–2	2×380	1,400	,,	136	154·5	196 10	229	4·92	87
Netherlands Ry., Holland .	,,	3 artic. cars 2–A1A–A1A–2	2×410	1,400	,,	95·5	—	271 8	160	—	70
Italian State Ry. . .	,,	1A–A1	2×145	1,700	Mechanical	28	32·5	74 5½	40	8·9	81
Buenos Ayres Provincial Ry.	Metre	Double bogie	270	1,100	Electric	36·5	43·4	65 8	62	6·2	49·5
Buenos Ayres Gt. Southern Ry.	5 6	1A–2	102	1,700	Fluid flywheel and mechanical	—	—	44 9	44	—	47·5
Argentine State Ry. . .	Metre	B–2	240	1,250	Mechanical	—	—	78 1	82	—	50
Egyptian ,, . .	4 8½	B–2	275	1,450	,,	36·8	42·5	72 2½	70	6·45	68
Italian ,, . .	,,	3 artic. cars	2×400	1,500	,,	85	95	193 6½	78	8·4	100
Danish ,, . .	,,	3 artic. cars 2–B_0–B_0–2	4×275	—	Electric	118	—	209 0	152	—	75
German ,, . .	,,	A–1	135	1,700	Mechanical	14·9	—	40 0 over buffers	42	—	45
,, ,, . .	,,	B–2	210	—	,,	29·5	—	67 6 over buffers	61	—	50
,, ,, . .	,,	B_0–2	300	1,100	Electric	42·2	48	72 3 over buffers	66	6·25	56
,, ,, . .	,,	B_0–2	420	1,400	,,	48	—	71 9 over buffers	61	—	62
,, ,, . .	,,	2 artic. cars 2–B_0–2	2×410	1,400	,,	93·47	—	145 0 over buffers	77	—	100
,, ,, . .	,,	3 artic. cars B–2–2–B	2×600	1,400	Hydraulic	117	—	196 0 over buffers	139	—	100

* See p. 522.

OIL-ENGINED TRAINS IN U.S.A.

Railway.	Gauge.	Type.*	Engine b.h.p.	at r.p.m.	Transmission.	Tare.	Weight. Loaded.	Length Overall.	Seats.	b.h.p. per Ton Loaded.	Max. Speed. m.p.h.
	ft. ins.					Tons.	Tons.	ft. ins.			
Boston and Maine . .	4 8½	'Flying Yankee' 3 artic. cars B_6–2–2–2	660	750	Electric	—	95·4	199 3	140	6·92	100
Gulf, Mobile and Northern .	,,	3 cars	660	740	,,	162	167	226 0	80+12 sleeping berths	3·95	90
Illinois Central . . .	,,	'Green Diamond' 5 artic. cars B_6–B_6–2–2–2–2	1,200	750	,,	212	—	328 6	146	—	80
New York, New Haven and Hartford	,,	'Comet' 3 artic. cars B_6–2–2–B_6	2×400	900	,,	—	126	207 0	160	6·35	110
Union Pacific . . .	,,	'City of Denver' 12 cars	2×1200	—	,,	569	596	864 0	257+ sleeping berths	4·02	102

* See p. 522.

OIL-ENGINED LOCOMOTIVES.

Railway.	Gauge.	Type.*	Service.	Engine b.h.p.	at r.p.m.	Transmission	Weight in Working Order.	Adhesive Weight.	Max. Tractive Effort.	Max. Speed.
	ft. ins.						Tons.	Tons.	Lbs.	m.p.h.
British Railways, L.M.R.	4 8½	C_0–C_0	Passenger	1,600	750	Electric	121·5	121·5	41,400	—
,, ,, ,,	,,	0–C–0	Shunting	350	680	,,	51	51	30,000	22
,, ,, ,,	,,	0–C–0	,,	180	900	Mechanical	30	30	14,400	13
P.L.M. Ry., France	,,	1–D_0–1	,,	600		Electric	94		41,000	35
C.B. & Q. Ry., U.S.A.	,,	B_0–B_0	,,	2×230	1,000	,,	65	65	40,000	50
Boston & Maine Ry., U.S.A.	,,	B_0–B_0	,,	2×300	550	,,	91·3	91·3	60,000	—
Baldwin Loco. Co., U.S.A.	,,	B_0–B_0	,,	660	600	,,	94·6	94·6	63,600	45
Illinois Central Ry., U.S.A.	,,	B_0–B_0+B_0–B_0	,,	2×900	750	,,	143	143	96,000	—
Union Pacific Ry., U.S.A.	,,	B_0–B_0+B_0–B_0	,,	2×1200	750	,,	192	192	—	—
Baltimore & Ohio Ry., U.S.A.	,,	B_0–B_0	,,	2×900	750	,,	116	116	50,400	—
Seaboard Air Line, U.S.A.	,,	2–D_0+D_0–2	Passenger	2×1500	625	,,	265	182·6	102,250	117
Pennsylvania, U.S.A.	,,	A–1–A+A–1–A	Freight	3×2000	850	,,	475 (3 units)		221,000	69
Atchison Topeka & St. Fe, U.S.A.	,,	A–1–A+A–1–A	Pass. or Freight	3×2000	1,000	,,	407 (3 units)	271		120 (P) 65 (F)
Ceinture Ry., Paris	,,	1–D_0–1	Shunting and freight	800	700	,,	87	65·4	41,000	37
B.A.G.S. Ry., S. America	5 6	1A–B_0+B_0–A1	Mixed traffic	2×850	550	,,	148·8	112	66,000	70
,, ,,	,,	1–C_0–1	,,	800	700	,,	78·6	51	28,500	70
,, ,,	,,	B_0–B_0	Passenger	420	650	,,	59	59		46·6
N.W. Ry., India	,,	1A–C_0–2	,,	1,200	630	,,	117·1	68·35	38,000	70
German State Ry.	4 8½	1–C–1	Mixed traffic	1,400	700	Hydraulic	77	47·3	—	2·5
Siamese ,,	Metre	2–D_0+D_0–2	Goods	2×750	600	Electric	125	86·5	63,800	28
,, ,,	,,	2–D_0–2	Passenger	1,000	800	,,	90			—
Algerian Ry.	4 8½	2–C_0–2	Express	920	700	,,	100	56	17,000	72
U.S.S.R.	5 0	2–D_0–1	Goods	1,200	450	,,	127	78·2	44,100	34·2

* See p. 522.

The London Midland Region has tried about eight small shunting locomotives ranging up to 200 b.h.p. with mechanical transmission, and larger locomotives of 350–400 b.h.p. with electric transmission; results have shown that electric transmission is the better for the larger powers. This company has in service a number of oil-electric shunters of 350 b.h.p.; these locomotives have six wheels, and some of them have a single motor driving a jackshaft through double reduction gearing and which is coupled to the wheels by rods, whilst the others have two axle-suspended motors with single reduction gearing and the three axles coupled by rods.

Similar locomotives are in use on the other Regions.

Where there is a demand for high tractive effort at slow speed, such as at hump yards, double-reduction gearing is of benefit. It saves some space and weight in the motors.

On the Continent there is no standardised design and orders have been usually for single units, excepting small shunting locomotives. Experiments have been made there with large powered locomotives having mechanical or fluid transmission, or direct drive, but none has had the success of the electric transmission.

In Siam the State Railway has embarked largely on oil-engined locomotives for future renewals, also the South American Railways have made considerable experiments in this direction, but owing to financial difficulties they have concentrated on the cheaper light railcar in preference to heavy locomotives.

Almost all large locomotives are fitted with the four-stroke airless injection engine as this is considered the most reliable and simplest for railway work. In the U.S.A. the two-stroke engine has been tried in the large sizes, but has not met with the same success as the four-stroke engine.

Super-chargers driven by exhaust gas turbines are now in use and are capable of increasing an engine output by 30 to 40 per cent.

Various transmission systems have been tried out as with railcars but the electric one has proved the most successful despite its high initial cost and low efficiency; the hydraulic transmission is making progress and may surpass the electric before long.

Railway oil engines usually run on light Diesel oil of about 0·87 or 0·88 specific gravity.

The general method of starting the engine is by electricity or compressed air. In the former case if the engine is small in cylinder capacity an electric motor is geared to the engine; with large engines having electric transmission the main generator has a starting winding and is motored by current from a battery to start the engine.

Where compressed air is used there are several ways available. For a small engine an air motor may be geared to it, or an air impulse starter may be fitted. With large engines the compressed air is admitted direct by special cams into some or all of the cylinders.

The fuel capacity of oil-engined locomotives is greater than that of steam ones; shunting locomotives can carry conveniently a week's supply, and main line locomotives and railcars can usually carry two or three days' supply.

Automatic control is a common feature of oil-engined rolling stock, such items as oil and water temperatures and pressures being governed within pre-determined limits by apparatus which can stop the main engine if the limits are exceeded.

Filtration of engine inlet air is very important as cylinder wear is increased by particles of dust in the engine.

One big advantage of electric transmission and of other types and engines electrically controlled, is the ease with which multiple unit running and remote control can be arranged. The Scharfenberg coupler is suitable for this purpose as it comprises one head which automatically couples the draw and buffing gear, brake pipes and electrical connections.

GAS TURBINE LOCOMOTIVES.

A gas turbine locomotive built by Brown Boveri has been running on the Swiss Federal and French National Railways since 1941. The turbine develops 8,200 h.p. of which 6,000 h.p. is absorbed by the compressor, leaving 2,200 h.p. for driving the electric generators which supply current to the traction motors. The locomotive is arranged for one-man operation. By substituting bunker oil for diesel oil the cost of operation has been reduced below that of a diesel-electric locomotive. Thermal efficiency of the gas turbine locomotive has been raised to 20 per cent.

The Western Region of British Railways has a similar but larger locomotive on order with Brown Boveri, and a second locomotive is to be supplied by Metropolitan Vickers.

Much development work is in hand in America on gas turbine plants destined for locomotives. Experimental work is being carried out on coal-burning gas turbines, the fuel being burnt in pulverised form. Fly ash is the main problem to be overcome.

ROLLING STOCK.

Recent Developments in Carriage and Wagon Stock.

Electric arc welding is being applied to the construction of underframes for carriages and wagons and for bogie frames. Some reduction in weight is brought about in dispensing with riveted connections.

On British railways galvanised steel plates of about $\frac{1}{16}$-in. thickness are used for panels of carriages in conjunction with a wooden framework. Plastic material is being tried for panels There is a trend, however, towards all-steel welded bodies.

Plastic materials have also been introduced for interior panels and upholstery, and some trial is being made of fluorescent lighting, mainly in America and France.

Air conditioning is used on the more important trains in America. Power requirements are such that there is a tendency to replace axle-driven generators by diesel-driven auxiliary plants to supply current for lighting, air conditioning, etc.

The Eastern and Southern Regions of British Railways use the American type of automatic coupling and vestibule for many of their main-line trains. In order that these vehicles can be attached to ordinary stock, the coupler is hinged and can be dropped down, exposing a hook for screw coupling. Side buffers are provided, and these have a loose sleeve which can be removed to allow the buffers to be pushed back when the automatic coupling and vestibule with central buffers are in use.

Articulated twin coaches consist of a pair of bodies hinged together. The outer ends rest on bogies in the usual way, but there is only one bogie for the inner ends, the bogie pin being at the hinge between the coaches. A flexible gangway connects the two bodies. This system has been extended to complete trains, so that a set train of five coaches has only six bogies.

In France and Switzerland trial is being made of light coaches carried on rubber-tyred wheels. As not more than about 1 ton can be carried per wheel the number of wheels per coach is increased to 20. They give improved riding comfort, reduced wear and tear, and greatly improved braking.

The high-capacity bogie wagon makes but slow progress on British railways, owing to the lack of terminal facilities. The tendency is to build 4-wheel wagons of 12, 15, or 20 tons capacity

Electric welding is now much used for underframes and in some cases for bodies.

The fitting of power brakes to goods vehicles is being slowly extended. Its more general application is retarded by the presence of a large number of private owners' wagons and the high capital costs involved. Braked freight trains are, however, a feature of all the main line companies to-day; they are made up entirely of vacuum braked stock and are hauled by the highest powered engines at scheduled speeds of 50 m.p.h. and enable overnight delivery of goods to be made between most of the large towns in the country. Other goods trains are arranged so that as many as possible braked wagons are connected to the engine brake.

The container system of transporting goods is much used on British railways. Large boxes fitted with doors are filled at the factories, transported by road and then placed on railway wagons by a crane.

A new type of wagon is available for the transport of fragile goods, in which the body is not rigidly fixed to the underframe but is connected to it through four slides which allow of some longitudinal movement but prevent lateral or vertical movement relative to the underframe. The shock-absorbing element consists of four sets of rubber buffing springs which are attached to the underframe and two horizontal sets attached to the body. The effect of any longitudinal shocks delivered to the wagon are largely absorbed by these springs and do not reach the body. The wagons are also fitted with special shock-absorbing buffers on the headstocks.

General utility vans have been introduced by the Southern Region. These are 4-wheeled **vehicles, with a wheel base of 21 ft., and they have end as well as side doors. By means of special fittings they are adaptable for a number of purposes besides passengers' luggage, such as milk cans, motor-cars, road vehicles, fruit baskets, theatrical scenery, aeroplane parts, etc.**

One of these vans has been experimentally constructed with panels of reinforced plastic material. The underframe is of the cantilever type.

In England and in France some trial is being made with aluminium alloys for bodywork in order to save weight. Much more extensive use of this metal is being made in America for bodies and in some cases for underframes as well. It gives reduction in weight consistent with exceptional strength, durability, high corrosive resistance and low centre of gravity. Low carbon high strength steels are an alternative means of saving weight, as also copper-bearing steels, as thinner plates and sections can be used owing to reduction of rusting in service. The light tare weight of these wagons necessitates the use of load-compensating brakes.

In Germany some pre-stressed reinforced concrete wagons are being made.

CARRIAGES.—

Original Railway.	Type of Vehicle.	Length of Body.	Width of Body.	Height from Rail to Top of Roof.	Corridor or Non-Corridor.
LONDON MIDLAND REGION.					
		ft. ins.	ft. ins.	ft. ins.	
L.M.S.	Third	57 0	9 0	12 7¼	Centre Corr.
,,	Compo	60 0	8 11¼	12 4¾	Side Corr.
,,	Sleeping	65 0	8 11¼	12 4⅞	Side Corr.
,,	Dining & Kitchen	68 0	8 11¼	—	Side Corr.
,,	Sleeping	69 0	9 0	—	Side Corr.
,,	Articulated Cars	109 9	—	—	Centre Corr.
,,	Compo.	60 0	8 11¼	12 4⅝	Corr.
,,	Third Brake Compo.	57 0	9 0	—	Corr.
,,	Third	57 0	9 0	—	Centre Corr.
EASTERN & NORTH-EASTERN REGIONS.					
G.N.	First Dining	65 6	9 0	—	Corr.
,,	Compo. Dining	65 6	9 0	—	Corr.
,,	Brake Compo.	58 6	8 6	—	Corr.
N.E.	Third	52 0	8 6	—	Non-Corr.
L.N.E.	Compo. Dining Triple Set	153 7 over vestibules	—	—	Corr.
,,	Sleeping	66 6	9 3	—	Corr.
,,	Sleeping	61 6	9 0	12 10	Corr.
,,	Sleeping	66 6	—	—	Corr.
,,	First	63 0	8 11¼	—	Corr.
,,	Third	61 6	8 11¼	12 6	Corr.
WESTERN REGION.					
G.W.	Brake Third	70 0	9 0	—	Corr.
,,	Third	70 8¼	9 0	—	Corr.
,,	Dining	71 4½	9 0	—	Corr.
,,	Sleeper Compo.	60 0	9 0	—	Corr.
,,	Third	61 4½	9 5¾	13 0	Corr.
,,	Kitchen	61 4½	9 5¾	13 0	Corr.
SOUTHERN REGION.					
Western	Third Brake Compo.	64 6	9 6	—	Corr.
,,	First & Third Compo.	64 6	9 6	—	Corr.
,,	First & First Dining	61 6	9 6	—	Corr.
,,	Third Dining & Kitchen	64 6	9 6	—	Corr.
,,	Third Saloon	64 6	9 6	—	Centre Corr.
,,	Third Brake Compo.	64 6	9 6	—	Corr.
Eastern	First	62 0	8 0¾	12 2	Corr.
,,	Brake First	62 0	8 0¾	12 2	Corr.
,,	Brake Second	62 0	8 0¾	12 2	Corr.
S.R.	Third	59 0	9 0	12 4	Centre Corr.
OTHER STOCK.					
Paris-London	Sleeping	59 0	9 0	12 10¾	Corr.
Nord Ry., France	3 Artic. Cars	190 3½	9 5¾	13 1½	Corr.
B.R.	Pullman	63 10	8 5¾	12 5	Corr.

BRITISH RAILWAYS

Weight Empty.	Compartments. 1st Class.	Compartments. 3rd Class.	Seats. 1st Class.	Seats. 3rd Class.	Bogie.	Bogie Wheel Base.	Centre of Bogies.	Remarks.
T. C.						ft. ins.	ft. ins.	
30 0	—	—	—	56	4 wheel	9 0	40 6	All-steel
—	—	—	18	24	,,	9 0	43 6	—
33 16	—	7	—	28	,,	9 0	48 0	—
—	—	—	24	—	6 wheel	12 6	45 0	—
41 16	12	—	12 berths	—	,,	12 6	46 0	Welded frame
49 0	—	—	—	112	,,	9 0	46 6¾	Welded frame
32 5	—	—	18	24	4 wheel	9 0	43 6	—
—	—	4	—	24	,,	9 0	—	—
—	—	2 saloons	—	56	,,	—	—	—
40 0	2	—	20	—	—	—	—	—
41 8	1	1	8	18	—	—	—	—
29 19	3	3	12	18	—	—	—	—
24 0	—	8	—	80	—	—	—	—
81 8	—	—	36	42	4 wheel	8 6	47 0 42 1 47 0	Articulated: 3 cars on 4 Bogies
37 15	—	8	—	32	,,	8 6	47 0	—
34 10	—	7	berths	28	,,	8 6	43 0	—
42 10	10	—	10	berths	,,	—	—	—
33 0	6	—	36	—	,,	8 0	43 6	—
33 0	—	7	—	42	,,	8 0	43 6	Auto-coupler
—	—	4	—	32	4 wheel	9 0	53 6	—
—	—	10	—	80	,,	9 0	53 6	—
—	—	—	18	32	,,	9 0	53 6	—
—	—	5 ord. 3 sleep	berths	40 12	,,	—	—	—
34 3	—	8	—	64	,,	9 0	44 6	—
42 13	—	—	—	—	,,	9 0	44 6	—
32 14	—	2 & 1 sln.	—	48	—	—	—	6-coach Sets. Radial Sided
34 6	4	3	24	24	—	—	—	—
32 1	3 & 1 sln.	—	18 & 24	—	—	—	—	—
38 15	—	1 saloon	—	32	—	—	—	—
32 9	—	2 saloons	—	64	—	—	—	—
32 14	—	2 & 1 sln.	—	48	—	—	—	—
31 14	6	—	24	—	4 wheel	8 0	44 0	Continental Boat Train Stock
30 9	5	—	20	—	,,	8 0	44 0	
30 9	—	6 Second	—	36 Second	,,	8 0	44 0	
35 0	—	2	—	56	,,	8 0	40 0	Open Third
50 0	9	—	—	—	4 wheel	8 2½	40 0	Wagon Lits Co.
73 16	—	—	—	274	,,	8 2½	57 5	Light metal construction
40 0	—	—	24 or	42	,,	10 0	43 4	All steel

WAGONS—

Original Railway.	Type of Wagon.	Carrying Capacity.	Length over Body.	Width over Body.
LONDON MIDLAND REGION.				
		Tons.	′ ″	′ ″
L. & N.W. . .	Open Goods . .	10	18 0	7 10
" . .	" " . .	20	21 0	7 9
" . .	Covered " . .	10	18 0	8 0
Midland . .	Open " . .	10	16 0	7 10
" . .	" " . .	12	16 0	7 10
" . .	" " . .	20	21 6	7 11
" . .	Covered " . .	10	17 6	7 8
L. & Y. . .	Open " . .	12	16 0	7 10
" . .	" " . .	20	21 6	8 0
" . .	Covered " . .	10	20 0	8 0
" .	" " . .	30	35 0	8 0
EASTERN & NORTH EASTERN REGIONS.				
G.N. . . .	Open Goods . .	10	19 0	8 0⅛
" . . .	" " . .	20	21 6	8 5
" . . .	Covered " . .	8	19 0	7 8
N.E. . . .	Open " . .	10	15 0	8 0
" . . .	" " . .	12	17 0	8 0
" . . .	" " . .	20	20 0	8 0
" . . .	Covered " . .	10	17 0	8 0
" . . .	" " . .	25	37 0	8 6
" . . .	Coal " . .	40	36 0	8 0
G.C. . . .	Open " . .	10	19 0	8 0⅛
" . . .	" " . .	20	21 6	7 9
" . . .	Covered " . .	10	19 0	7 8
G.E. . . .	Open " . .	10	17 0	7 7
" . . .	" " . .	20	21 6	8 2
" . . .	Covered " . .	10	19 0	7 9
" . . .	Guard's Brake . .	—	17 6	7 9½
" . .	Cattle Wagon . .	10	19 3	7 11⅛
WESTERN REGION.				
G.W. . . .	Open Goods . .	10	16 0	8 0
" . . .	" " . .	20	21 0	7 9
" . . .	Covered " . .	10	16 0	7 10
" . . .	Coal . . .	20	21 6	8 0
SOUTHERN REGION.				
Western . .	Open Goods . .	10	16 0	7 11½
" . . .	" " . .	12	18 0	7 11
" . . .	Covered " . .	10	18 0	8 0
Central . . .	Open " . .	10	15 5	7 9
" . . .	" " . .	12	16 6	7 11
" . . .	Covered " . .	8	18 4	7 8½
" . . .	Ballast . .	15	20 0	8 0
" . . .	Cattle Wagon . .	6	18 4	8 0
S.E. . . .	Guard's Brake . .	—	13 10¾ (24 0 over head stocks)	8 4 (over look-outs)

BRITISH RAILWAYS.

Depth inside.	Tare Weight.	Wheel Base.	Centres of Journals.	Size of Journals.	Remarks.
′ ″	T. C. Q.	′ ″	′ ″	″ ″	
3 0	6 13 0	9 9	6 6	9×4	
4 7	8 0 2	12 0	6 6	10×5	
7 $8\frac{3}{8}$	7 10 1	9 9	6 6	9×4	
3 2	5 17 3	9 6	6 6	9×$3\frac{3}{4}$	
4 $1\frac{3}{8}$	6 10 1	9 6	6 6	9×$4\frac{1}{4}$	
5 $1\frac{1}{8}$	8 7 0	12 0	6 6	10×5	
7 9	7 12 3	10 0	6 6	9×$3\frac{3}{4}$	
4 $1\frac{3}{8}$	6 14 2	10 6	6 4	9×$4\frac{1}{4}$	
5 $11\frac{5}{16}$	9 16 0	12 0	6 6	10×5	
7 8	7 11 1	10 6	6 4	9×$4\frac{1}{2}$	
—	14 17 1	—	—	—	Bogie Wagon
3 5	6 9 0	10 0	6 $4\frac{1}{2}$	8×$3\frac{3}{4}$	
5 $2\frac{5}{8}$	8 9 0	12 0	6 6	10×5	
7 1	6 18 0	10 0	6 $4\frac{1}{2}$	8×$3\frac{3}{4}$	
2 $2\frac{1}{8}$	5 10 0	8 6	6 5	8×4	
3 $1\frac{5}{8}$	6 7 0	9 6	6 5	$9\frac{1}{2}$×$4\frac{1}{2}$	
6 $10\frac{7}{8}$	9 1 0	10 6	6 6	10×5	
7 9	7 2 0	9 6	6 5	8×4	
7 10	15 0 0	—	—	—	Bogie Wagon
6 0	16 5 0	—	—	—	Bogie Hopper Wagon
3 5	6 10 0	10 0	6 6	8×4	
4 10	7 17 0	12 0	6 6	10×5	
7 1	7 10 0	10 6	6 6	8×4	
4 $0\frac{1}{8}$	6 7 2	9 6	6 6	8×$3\frac{3}{4}$	
4 7	9 1 1	12 0	6 $7\frac{1}{2}$	10×5	
7 $0\frac{7}{8}$	6 9 2	10 6	6 6	8×$3\frac{3}{4}$	
—	20 0 0	—	—	—	
—	7 6 2	—	—	—	
3 3	6 0 0	9 0	6 6	8×$3\frac{3}{4}$	
4 $9\frac{3}{8}$	8 12 0	12 0	6 6	10×5	
7 $7\frac{1}{8}$	6 10 0	9 0	6 6	8×4	
4 $9\frac{1}{2}$	10 3 0	12 0	6 6	10×5	All-steel
2 $8\frac{7}{8}$	5 19 0	9 0	6 8	9×4	
4 $8\frac{3}{8}$	6 18 2	10 6	6 8	9×4	
7 $7\frac{1}{2}$	6 17 0	10 6	6 8	9×4	
2 $10\frac{5}{8}$	5 18 1	9 3	6 3	8×4	
4 $2\frac{3}{8}$	6 17 1	9 0	6 6	9×$4\frac{1}{2}$	
7 1	6 8 0	9 9	6 3	8×4	
2 $3\frac{1}{2}$	6 15 0	12 0	—	—	
7 9	6 13 0	11 2	—	—	
—	26 0 0	16 0	—	—	

Rolling Stock for Southern Railway Electrified Lines.

Extensive electrification of suburban and main lines for passenger train working has been carried out on 600 volt d.c. third-rail system. Rolling stock takes the form of set trains of two, three, four, five or six coaches, permanently coupled together to form the various units. Two or more units are joined together to form longer trains, as required. Motor bogie has two nose-suspended traction motors, each driving an axle through gear wheels.

***Two-Coach Set.*—For branch and light traffic. One motor coach with motor bogie at outer end; one trailer coach with driving compartment. Non-corridor sets, 62 ft. 6 in. long by 8 ft. ¾ in. wide bodies, 129 ft. 5½ in. over buffers, 70¾ tons tare. Seats, 24 first class, 135 third. Corridor sets, 62 ft. 6 in. long by 9 ft. bodies, 129 ft. 5½ in. over buffers, 74¾ tons tare. Seats 24 first class and 88 third. Two 275 b.h.p. motors, geared 1 to 2·8.**

***Three-coach Set.*—For suburban services. Two motor coaches with trailer between them. Non-corridor, 62 ft. 6 in. by 8 ft. ¾ in. bodies, 193 ft. 10½ in. over buffers, 109·6 tons tare. Seats 56 first and 180 third. Four 275 b.h.p. (or 300 b.h.p.) motors, geared 1 to 2·8. Motor bogies at outer ends.**

In conjunction with these sets, two-coach trailer sets are used to form eight-coach trains with another driving unit, when traffic is heavy. In some cases three three-coach sets are used to form trains of nine coaches. Two-coach trailer units have bodies 50 ft. by 8 ft are 107 ft. 1 in. over buffers, and weigh 43·7 tons. They provide 180 third-class seats.

***Four-coach Set.*—For semi-fast services. Two motor coaches with two trailers between them, one trailer being a corridor coach with lavatories for first and third class. Other vehicles are non-corridor. Bodies, 62 ft. 6 in. by 9 ft., 256 ft. 11 in. over buffers, 139 tons tare. Seats 70 first class and 204 third. Four motors of 275 b.h.p., geared 1 to 2·8. Motor bogies at outer ends. Two sets are joined to form eight-coach trains when required.**

***Six-coach Set.*—For express services. Two all-steel open saloon-type motor coaches with four trailers between them. Two motor bogies per motor coach, each with two 225 b.h.p. traction motors, gear ratio 1 to 2·5, total 1,800 b.h.p. per set. All bodies are 62 ft. 6 in. by 9 ft.**

Sets with Pullman car:

Motor coach	59 tons.	Seats 52 third class		
Compo.	35½ „	„ 24 „ „	and 30 first.	
Pullman	40 „	„ 16 „ „	„ 12 „	
Compo.	35½ „	„ 24 „ „	„ 30 „	
Corridor (third)	31½ „	„ 68		
Motor coach	59 „	„ 52		
Total	260½ „	„ 236 „ „	72 „	

Sets without Pullman:

Motor coach	59 tons.	Seats 52 third class,	
Corridor (third)	31½ „	„ 68 „ „	
Buffet and first	32 „	„ — „ „	30 first class.
Corridor (first)	31 „	„ — „ „	42 „ „
Corridor (third)	31½ „	„ 68 „ „	
Motor coach	59 „	„ 52 „ „	
Total	244 „	„ 240 „ „	72 „ „

When required, two sets are joined to form twelve-coach trains.

***Five-coach Pullman Sets.*—For express services. Two motor coaches, each equipped with four 225 b.h.p. motors, with three trailers between. All Pullmans, first and third class. When required, two sets all joined to form ten-coach trains.**

Axle Boxes.

These are usually of cast iron, but a welded axle box fabricated from ¼-in. plates and a ½-in. backplate is now being introduced for wagons; for carriages oil boxes are universal, and a large number of new wagons are now being fitted with them instead of grease boxes. An oil reservoir in the bottom of the box conveys oil by capillary attraction, either by cotton-waste packing or by means of a worsted pad with tail trimmings supported on springs. The bearing or 'brass' is of gun-metal, or gun-metal lined with white metal. Some white metals have a tin base and contain about 60 per cent. tin and 25 per cent. of lead, with 10 per cent. antimony and a little copper. The lead base alloys have about 76 per cent. of lead, 15 per cent. of antimony, 6 per cent. tin, and a small percentage of copper.

The oil used is a mineral oil, and sometimes contains a trace of fatty oil.

A leather-faced dust shield prevents dirt or water creeping in at the back of the box.

In India axle guards for containing the axle boxes are being successfully replaced by radius links. Wear is found to be reduced.

Roller Bearings.

In some countries roller-bearing axle boxes are being increasingly used for passenger and freight vehicles, notably in U.S.A.

Experiments made on British railways seem to indicate that the friction of a plain bearing lubricated by an underpad is even less while running than a roller bearing, although the starting effort required with the latter is reduced, as also the oil consumption.

Owing to the higher capital cost there is, however, little or no inducement to adopt the roller bearing on British railways. On railways abroad, where the workmanship, maintenance, and materials are not so high-class, so that friction is greater, some advantage is gained by the adoption of roller bearings, and a small saving in fuel is found, sufficient with the other advantages realised to justify the adoption, this tendency being more marked in the Northern countries in Europe having cold winters. The bearings are also applied to trains de luxe in France and elsewhere on the Continent, such as the 'Blue Train' and the 'Golden Arrow.' Favourable conditions for the use of roller bearings are also found in North America, where some of the steel passenger coaches weigh as much as 70 to 80 tons each. Owing to the axle loading the journals are of large diameter, while the wheels are smaller than in England, so that the frictional resistance is high, and the risk of heating increased. The use of roller bearings reduces the friction, enables an easier and smoother start to be made, and obviates the risk of hot bearings, besides reducing oil consumption.

Wheels.

For carriages the wheel with wooden centre, once universal, has now been superseded to a large extent by a steel disc centre of slightly dished form, on which is shrunk a steel tyre having a lip on outside, and secured by retaining ring inside. Tyres are 5 ins. wide and are coned 1 in 20. In order to ensure smooth running, all carriage wheels are carefully balanced. Each pair of wheels is placed on a machine, the journals resting in bearings mounted on springs. When the wheels are revolved at high speed any irregularity in the running can be detected, and a correction made by fastening small plates to the inside of the wheel, near the circumference, so that smooth running results.

Wood centres must be bonded by a copper strip between tyre and hub, so as to conduct electric current from rail to rail for operating signalling appliances. Tyre shrinkages are $\frac{3}{16}$ in. for wood centres, and about $\frac{1}{64}$ in. for steel centres.

Wagon wheels are made in a variety of ways. The bossed spoke wheel consists of spokes made from flat bar bent into triangular shapes. These are assembled in position and the boss is formed on them by placing two steel discs brought to welding heat on either side of the spokes and forcing them together in a powerful press. The bossed spoke wheel was formerly made by casting a cast iron boss on them, but this practice is no longer in favour. Spoked wheels have also been made by forging spoke bars and building up a wheel. But this has been largely superseded by a cast-steel spoke wheel. A few wheels are being made with steel disc centres.

Solid disc wheels are sometimes used in the case of guards' brake vans where the heat generated by prolonged braking is liable to loosen tyres shrunk on wheel centres.

In Canada rubber-cushioned wheels are being produced. They give smoother riding and noise is reduced by over 80 per cent.

Journals of axles are sometimes case-hardened, but very good results can be obtained by the simple process of burnishing in the lathe or by grinding, after the journal is machined. In the case of the former, a tool having a small roller at the end is fixed in the slide rest and the journal is rolled all over with considerable pressure; this closes all the pores and leaves a hard, bright surface.

The maximum bearing pressure for journals varies from 500 to 700 lb. per sq. in. of projected area.

A reduction in diameter of journal by $\frac{1}{4}$ in. is allowed before axles are scrapped. In India worn axles are being given a longer life by turning down the journals and sleeving them with a steel bush.

Springs.

Laminated bearing springs are usually $3\frac{1}{2}$ ins. wide for carriages, and 4 ins. for wagons, with plates $\frac{3}{8}$ to $\frac{1}{2}$ in. thick, approximately.

If L = distance in inches between centres of bearings measured along top plate; b = breadth of plates in inches; t = thickness of plates in sixteenths of an inch; n = number of plates; W = safe load in tons; D = deflection in sixteenths of an inch per ton of load,

$$n = \frac{15LW}{bt^2};\quad W = \frac{bt^2n}{15L};\quad D = \frac{2L^3}{bt^3n}.$$

If there are different thicknesses of plates in one spring, it is necessary to find the equivalent number of plates of uniform thickness from the square (or cubes in the case of deflection) of the thickness, thus :—

$$n = \frac{n_1 t_1^2 + n_2 t_2^2}{t_1^2}.$$

See also p. 551.

Buffer springs are largely made of rubber, and are formed of rubber discs 2½ to 3 ins. thick placed between 6 or 8 metal discs.

For *Helical Springs* see p. 553.

Couplings.

Draw bars should be 1⅞ ins. diameter; they should be made from wrought iron, Grade A, or welding steel. Area of end of eye = area of two sides × 1·2.

Couplings should be made of wrought iron, Grade A, or welding steel. Standard diameter = 1½ in.

Screw couplings are of wrought iron, or sometimes of steel (B.S.S. 9B).

Draw hooks should be of wrought iron, Grade A, or welding steel. If stamped out, the grain of the iron should be kept running with the hook, an important point sometimes overlooked in second-rate work.

FOR BRITISH STANDARD SPECIFICATIONS FOR MATERIALS USED IN THE CONSTRUCTION OF RAILWAY ROLLING STOCK, see pp. 550 to 561.

Heating Railway Carriages by Steam.

Heating of trains is by steam from the locomotive boiler. The train piping consists of 1½ in. bore iron piping, and flexible rubber hose connections with universal couplings are provided between carriages. Automatic ball valves and thermostats are fitted for drainage of water. A pressure of 60 lb. per square inch is usually the maximum, but 30 lb. is quite an average pressure for ordinary working. The G.W.R. adopts a maximum pressure of 80 lb. per square inch.

Several systems are in use, but they may be classed under two headings, high-pressure and atmospheric. In the high-pressure systems, long cylindrical heaters, 4 ins. to 5 ins. diameter, are placed under one seat of a compartment. These are in direct communication with the train pipe, and steam under pressure fills them. As condensation takes place the water is drained away through a thermostatic valve. There are several modifications of this system.

In the atmospheric systems the heaters have an open outlet at one end, and a jet of steam entering at the opposite end expands to atmospheric pressure and condenses on the walls of the heater, the water draining away at the outlet. In some systems regulation of steam is by means of a thermostat, which automatically controls the steam. In others a choke plug in the branch pipe to the heater restricts the supply of steam to the necessary amount.

With coal of such a quality that it produces 7½ times its own weight of steam in the locomotive boiler, the average consumption of coal for the three coldest months should not exceed about 6½ lb. per axle per hour.

An approximation for the steam required for heating in Great Britain is 100–120 lb. per hour per coach.

American practice is different from that in Europe. External temperatures are lower and the internal temperature is higher, usually about 70° F., consequently the steam consumption is much greater and for a 70–80 ft. coach may be as much as 200–250 lb. per hour. The pressure in the train pipe is about 175–200 lb. per sq. in. maximum, and rigid pipes with ball joints and couplings are used instead of hoses.

On trains hauled by diesel traction the locomotive carries a boiler heated by exhaust gas, with oil-firing as auxiliary when required. A tank of water also has to be carried. A similar arrangement is provided on electric locomotives: the boiler in this case is electrically heated. It is more usual, however, for a boiler truck with a coal or oil-fired boiler to be attached at the rear of the train. In South Africa a special boiler vehicle is attached next to the locomotive where it is under the supervision of the driver. It has an electrically-heated boiler, current being drawn by a collector bow from the overhead conductor.

BRAKES.

Latest Developments.

The Westinghouse automatic empty and load brake with straight air control has been introduced for railways having long gradients necessitating prolonged braking periods. Straight air is used for normal application, while the automatic action is reserved for emergencies or breakaways. A change-over valve is provided on each vehicle; it has two positions—in one reduced pressure is obtained for empty to half-full vehicles. Full pressure is given by the other position for vehicles more than half-full. The control is set by hand.

The use of light-weight stock by reducing tare in relation to gross weight has called for a load compensating brake in U.S.A. Full force of each brake cylinder is reduced on empty and part-filled vehicles by the admission of air on the opposite side of the piston. Its pressure is governed by a weighing device which pre-sets a valve.

On passenger stock electrical control in parallel to the pneumatic system ensures simultaneous and quicker setting of the brake.

Some trains on high speed service have a 'Decelostat' connected to each axle. By this means greater brake block pressure can be applied. If a pair of wheels tends to skid the appliance momentarily reduces air pressure until the wheels are revolving normally.

Action of Brakes.

f = proportion which resistance produced by brakes bears to weight of train.

v = speed in feet per second.

V = speed in miles per hour.

Distance in feet in which the train is stopped on level $= \dfrac{v^2}{64 \cdot 4 f} = \dfrac{V^2}{30 f}$

Brakes self-acting on all the wheels, $f = \cdot 14$.

Ordinary brakes worked by hand in brake-vans, $f = \cdot 031$ to $\cdot 023$.

On gradient *ascending* 1 in n, resistance $= f + \dfrac{1}{n}$.

On gradient *descending* 1 in n, resistance $= f - \dfrac{1}{n}$.

Distance in feet for stopping $= \dfrac{f \times \text{distance on level}}{f \pm \frac{1}{n}}$

Power required by Brakes.

The following results have been obtained as regards consumption of energy with different kinds of air brakes :—1. On city lines, the consumption of energy for excentric compressors was 41·5 watt-hours per car kilom. (66·4 watt-hours per car mile); with geared compressors, 31·2 watt-hours per car kilom. (49·9 watt-hours per car mile); and with motor compressors, 16·16 watt-hours per car kilom. (25·86 watt-hours per car mile). 2. On suburban lines, the consumption for excentric compressors was 22·8 watt-hours per car kilom. (36·5 watt-hours per car mile); with geared compressors, 14·6 watt-hours per car kilom. (23·4 watt-hours per car mile); and with motor compressors, 6·32 watt-hours per car kilom. (10·11 watt-hours per car mile).

The Westinghouse Automatic Brake.

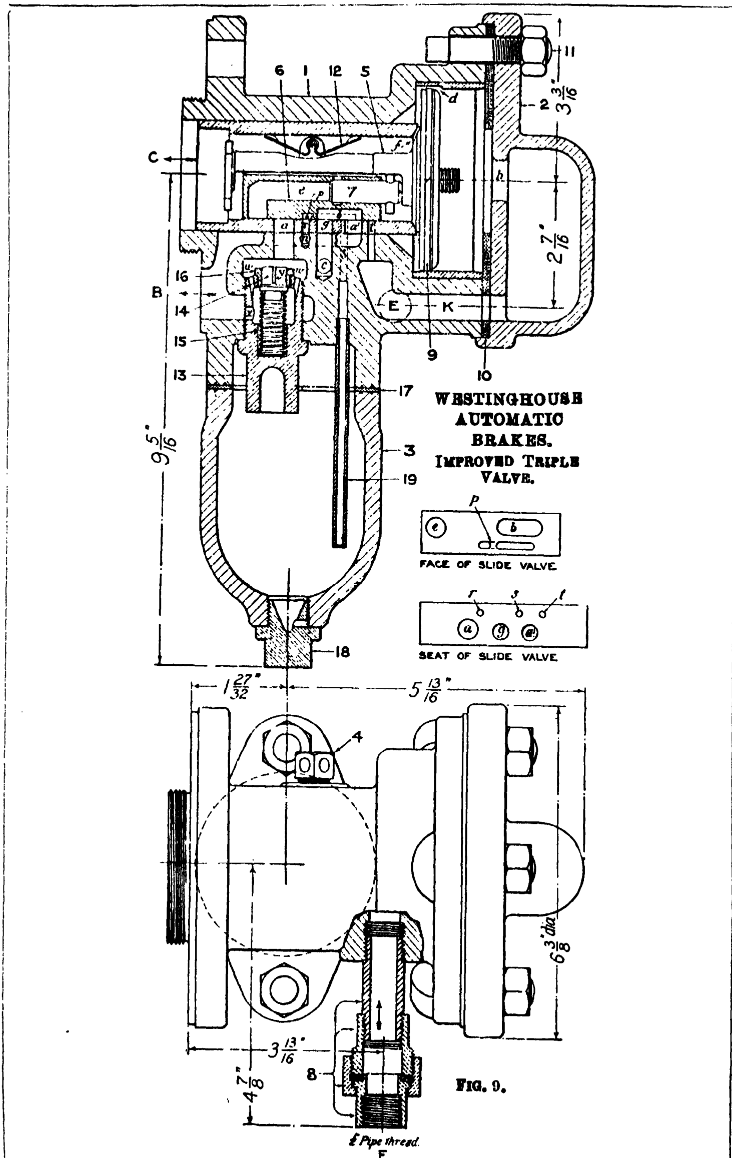
The Westinghouse automatic brake is continuous throughout the train and is operated by compressed air furnished by the compressor and stored in the main reservoir on the engine. This compressed air is fed by the driver's brake valve into the train pipe and, past the triple valves, into the auxiliary reservoir on each vehicle.

The brake is applied by reducing the pressure in the brake pipe, which causes the pistons of the triple valve to move and permit some of the compressed air stored in the auxiliary reservoirs to pass to the brake cylinders, the pistons of which are forced outwards, applying the brake blocks to the wheels.

The brake is released by restoring the air pressure in the train pipe, which causes the triple valves to close the communication between auxiliary reservoirs and brake cylinders and open a port from the brake cylinder to the atmosphere, through which the compressed air escapes from the cylinder. The spring in the cylinder can then push back the piston and withdraw the blocks from the wheels.

The brakes are usually applied by the driver, or in cases of emergency by the guard, but a break-away, or rupture of a hose coupling, or other accident causing an escape of air from the brake pipe, also immediately applies the brakes.

It is recommended that a pressure of 90 lb. per sq. in. be maintained in the main reservoir, and 70 lb. per sq. in. in the brake pipe.



WESTINGHOUSE AUTOMATIC BRAKES.
IMPROVED TRIPLE VALVE.

FIG. 9.

WESTINGHOUSE AUTOMATIC BRAKE.
DRIVER'S EQUALISING BRAKE VALVE.

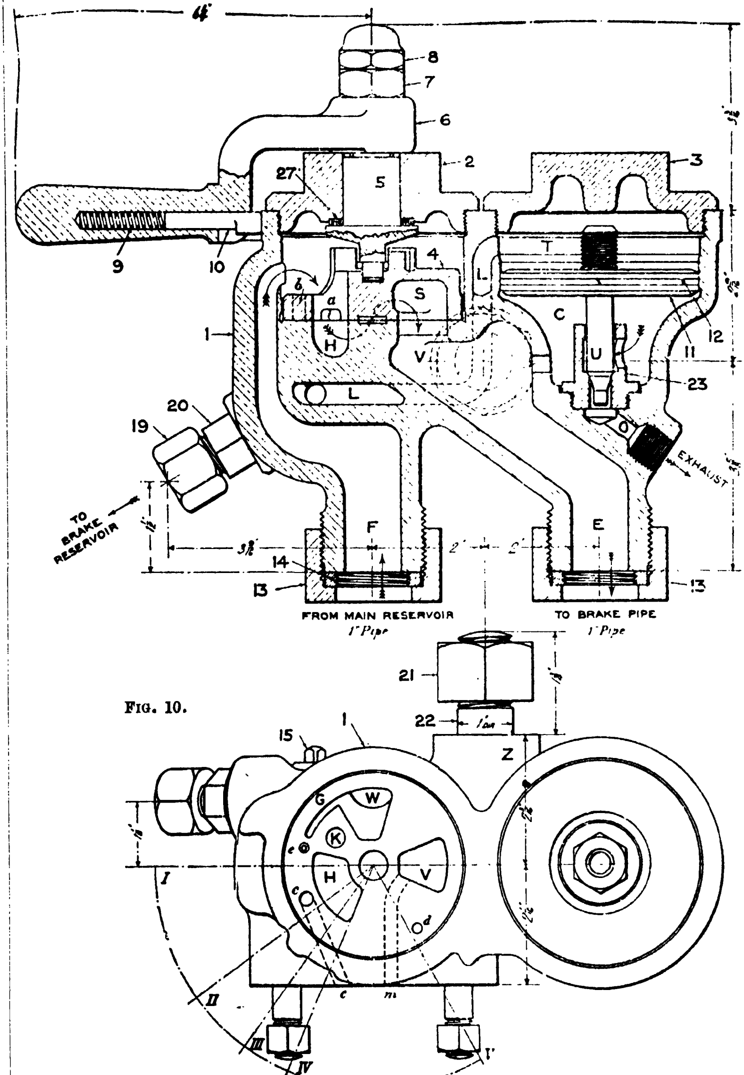


FIG. 10.

IMPROVED TRIPLE VALVE.

This valve has been designed to give a closer approach to simultaneous action of all the triple valves in a train than heretofore. As shown in fig. 9, p. 580, the valve is in the release position and the bulb is open to the atmosphere. When the slide valve 6 is moved in applying the brakes, the bulb 3 is closed to the atmosphere and opened to the brake pipe by the cavity *p* and port *s* shown in dotted lines. The local reductions of brake pipe pressure thereby produced by the forward triples in a train cause an earlier action of the rearward triple valves, which results in a more nearly simultaneous braking effect being produced throughout the train in every case of first setting the brakes than heretofore, with a smoother and more efficient brake service generally.

A further improvement, also tending to smooth and even action of all the brakes, is shown in the passage leading from port *a* in the seat of the slide valve 6 to the brake cylinder at B. It consists of a removable plug 13, perforated with holes *w* and *x*, which are always open, and an additional hole *y* carrying an automatic check valve 14, supported by a spring 15. It is so arranged that, when the brakes are first set, the automatic valve is opened against the spring resistance by the excess of air pressure above it, so that the hole *x* controls the rate of flow of air to the brake cylinder: as the air pressure on the under side of the valve increases, the spring closes the valve, when the smaller holes *w* control the rate of flow to the brake cylinder. The diameters of the regulating holes *w* and *x* are made proportional to the diameter of the brake cylinder with which the triple valve is to be used. By changing these perforated plugs, the same pattern of valve is thereby suited to cylinders of various sizes.

The bulb at the bottom of the triple valve is made in different sizes, proportioned in content to the volume of the brake pipe of the vehicle on which the valve is to be used.

A separate passage a^1 is provided in the triple valve body from brake cylinder to slide valve 6 for the air to escape from the brake cylinder on releasing the brakes. The rate of release is controlled as heretofore, by a nipple screwed into the outlet hole *c* in the triple valve body.

DRIVER'S EQUALISING BRAKE VALVE.

The principle of this brake valve, fig. 10, p. 581, is not to allow the driver in ordinary applications of the brakes to discharge air directly from the brake pipe, but from a small reservoir connected with chamber T of the valve. The reduction of air pressure, thus effected in the reservoir, is then at once automatically and properly repeated in the brake pipe by means of a small equalising piston, 11, placed between the air in the chamber T, and that in the brake pipe E. The piston moves in accordance with the variations in pressure above and below it, and governs an exhaust valve U in such a manner that the air pressure in the brake pipe must always equalise with that in the small reservoir connected with the valve chamber T. Even if the exhaust of air from the brake valve reservoir is abruptly stopped by the driver, the discharge valve U is always gradually closed by the piston, when equilibrium of pressure is established throughout the train. The equalising device, therefore, will under all circumstances provide for a uniform reduction of the pressure in the main brake pipe, thus ensuring a uniform application of the brakes to all the vehicles in the train.

The compression of air is effected by a Westinghouse compressor.

PIPE COUPLINGS.

In the connecting-pipe *coupling*, which is shown by fig. 11, the two halves of couplings, 1, are exactly alike, and an air-tight joint is formed by means of the rubber packing-ring, 3, 1 in each, which rings face each other when the couplings are united. The air-pressure in the

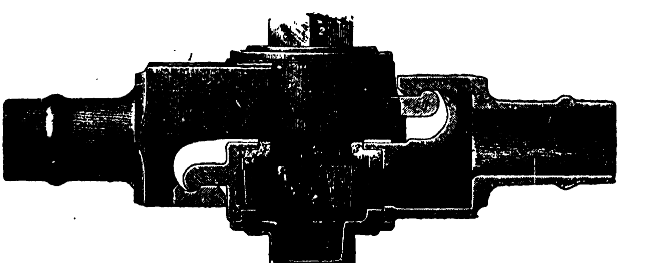


FIG. 11.

couplings tends to force these rings towards one another, so that the joint becomes tighter with increase of pressure. The tendency of the air-pressure in the couplings is also to force them apart in a direction at a right angle to the line of the india-rubber hose-pipes, and, consequently

the greater the pressure, the more firmly are the couplings held together by the projecting piece, *g*, of each coupling, which fits in a corresponding groove of the other coupling. No damage is done if the couplings be drawn apart forcibly by the separation of the train, as the rubber rings, 3, 3, are forced into their respective couplings far enough to permit the projections to disengage from their grooves. These couplings are united by simply placing them face to face at right angles, the stop-pins being on the under side, and then turning the projection of the one into the groove of the other.

The Vacuum Automatic Brake.

The Vacuum Automatic Brake is applied by admitting air to the brake cylinders through a continuous brake pipe, and released by withdrawing the air. To charge the brake cylinders, air from above the piston is withdrawn through a non-return ball valve to below the piston and thence direct through the brake pipe to a high-speed rotary exhauster in the case of an electric locomotive, or an ejector if a steam locomotive.

A service brake application is made by the driver admitting air to the brake pipe through a valve on the locomotive, and atmospheric pressure entering the brake cylinder below the piston acts against the vacuum above the piston and applies the brake at a pressure proportionate to the reduction of vacuum in the brake pipe. Air is prevented from entering the evacuated space above the piston by the non-return ball valve.

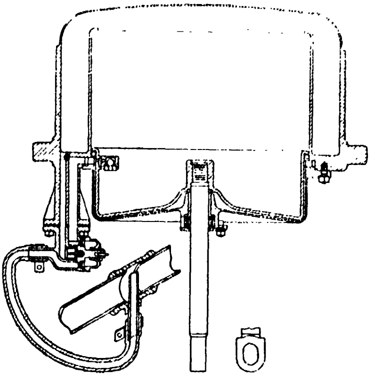


FIG. 12. 'Prestall' Wrought Steel Vacuum Brake Cylinder with internal ball valve and external release valve. Piston in release position. Air above piston is exhausted through the internal ball valve. The brake pipe connection to cylinder (shown in section) is provided with a siphon to prevent condensed moisture from being drawn into cylinder.

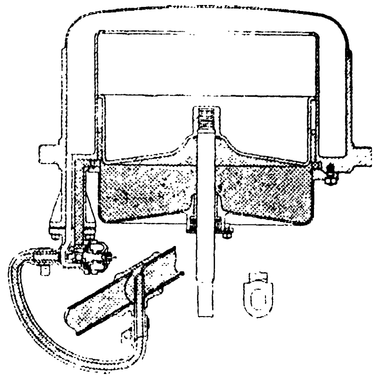


FIG. 13. 'Prestall' Wrought Steel Vacuum Brake Cylinder with external ball valve. Piston in 'brake on' position. A brake application is made by admitting air to the shaded portion below piston. The non-return ball valve prevents air entering the space above piston.

An emergency brake application may be made with the guard's van valve at rear of train, or a slow brake application by the passenger emergency valve fitted on each carriage. The brake is automatically applied throughout the train if an accident occurs sufficient to part the couplings.

Passenger trains are generally operated at 20-in. vacuum, but a few railways use 24-in. vacuum; and freight trains at 20-in. vacuum, or 16-in. vacuum on long trains where high vacuum is difficult to maintain.

Brake cylinders are either of the combined type with the vacuum chamber forming the outer casing of the cylinder (see figs. 12 and 13) or separate type with an independent vacuum chamber. In the former case, the cylinder is connected to the brake pipe by a single branch valve, and in the latter case a double branch is used, the second branch connecting the vacuum space above piston with an independent vacuum chamber.

The ball valve is housed either inside the piston as shown in fig. 12, or in the casing of a detachable valve as fig. 13. The former type of valve is less liable to leak through grit being drawn on to the valve seat, and it possesses a further advantage that when the piston moves upwards during a brake application the port in the wall of the piston (leading to the ball valve) passes to the

vacuum side of rolling ring, and thus isolates the valve and cuts off a possible source of leakage that might render the brake inoperative. The poppet type release valve (see fig. 13) used on cylinders with an internal ball valve, effects a more rapid manual release of the brake than is possible where the ball valve (see fig. 13) combines the functions of check valve and manual release valve as with the externally housed ball valve.

The tendency to reduce deadweight wherever possible, and the use of higher tensile materials in the construction of rolling stock has led to the introduction by the Westinghouse Brake & Saxby Signal Co., of brake cylinders made entirely from pressed steel. These are 30 to 40 per cent. lighter than cast-iron cylinders and possess greater resistance to fracture or breakage. These cylinders, known as the 'Prestall' wrought steel cylinders (figs. 12 and 13) have a detachable base for the easy renewal of the rubber rolling ring packing between cylinder and piston, the total detachable weight being 50 per cent. less than the cast-iron type in which it is necessary to remove the entire cylinder from casing in order to renew the rolling ring. The piston rod is guided by a gunmetal bush and rendered airtight by a rubber gland arranged so that atmospheric pressure acting

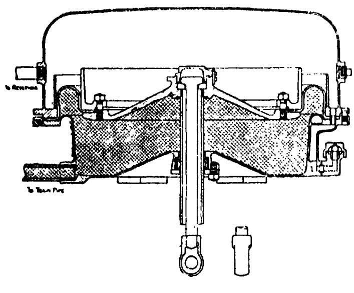


FIG. 14. 'Prestall' Wrought Steel Diaphragm Cylinder. The rolling action of the diaphragm between the vertical flanges is shown with the Piston at 'brake on' position; the shaded portion represents atmospheric pressure.

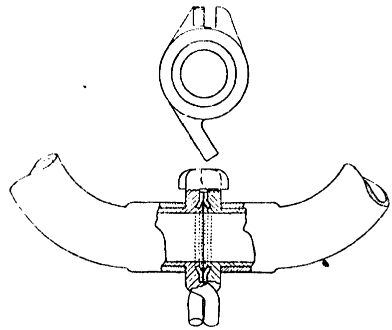


FIG. 15. Vacuum Hose Coupling. *Above*, End Elevation; *Below*, Sectional side Elevation.

on the outside diameter of the gland forces it with an even pressure around the piston rod. The rubber gland is easily renewed without dismantling the cylinder by simply unbolting the steel cover-plate forming the gland cap.

In all cylinders with rolling ring packing a certain amount of leakage is liable to occur after the rolling ring has become worn, and to avoid this, certain railways use a cylinder with a reinforced rubber diaphragm, since the vacuum can be maintained above piston for a longer period with a diaphragm than is possible with the rolling ring packing. This is an important feature for railways with long gradients, since it is impossible to recreate the vacuum above piston without first releasing the brake. The 'Prestall' wrought steel diaphragm cylinder, fig. 14, is provided with a vertical guide flange on the piston follower, and a corresponding flange in the dome of the cylinder, so that the diaphragm rolls between these two walls and the atmospheric pressure beneath the diaphragm is confined to a vertical path in the line of piston travel. This enables a longer and more powerful stroke than is possible when the diaphragm is not so guided.

The continuous brake pipe is generally 2-in. bore, but a few narrow-gauge railways use 1½-in. bore. The brake pipe is connected between vehicles by means of flexible internally armoured rubber hose, at the end of which is secured a metal coupling head with rubber jointing washer. These couplings when connected are held together by the horn at the bottom of coupling, which hooks on to a similar horn on the opposite coupling, whilst a lug and slot at the top of the coupling fits into a similar lug and slot on the opposite coupling, and keeps the bore of the two couplings in register. This coupling in the form approved by the Railway Clearing House is shown in fig. 15.

On their latest rolling stock the Southern Region is using one 30-in. Prestall cylinder instead of two of 22 in. as heretofore. Brake shafts can be dispensed with and much weight saved.

See also Descriptive Section XXXII, Part II:
Superheater Company, Ltd.
United Kingdom Self-Adjusting Anti-Friction Metallic Packing Syndicate, Ltd.

SECTION XXXII

PART III

ELECTRIC RAIL TRACTION

(Revised by W. J. Arnold Sykes, A.M.I.Mech.E.)

British Standard Specifications relating to Electric Traction.

No.	Title.
2—1944.	Tramway Rails and Fishplates.
8—1939.	Tubular Traction Poles.
23—1933.	Trolley Groove and Wire.
68—1914.	Steel Conductor Rails (Resistance).
74—1937.	Charging Plugs and Sockets for Battery Traction.
24 (2)—1942.	Locomotive and Carriage Tyres.
101—1929.	Tramway Tyres.
102—1930.	Tramway Axles.
468—1932.	Rolled Steel Wheels and Wheel Centres.
150—1922.	Cast Steel Wheel Centres for Tramcars.
173—1941.	Electrical Performance of Traction Motors.

FORMS OF ELECTRIC TRACTION.

Forms Involving Electrification of Route.

These include long-distance and suburban railways, tramways, and trolley-bus routes, the electric energy being supplied to the vehicle from overhead lines or additional rails.

Forms not Involving Electrification of Route.

These include: (*a*) all forms of battery locomotives, road vehicles, trucks, etc., and entail charging stations and charging periods; (*b*) oil-electric traction, where the power of the internal combustion engine is conveyed electrically to the driving axles. Here the electric generator and motors merely form a transmission and so replace the gear box or other mechanical mode of speed change.

ELECTRIC TRACTION ON LONG-DISTANCE RAILWAYS.

There are at present four systems for electric traction on long-distance railways: (1) Direct-current, (2) Three-phase, (3) Single-phase, and (4) Split-phase—a combination of (2) and (3). The Three-phase system, although a complete success, has made considerable headway in but one country—Italy; its drawback is the double overhead conductor. The Direct-current, Single-phase, and Split-phase systems only are in active conflict to-day.

In most European countries the choice of system has been definitely decided by national commissions: these bodies, after weighing the facts, have set forth the details of the system most suited to their needs. Thus, in Great Britain the well-known report of the Electrification of Railways Advisory Committee has recommended the adoption of Direct-current at 1,500 volts. The same conclusion has been reached by the authorities in Holland, Belgium, Czecho-Slovakia, and France; whereas Switzerland, Germany, Austria, Sweden, and Norway have standardised the Single-phase system at low frequency. In the U.S.A. the electrification of the Chicago, Milwaukee, and St. Paul Railway, a large trans-continental route, on the Direct-current system at the high pressure of 3,000 volts, has had a considerable influence on some European decisions. The more recent adoption by the Virginia Railroad of the Split-phase arrangement for one of the heaviest freight services in the world has, however, again drawn attention to the capabilities of A.C. for electric train-haulage, while the Pennsylvania Railroad has adopted single-phase for a main line with heavy traffic.

The motors employed for Direct-current traction are universally of the Series type, yielding a torque which is greatest at starting, falling off as the speed rises. (See fig. 2, p. 590.) *Speed regulation*—a very important matter for locomotive work—is obtained by series-parallel grouping of the motors—the standard method employed in traction to vary the motor terminal voltage. Further speeds entail weakening of the motor field flux: a method, while economical, resulting in the reduction of the useful torque per ampere of armature current. Starting is effected by the insertion of resistance in the motor circuit. For passenger and express work the losses in the rheostats may well be insignificant, but for goods and shunting operation, or where stops are frequent, these losses may become important.

For Single-phase traction in Europe, the Single-phase neutralised series motor is used almost universally. The machine is similar in principle and performance to its D.C. counterpart; some inherent defects have been overcome by sound design and the use of a low frequency ($\frac{50}{3} = 16\frac{2}{3}$ periods). The decisive advantage, however, is the feasibility of a high contact line pressure; 20,000 volts has been successfully tried, and 16,000 volts is standard practice. Voltage reduction by tapped static transformer obviates the necessity for starting resistances by providing a simple means of regulating the current to the motors. Tappings are usually on the low-tension winding, but a later development in Sweden and Switzerland is the use of high tension control with the tappings on the high tension side, thereby avoiding the switching of the very heavy secondary currents. The tappings can be graded so as to allow of small and regular increases in starting tractive effort, thus making the maximum possible use of the adhesion without slipping the wheels. Experiments are being made in taking the supply from national Three-phase 50-cycle lines, either by using grid controlled mercury arc rectifiers to reduce the frequency or obtain D.C., or by the use of 50-cycle current in the traction motors.

In America the later developments in Single-phase train-haulage have been with the Split-phase arrangement. Here a higher frequency can be used on the lines with advantage. The high-voltage Single-phase energy taken from the trolley wire is transformed down on the locomotives and passed through a phase-converter, which supplies two- or three-phase energy to the induction motors used for traction. The use of these robust and efficient motors is a great advantage, and commutators are avoided. This is important in the U.S.A., where 25 cycles has become the standard frequency for A.C. traction. Polyphase induction motors are lighter than either D.C. or Single-phase machines, but speed regulation is difficult, complex pole-changing and cascading devices being required. Rheostatic starting is essential. The recuperation of energy by the motors when descending a grade is automatic, and this explains the adoption of the Split-phase system on lines dealing with heavy mineral trains on steep gradients; moreover, braking is positive; energy is returned to the line; tyres, brake blocks, and rails are subjected to less wear; and the high line-voltage makes the collection of large quantities of energy a simple matter. More recently the Single-phase-D.C. converter locomotive has been developed.

The straight Three-phase system possesses the disadvantage of requiring two overhead conductors, the track forming the third phase. The voltage between phases is thus limited, the load is always more or less unbalanced, and overhead work at junctions and termini is very complicated. Induction motors are used as for the Split-phase system.

The systems employed for electric traction which have proved to be technically sound are shown in the following table, which traces the connection from locomotive to power-station:—

Equipment of			Supply.
Locomotive.	Track.	Substation.	
D.C. motor . .	Overhead wire . / Third rail . .	Rotary converter . / Motor generator . / Mercury rectifier and transformer .	Three-phase at normal or low frequency.
Polyphase induction motor . .	Two overhead wires . .	Transformer . .	Three-phase at normal or low frequency.
Polyphase induction motor, with phase converter . .	Overhead wire .	Transformer . .	Three-phase at normal or low frequency.
Single-phase series motor and transformer . .	Overhead wire .	Motor generator and transformer . .	Three-phase at normal or low frequency.
		Transformer . .	Single-phase at mainly low frequency.

The choice of an electrification system in any specified case is first and foremost a matter of finance. In the main, the installation which is cheapest when taken from every standpoint

is the system to be installed. An important secondary matter is interference with communication circuits owing to the cost of essential safeguards. In total mileage the amount of Single-phase is about equal to Direct-current electrification.

The following summary shows the chief developments in long-distance and main-line electric traction.

SYSTEMS IN USE IN MAIN-LINE ELECTRIFICATION.

Single-phase System.—With overhead conductor, usually 15,000 volts at 16⅔ cycles—Austria, Germany, Scandinavia and Switzerland. In U.S.A., a voltage of 11,000 volts at 25 cycles is used. In Hungary, the frequency is 50 cycles with split-phase supply from phase-converters to the locomotives.

Direct-current System.—With overhead conductor or third rail—France (1,500 volts), Great Britain (1,500 to 600 volts), Holland (1,500 volts), Italy (3,000 volts), U.S.A. (3,000 to 600 volts).

Three-phase System.—With two overhead conductors, practically confined to original development in Northern Italy (3,700 volts, 16⅔ cycles).

GREAT BRITAIN.—Great Britain has comparatively little to show in the way of long-distance railway electrification, although there are numerous schemes and projects under consideration.

Standards for British Railway Electrification.—An attempt has been made to standardise the system, etc., for future developments, but a certain amount of latitude has been allowed, as the following will show:—

The Electrification of Railways Advisory Committee, appointed by the Minister of Transport in March 1920, issued its final report in September 1921, recommending that all new electrification should be on the direct-current system, and that the standard pressure should be 1,500 volts, except in the case of railways already operating at 600 or 1,200 volts D.C., or with alternating current. The adoption of 750 or 3,000 volts should be permitted where approved by the Minister. Both overhead and rail conductor collection were recommended, provided that the form and design of the conductors and structures were in accordance with specified regulations. Current should be generated on the Three-phase system, at the frequency in general use in the district concerned; 50 cycles could be used satisfactorily. Suggested regulations were given, with drawings, for the standardisation of contact rails.

The Railway Electrification Committee (1927) reviewed the 1921 recommendations in a report issued in 1928. It was found that extensions had been made to existing lines at 600 and 650 volts, making a total of 1,404 miles of track at low-voltage D.C., 77 miles of 1,500-volt D.C., and 19 miles of Single-phase, the Brighton section of the Southern Rly. having been converted to D.C. The Committee therefore recommended the adoption of D.C. working at 'maximum' standards of 750 and 1,500 volts, with third rail and overhead contact systems respectively, and with generation at 50 cycles, 3-phase. The Committee did not consider it necessary to provide for the inter-running of multiple-unit rolling stock between high- and low-voltage electrified sections. For locomotives, however, to enable inter-running to be effected, two maximum loading gauges were prepared. No. 1, with a maximum height of 12 ft. 8 ins. above rail level, will be suitable for inter-running on all British railways, except on certain short lines, such as light railways, tube lines, and goods branches, totalling 264 miles. Such locomotives will have a scope of practically the whole of the 20,000 route miles of railway. No. 2 takes advantage of the higher (13 ft. 2 ins.) loading gauges of the L.M. & S., L. & N.E., G.W., and Southern (central section main line) Railways, and locomotives constructed to this gauge will have a scope of rather more than 13,000 miles. A suitable overhead-contact current collector was suggested. To permit of inter-running electrically, locomotives should be equipped with 750-volt motors capable of being connected in series-pairs for operation on 1,500 volts, and both overhead and shoe collectors should be provided. The reservation was made that the electrical standardisation need not be effected by a railway company until one-third of its total locomotive stock has been replaced by electric engines. Details of third rail and overhead clearances were specified. The Committee considered it undesirable to set limits to the voltage drop and uninsulated return conductors, a decision with which representatives of the G.P.O. were in disagreement.

Report of the Ministry of Transport Committee on Main-Line Railway Electrification, 1931.

In 1929 a Committee on Main-Line Railway Electrification was appointed with Lord Weir as Chairman. The basis of the recommendation for an extensive conversion of British railways to electric working was the existence of the 'National Grid' network. At an estimated cost of £80 millions, the Central Electricity Board would construct and equip the necessary additional transmission lines and supply the railways with energy at 0·475*d.* per kWh., measured at the substation d.c. busbars.

The selected system was 1,500 volts d.c. with overhead contact line. The possibility of using oil-electric traction on branch lines with light traffic was also envisaged.

The advantages claimed for electrification were: increase of average speeds without increase of maxima; increased station capacity; absence of smoke; development of real estate sites over large stations; passenger amenities; increase of axle loads on account of the even torque; better conditions for train crews; flexibility to suit traffic demands; avoidance of fire from locomotive sparks. The concomitant disadvantages were: the heavy capital expenditure; possibility of

wholesale disorganisation by enemies, disaffected persons, or accident to supply; electrolysis, interference with communication circuits; possibility of fire after derailment; risk of shock.

The economic investigation showed that limited main-line schemes of electrification were not worth while. Only full and complete electrification was deemed to show sufficient financial advantage.

ESTIMATED CAPITAL COST OF ELECTRIFICATION OF ALL STANDARD GAUGE STEAM-OPERATED RAILWAYS IN GREAT BRITAIN.

Item.	£ millions.
Track equipment	129·7
Auxiliary power cable	13·5
Alterations to Way and Works. . .	13·7
Electric tractors	136·5
Running sheds, shops, stores, etc. . .	4·5
Spare parts	5·5
Auxiliary power supplies . . .	2·25
Interest during construction . . .	12·5
Engineering expenses	5 0
Total gross cost	£323·15
Total of credit items	62·28
Net capital cost	£260·87
Additional expenditure on generating plant, transmission lines and substations, to be incurred by the C.E.B. and other authorised undertakers, approximately, millions.	£80

The total mileage to be equipped is 36,000 track miles of running line and 15,500 miles of sidings. Average rates of £2,500 and £1,800 per mile were respectively taken for track equipment. About 10,400 electric locomotives and 4,800 three- or four-coach multiple-unit equipments were allowed for.

ESTIMATE OF COMPARATIVE WORKING COSTS, STEAM AND ELECTRIC.
(Including only items affected by electrification.)

	£ millions.		
	Steam.	Electric.	Saving.
Locomotive fuel or energy	12·31	11·28	1·03
Locomotive wages	20·93	10·78	10·15
Repairs to tractors	10·82	4·66	6·16
Water for locomotives	0·88	—	0·88
Stores, etc.	0·91	0·45	0·46
Lubrication	0·29	0·10	0·19
Maintenance of sheds and shops . . .	0·44	0·18	0·26
Guards' wages	4·30	3·65	0·65
Cleansing of vehicles	0·93	0·70	0·23
Insurance, pensions, etc.	0·79	0·40	0·39
Savings on maintenance of train lighting .	—	—	0·51
Savings on auxiliary power, etc. . . .	—	—	0·84
Totals .	52·60	32·20	21·75
Additional costs of electrification :—			
Maintenance and renewals of track		3·38	
Maintenance and operation of substations		1·05	
Increase in depreciation of electric tractors compared with steam		0·02	
		4·45	
Net saving due to electrification			17·30
Revenue on haulage of power station coal			0·25
Total .			£17·55

The saving represents 6·7 per cent. on the capital cost of £261 millions. The item for energy is the total at 50 Wh. per ton-mile of 114,000 million trailing ton-miles at 0·475*d*. per kWh.

The return is moderate, but it would be augmented by the revenue from suburban railways, which would in practice form the initial part of any scheme of large-scale conversion.

Electric Locomotives.

The first electric locomotives were developed directly from the street tramway, and had bogie trucks provided with nose-suspended motors. Progress in America has in general avoided mechanisms involving reciprocating motion, and the nose-suspension and the quill drives have been widely applied to passenger, freight and shunting locomotives. On the Continent of Europe there was a widespread tendency at first to apply the steam locomotive side-rod mechanism to the electric locomotive. Many difficulties were encountered and generally overcome, but the continuous torque of the electric motor was fundamentally unsuited to the reciprocating nature of the drive, whilst the effect of steam cushioning in the steam locomotive cylinders had no counterpart in the electric drive. Subsequent designs introduced springs and similar elastic members to provide for this, and to obviate the troublesome oscillatory phenomena in side-rod drives so damaging to the bearings. The present tendency in Europe is to return to individual-axle drives with nose-suspended motors or with some form of special gearing enabling the motors to be entirely spring-borne. Among these are the Westinghouse (geared-quill, with spring connection between gear wheel and driving wheel) and the Buchli (link connection) drives, and a later form of bevel drive in which the motors have vertical shafts. Individual-axle drives allow greater freedom in the running-gear design and are adaptable to most classes of service, tending to simplification of spare parts and maintenance. The locomotives built in Great Britain have been mostly fitted with nose-suspension drives. Various types of contactor control are used.

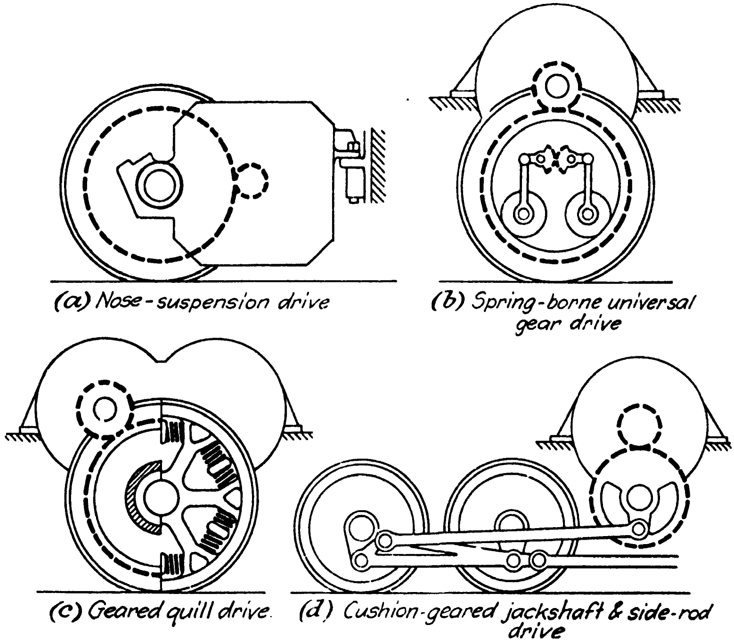


FIG. 1.—Typical Driving Mechanisms for Electric Locomotives.

Fig. 1 shows diagrammatically the arrangement of the more usual forms of individual-axle drive. (*a*) The nose-suspension (or tram) drive is extensively used for all classes of locomotives, as well as for all multiple-unit rolling stock and most tramway cars. The weight of the motor is divided between the frame and the driving axle, the former by means of a 'nose' (sometimes spring-loaded) and the latter by means of an outboard bearing. The weight is low and almost entirely non-spring borne, the gauge and the wheel diameter limit the motor output, and the motor is badly placed for maintenance; nevertheless the simplicity of the construction has led to its wide adoption even for some express locomotives. Improvements in output and design are continually being made. (*b*) The Buchli drive is typical of several devised to avoid the restrictions imposed by the nose-suspension drive particularly for high speeds. The motor is

completely spring-borne, together with its pinion and gear wheel. The essential elastic connection between the gear and driving wheels is obtained by means of a link mechanism. (*c*) The Westinghouse drive employs twin motors in a common housing, the two pinions engaging with a common gear wheel. The latter is within the driving wheels, and is carried on a 'quill' or hollow shaft, through which the driving axle is carried. The twin motors, pinion, gear and quill are all spring-borne. The drive is transmitted from the quill by means of a circular fork projection, which is attached to the ends of short, helical springs, the other ends of which are pinned to the driving-wheel spokes. (*d*) The jack-shaft drive with geared motor is still common. The gear-driven jack-shaft, carried on the frame, is coupled to the driving wheels by means of short side-rods. The gear wheel is provided with elastic connection between the toothed rim and the hub.

ELECTRIC TRACTION ON SUBURBAN RAILWAYS.

Much of the electrification work hitherto has been in connection with suburban railways around large cities like London, New York, and Paris. Such lines form a most promising field for the utilisation of electric power, as they usually carry heavy and dense traffic, a condition

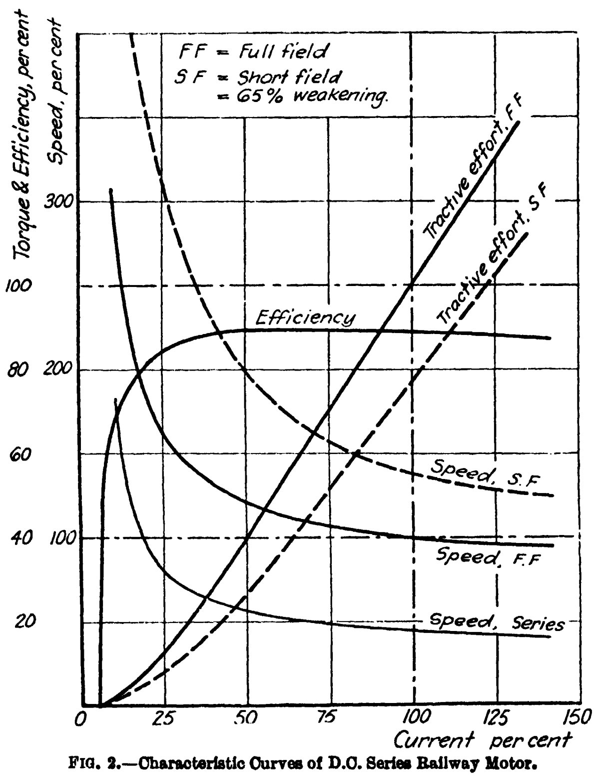


FIG. 2.—Characteristic Curves of D.C. Series Railway Motor.

very favourable to electric traction. Of course, on tube and similar underground railways anything but electric haulage in the light of modern requirements is unthinkable. For suburban and urban traffic the single-phase system is seldom considered unless some ulterior condition makes this advisable: in practically all cases the direct-current system is used, at voltages of 600 to 1,500, with third-rail collection. It is standard practice to generate three-phase energy

in the power stations, transforming to suitable pressures for transmission to the substations. Rotary converters have been widely used for A.C.-D.C. conversion, but are now being ousted by the mercury arc rectifier, often with grid control, and remotely controlled from a Central Supervisory Control Room.

For heavy electric traction on surburban lines using 600 volts and over, the tramway type of controller is replaced by 'multiple unit control,' consisting of small 'master-controllers' operating separate switches for establishing motor combinations and for cutting out resistances. Multicore cables between adjacent coaches enable any master controller to control the whole train. The great advantages of multiple unit control lie in the flexibility of train length, the provision of a motor capacity always proportioned to the load (for the maintenance of schedules under all load conditions), and the efficient utilisation of the available adhesion (for rapid acceleration).

For leading the current to the moving trains a third rail is provided for each track, the motor coaches being equipped with cast-iron contact shoes for current collection. In England, where the Board of Trade regulations require a very small drop in the earth return, it has been necessary on occasion to add an insulated fourth rail in order to avoid the drop in the running rails which elsewhere serve as a return. This may often be of considerable advantage in cases where track-circuiting is required for automatic train-signalling.

On most city and suburban electrified railways dealing with heavy traffic, automatic signalling is employed; indeed for many of the congested services it is essential. Where a fourth-rail return is used, the running rails are used to convey direct or alternating current for the track circuits. Where, however, the running rails carry the power current, alternating current alone may be employed for signalling. For automatic signalling the reader should consult Section XXXII, Part IV.

For lines with infrequent service the efficiency of the substations is comparatively low, while the running costs are high in consequence of the fact that the numerous substations must be capable of supplying the peak load. The transmission losses in the conductor rails are also considerable. In such cases automatic or semi-automatic substations, with higher line voltages, may be used.

Low-pressure traction motors are now standardised, as also is their control. Series-parallel arrangement of the motors of each bogie (of the motor coach) is universal practice. In special cases field-shunting is used, as, for instance, when trains make occasional runs to outlying districts.

Fig. 2 shows typical characteristic curves of a D.C. railway motor. Series-parallel control gives two economical speeds. By field weakening (usually by tappings) additional economical speeds can be obtained; the figures show the speed/current and torque/current characteristics for 50 per cent. field weakening.

Train units are usually made up of motor-coaches and trailer coaches. A common arrangement is one motor-coach semi-permanently coupled to one or more trailers. In such a case the end trailer is a driver trailer, and is provided with a driver's cabin with a master controller and the necessary apparatus for reversed operation. One, two, three, or more units may be coupled and controlled by one driver to form a train of any desired size according to traffic needs, and the same schedule may be maintained regardless of the train-length, since the power/weight ratio remains constant.

The following are brief details of some important suburban electrification systems:

Important Surburban Railway Electrification (All D.C.).

London.—(600 volts third or third and fourth rail.) Comprises a large network of surface and sub-surface and deep-level tubes; also main line termini. The Southern Region of British Railways has the largest suburban electrification in the world.

Manchester.—Bury line: 1,200 volts, third rail. Altrincham line: 1,500 volts, overhead conductor.

Paris.—(600 to 1,500 volts, third rail and overhead conductor.) City and suburban lines.

Berlin.—(750 volts, third rail.) Sub-surface and overhead railways.

New York.—(600 volts, third rail.) Tunnel, surface and elevated lines.

Melbourne and Sydney.—(1,500 volts, overhead line.) Suburban.

EQUIPMENT FOR MOTOR-COACH TRAINS.

Camshaft Control for Direct-Current Motors.

The main contactors in this system are operated by means of a camshaft under the control of a small pilot motor. The equipment includes the following essential features:—

(a) A camshaft composed of steel cams mounted on a mica-insulated steel shaft of square section. On rotating this shaft the cams close and open the main contactors in a definite predetermined sequence.

(b) A small pilot motor which drives the camshaft through suitable reduction gearing. This motor is controlled by the master controller and the position regulator, the latter being mounted on an extension to the camshaft itself.

(c) Two solenoid or cam-operated line breakers arranged in series to break the main circuit.

(d) All necessary auxiliary control gear, such as an electrically operated reverse, motor cut-out switch, overload relay, etc.

The closing of each contactor is effected against a strong spring by the rotation of the camshaft, so that the action is positive and the pressure constant, irrespective of the line voltage. Furthermore, the camshaft gives a definite mechanical interlock, controlling the sequence of operation of the contactors, and obviates the necessity for fitting electric interlocks. Since all current is broken on the line breakers, and the sequence of operation of all contactors is definite, the resistance contactors do not break any current, and are therefore not fitted with blowouts. The two contactors which deal with the operation of transition are called upon to break relatively small currents, and are fitted with blowouts. This arrangement effects a considerable reduction in the number of current-breaking contactors.

Electro-magnetic and Electro-pneumatic Control.

In the electro-magnetic system the contactors for effecting the motor and rheostat combinations are controlled electrically from the master controller by means of magnet coils energised from the line (if the voltage be below 600 volts), or from a low (*e.g.* 32-volt) circuit, comprising a dynamotor or motor-generator or battery. Interlocking is effected by auxiliary contacts on each contactor. After being closed the control of the contactor is transfered from a 'closing' to a 'holding' coil demanding less energy. Each contactor may require several pairs of interlock contacts, according to the system or design adopted. The direction of rotation of the motors is controlled by a separate reverser. The electro-pneumatic system embodies electrically-controlled needle valves for compressed air cylinders operating the contactors mechanically. Later developments involve the use of individually-operated pneumatic contactors effecting the motor combinations, the rheostats being cut out by means of a drum-type controller operated by a differential air motor. These systems are lighter, cheaper, and less complicated for automatic control. Pneumatic control has the advantage that the control voltage is low; a large mechanical force can be developed by a small and light compressed-air cylinder, and a spring relied upon for opening the contacts; the mechanical operations are performed independently of the line voltage, avoiding the tendency of electro-magnetic contactors (fed from the line voltage) to vibrate and weld together when the line voltage falls to a low value. The control is frequently automatic, the contactors being energised in sequence according to the operation of a current-limit relay.

The following details refer to typical motor-coach equipments with multiple unit control:

Details.	North-Eastern Region British Rlys.	London Transport Executive.	Bombay–Baroda Cent. Ind. Rly.	German State Rlys.
Motors per coach.	2	2	2	2
Line voltage	600	600	1,500	15,000 (1 ph.)
Motor h.p. (1 hr.)	145	235	275	330
Current collectors.	Shoes	Shoes	Pantographs	Pantographs
Gauge	4 ft. 8½ in.	4 ft. 8½ in.	5 ft. 6 in.	1·435 metres
Length over headstocks	55 ft. 0 in.	52 ft. 3¾ in.	68 ft. 0 in.	72 ft. 8 in.
Bogie centres	40 ft. 0 in.	33 ft. 6 in.	48 ft. 0 in.	50 ft. 10¼ in.
Bogie wheelbase	7 ft. 0 in.	6 ft. 3 in.	10 ft. 0 in.	11 ft. 9¼ in.
Wheel diam.	36 in.	31 in.	43 in.	38¼ in.
Weight empty, tons	34	27·4	69	47·9
Seating capacity	64	42	80	66
Control	electromagnetic*	electromagnetic*	electropneumatic* camshaft	electromagnetic* camshaft

* With automatic acceleration.

Metadyne Control.

A new method of traction motor control due to Pestarini and developed by Metropolitan-Vickers is the so-called *Metadyne* system,* employing a special rotating convertor interposed between the traction motors and the D.C. supply. This metadyne machine converts power at constant voltage, as obtained from the D.C. supply, to power at constant current as required by the traction motors. The advantage of the metadyne system lies in the maintenance of constant current in the motors throughout the accelerating period, so that the maximum possible use is

*** A good account of this novel system is given in the *Metropolitan-Vickers Gazette*, November 1938.**

made of the available adhesion. With standard rheostatic control the average torque during acceleration only reaches from 80 to 85 per cent. of the torque limit of adhesion. The metadyne system also dispenses with the use of heavy starting resistances and their associated control gear. Furthermore the equipment is readily adapted for regeneration and can be arranged to produce a retarding effort down to zero speed if required.

Control of Alternating-Current Traction Motors.

The majority of single-phase motors are of the neutralised series type, and the basic method of speed control is the variation of the motor terminal voltage. Since a main transformer is invariably used, the methods available are: (*a*) Transformer tappings, controlled by contactor gear. To avoid interrupting the motor supply, a second tapping must be closed before the first is opened. The section of the transformer between the two tappings is protected by an auxiliary choking or 'preventive' coil. (*b*) Transformer tappings with sliding contact switch, driven by a screwed shaft operated by a control motor from the master controller. (*c*) Transformer tappings with induction regulator. This allows of fewer tappings, but owing to its greater complication has found only a restricted application. The advantage is the absence of jerky control.

Polyphase induction motors with slip-ring rotors are controlled by means of rotor rheostats (usually of the liquid type) combined with cascade connection. Motors with cage rotors are started with reduced voltage (by auto-transformer), combined with pole-changing connections.

Direct-Current Traction Motors.

Direct-Current Series-Wound Traction Motors.

British Standard Specification for Traction Motors and Associated Rotating Electrical Machines. No. 173—1941 (Abstract).

Types of enclosure are: (1) Totally enclosed, there being no air circulation between inside and outside of case; (2) ventilated, having free circulation between inside and outside of case; (3) self-ventilated, having air-fan; and (4) forced ventilated (draught) type, in which air is forced through by external means.

Classes of Rating.—If excitation can be varied independently of armature current, the rating shall be associated with a stated condition of excitation and corresponding speed. (1) Nominal one-hour rating to be the h.p. output at motor shaft, at the rated voltage and constant current, which the motor can produce for one hour, the limit of temperature-rise measured by resistance being 100° C. for Class A and 120° C. for Class B insulation, and 90° C. on commutator. (2) Continuous rating is defined as the output at the motor shaft after working continuously with covers, etc., arranged as in service, the temperature-rise measured by thermometer not to exceed 85° C. on commutator, and on insulated windings measured by resistance 85° C. and 120° C. for Class A and Class B insulation respectively. Class A includes enamelled wire and cotton, silk, paper, etc., when impregnated or oil immersed. Class B, asbestos and built-up and cemented mica insulation. In ventilated motors, the rating shall be the output at rated motor voltage; in totally enclosed motors the output at half the rated motor voltage.

The motor must be run in both directions with fixed brush position under specified test conditions. The statement of efficiency to be made on result of direct measurement corrected to 75° C. and not on the basis of addition of losses. The new machine must withstand, at the conclusion of the one-hour test, an alternating voltage of 1,500 plus twice the line voltage applied for one minute, between the windings and frame. The insulation resistance in megohms, when the high-voltage test is applied, to be not less than line volts/(1,000 + Rated B.H.P.). The specification lays down methods of carrying out performance and voltage tests, and the information as regards type, voltage, speeds, weight, and characteristics, to be supplied by the maker.

SECTION XXXII

PART IV

RAILWAY SIGNALLING.

(By O. S. Nock, B.Sc., A.M.I.C.E., M.I.Mech.E., M.I.R.S.E.)

Rapid and far-reaching developments in this branch of railway engineering have taken place in recent years. The safety of train operation, once practically the sole consideration, has now been brought to such a degree as to be taken for granted in the study of any new signalling scheme. While safety in operation remains the basic and essential factor, signalling is now recognised as one of the principal aids towards the maintenance of a frequent service of fast trains in congested areas, and in all localities as a potent instrument for securing the maximum utilisation from any stretch of line. In Great Britain the signalling is still mechanically controlled and operated, except in a relatively few districts where the traffic is heavy; but in what may be termed mechanical territory the use of electrical apparatus to supplement existing plant is being greatly increased. Large terminal stations, and important centres of traffic have been equipped with complete installations of all-electric, or electro-pneumatic signalling, and it is in connection with these plants that rapid developments were taking place until the outbreak of war in 1939, when the likelihood of damage from aerial attack made it necessary to postpone the completion of certain schemes of signalling concentration, and to defer consideration of others.

It is now generally recognised that if control of train movements in a busy and complicated area is concentrated in one signal box better regulation of traffic is likely to be obtained than if a district superintendent has to co-ordinate the work of, maybe, five or six signal boxes, each of which is concerned with one particular section of the working. But while the principle of concentrating control in a single box can be readily accepted, very great care must be taken to relieve the signalmen, or traffic controllers, of all but the most essential duties. Manual work, and routine tasks, must be reduced to the very minimum so that the men are free to devote their time to the regulation of traffic movements. It is to this end that many recent developments in railway signalling practice have been directed. But, although the means that have lately been employed are in many cases novel, all, without exception, continue to be based upon the fundamental principles finally established some seventy years ago: the first of these—a principle of operation—is the use of the Block System; the second—a principle of engineering design—is that, in the event of any failure of mechanism or power supply, the signals immediately concerned shall display the danger indication.

THE BLOCK SYSTEM AND SIGNAL ASPECTS.

The principle of the block system is illustrated in fig. 1. A line of railway is divided into a number of block sections, and the dividing points between adjacent blocks, for example, the locations A, B and C, are known as block posts. The system is simply that only one train is permitted to be in a block at any one time. The diagram fig. 1 (*a*) relates to one direction of running, and it will be noted that there is a signal at the entrance to each block; such are termed 'absolute stop' signals, and must on no account be passed if displaying the danger indication. To permit of high speed running it is necessary to give the driver of a train ample warning that he is approaching an 'absolute stop' signal; accordingly what are termed 'distant' signals are installed, and these are so sited that there is full braking distance for the fastest trains between the 'distant' and the 'stop' signal. A 'distant' signal can be passed when it is displaying the warning indication, but a driver so doing must take immediate steps to reduce speed so as to be able to bring his train to a stand at the 'stop' signal. Semaphore 'distant' signals are distinguished from 'absolute stop' signals by having the blade painted yellow with a black chevron, instead of red with a white stripe; the ends of 'distant' signal arms are fishtailed.

To assist in the handling of traffic block posts are equipped with two or more 'stop' signals. The first one encountered by a train arriving at a block post is usually the 'home,' and the signalman is not permitted to accept a train from the preceding signal box unless the line is clear, not

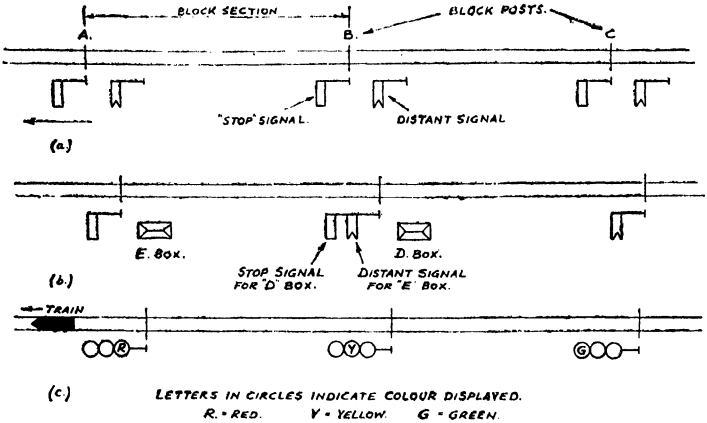


FIG. 1.—Principle of the Block System.

merely up to the home signal but for a distance of 440 yds. beyond it in addition. This rule affords a safeguard should an arriving train overrun the home signal. Supposing the block section ahead is occupied, if the block post in question is equipped with a second 'stop' signal, termed the starting signal, a train could be allowed to draw forward to that signal and await the clearance of the block ahead. If the starting signal is so located that the tail end of the train is more than 440 yds. from the 'home' signal, a second train can be accepted from the preceding block post before the first train has moved away, and traffic can thus be kept moving more freely. In larger and more complicated layouts there are outer and inner 'home' signals, followed by a 'starter,' and lastly an 'advanced starter'—four 'stop' signals in all. The 'distant' signal for the particular block post cannot be lowered unless all the 'stop' signals are in the clear position.

At larger stations equipped with mechanical signalling, there are usually two signal boxes, one controlling each end of the layout. In such cases, or any similar locations where two block posts are close together, it is often convenient to mount the 'distant' signal for the second box on the same mast as one of the 'stop' signals for the first, as shown in fig. 1 (*b*). The composite signal shown at D can then display any one of three indications: both arms 'on,' signifying stop; the upper arm 'off' and the lower arm 'on,' signifying 'caution'; both arms 'off,' signifying 'clear.' These indications, together with the corresponding lights for night working, are shown in fig. 2. The concurrent use, in different parts of Great Britain, of semaphore signals working

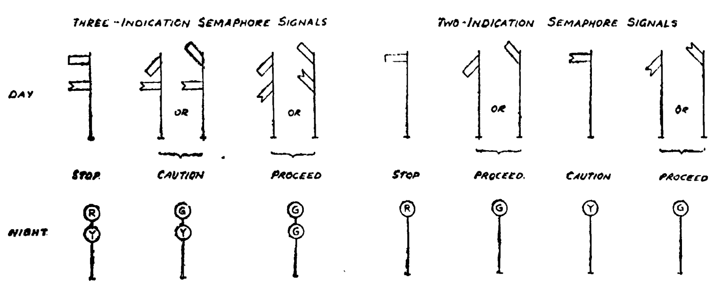


FIG. 2.—Semaphore Signal Aspects.

in the upper and the lower quadrant will be mentioned later. With semaphore signals having only two positions, 'on' and 'off,' it is necessary to use two arms in order to provide three indications from a single assembly, as shown in fig. 2; the day colour light signal, however, affords very simple means of providing three indications, each one of which consists of but one light—

red, yellow, green. Fig. 1 (c) shows the application of multi-aspect colour light signals to block working, where each of the signals constitutes, in effect, a block post, and each can display three indications. With signals arranged as in fig. 1 (c) the spacing requires to be designed to provide a uniform headway between succeeding trains; this problem, and the use of four-aspect colour light signals was discussed in *The Engineer*, December 8, 1939.

On routes where a few express passenger trains are run at average speeds considerably higher than that of the remaining passenger services, such as the 'Cheltenham Flyer' and the 'Bristolian' of the former G.W.R., and the streamlined trains of the former L.N.E.R., special attention has to be paid to the signalling arrangements to provide adequate braking distance before a signal displaying the danger indication is reached. These trains run at average speeds of over 80 m.p.h. for an hour on end, or more. In multi-aspect colour light territory the use of four indication signals provided a satisfactory solution, but over the long mileages equipped with manual block signalling the L.N.E.R. treated each individual location on its merits. In many places outer distant signals were installed. These were of the colour light type and displayed a 'double-yellow' when the ordinary semaphore distant was in the 'caution' position. Ordinary traffic could afford to ignore the outer distant signal and maintain full speed until the indication displayed by the inner distant was seen. On the G.W.R. special block regulations were in force for signalling the high speed trains, whereby a succession of block sections were required to be clear before the signals preceding such a group of sections was pulled to the clear position.

Apparatus for Block Working.

In the operation of the block system communication between adjacent signal boxes plays a very important part. Before the starting signal at one box can be 'cleared,' and a train allowed to proceed to the next box, messages must be exchanged between the two signalmen concerned to ascertain that the line is clear. These messages are conveyed through the use of a standardised code of bell signals. Thirty-two different messages are included in the Railway Clearing House standard code, and in addition to messages required for the ordinary working of traffic, describing the various classes of trains, the code includes messages for use in a variety of emergencies, such as a train becoming divided, runaway vehicles, obstruction on the line, and a call for fog signalmen. The bells, plungers or tapper keys are usually incorporated in the Block Instruments, which indicate the occupancy or otherwise of a particular block section. With manual block working, such as discussed so far, the line is normally closed, and requires to be opened for the passage of a train. A variety of block instruments is in use in Great Britain; one of the commonest forms used on double lines is the three-position, single needle instrument, and in this the needle normally points to the position LINE BLOCKED. These instruments are operated in pairs, one at each end of a block section. The signalman at the entering end, after giving the 'call attention' signal on the bell code, sends the code 'Is line clear for . . .'; the signalman at the leaving end switches his instrument so that the needle indicator points to LINE CLEAR, and this indication is repeated in the signal box at the entering end of the section. When the train passes the starting signal at the entering end the signalman there sends TRAIN ENTERING SECTION on the bell code; the signalman at the leaving end switches his instrument so that the needle points to the third position TRAIN ON LINE and this indication is repeated in the signal box at the entering end. On a double line of railway equipped with separate block instruments for each line working in the manner described, three line wires are required between adjacent signal boxes.

The above is a necessarily abbreviated description of the procedure taken prior to the lowering of the signals themselves, but it should be emphasised that the responsibility for correct operation of the instruments rests upon the individual signalman, just as the responsibility for correct interpretation of the signals rests upon the locomotive drivers. On some lines certain electrical apparatus has been introduced to safeguard against irregularities in block working by the signalmen, and where mechanical signalling is concerned the system of 'Lock and Block' has been widely used. In applying this system to meet particular requirements the practice of pre-grouping railway companies differed somewhat in detail; but the broad principle is the same, namely, that the starting signal for entering a block section is so interlocked with the block instrument that the signal lever cannot be pulled unless the signalman at the leaving end of the section has accepted the train. Once this has been done the accepting signalman cannot again give LINE CLEAR, and so accept a second train until the first one has arrived and cleared the controlling track circuit, or electric treadle, and the signals have been put to danger behind it. This electric inter-locking thus provides a positive safeguard against irregularities in the block working.

On single lines of railway the observance of correct block working is of vital importance, and on entering a single line block section the driver is handed a 'token,' without which he is not permitted to proceed, even though the starting signal might have been pulled to the 'clear' position. This 'token,' which on some railways takes the form of a steel tablet, and on others the form of a staff, is a positive assurance to the driver that the section is clear. The block instruments for single line working are of special construction and a number of tokens for a particular section are kept in the instruments at each end. Before a token can be extracted co-operative action between the signalmen at each end of the section is required, and once a token is 'out' both instruments are electrically locked until that token is inserted again in one or other of the instruments. This electric interlocking between the instruments prevents more than one token being out at a time, and thus the possession of a token by a driver is at the same time an assurance, as well as his authority to proceed. For ease in handling, the token is carried in a leather pouch,

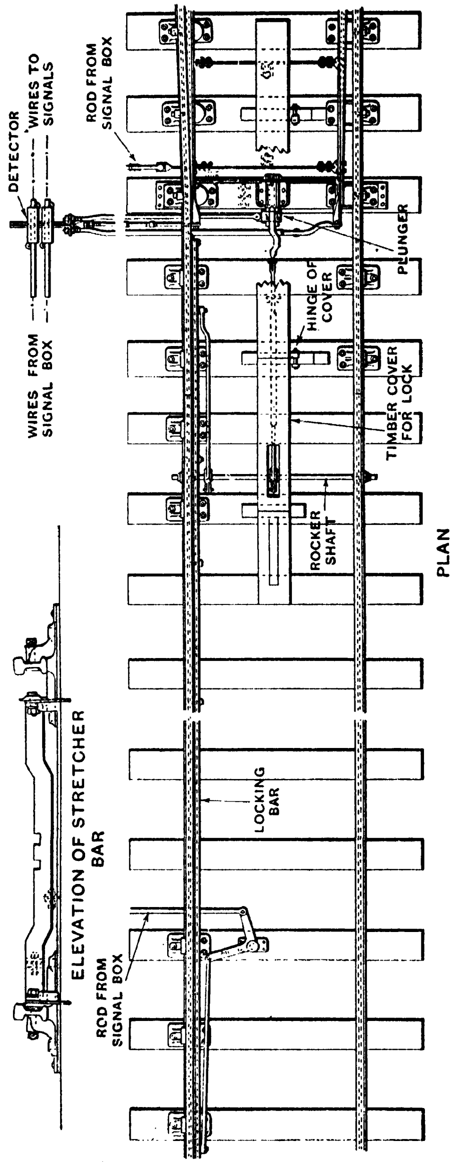


FIG. 3.—Standard Facing Point Layout.

fitted with a large diameter hoop; with this attachment the token can be collected from a lineside standard without the necessity for stopping. On lines where express services are operated over single-line sections mechanical token-exchanging apparatus is used, whereby the token for one section is released and that for the next section collected at speeds up to 60 m.p.h. In the ordinary way 15 m.p.h. is about the limit for hand exchanging of tokens.

Apparatus for Mechanical Signalling.

(a) *Signals.*—Two forms of semaphore signal are in use on the British Railways. The older form, working in the lower quadrant, is standard on the WESTERN REGION; but, while still widely used elsewhere it was in course of replacement by the newer form operating in the upper quadrant, on the former L.M.S.R., the L.N.E.R. and the S.R. Under nationalisation a standard will no doubt be decided upon, in due course, for the entire British railway system. While the indications displayed by these two forms is different the fundamental principle of operation is the same. In both the arm is normally in the horizontal position, denoting danger, and the wire leading from the signal to the cabin requires to be pulled in order to move the arm to the clear position. In the case of upper quadrant signals the weight of the blade itself restores the arm to the danger position when the wire is released; with lower quadrant signals the spectacle carrying the glasses is designed to provide a counter-weight so as to restore the arm to the horizontal position. The actuating wire in the case of a lower-quadrant signal is attached to a reversing lever, which translates the pull in the wire to an upward thrust in the signal down-rod, so pushing the arm into the familiar clear position, pointing downwards at about 45°. The wire between the cabin and the signal post is kept taut by the balance weight at the foot of the post.

The difference between 'absolute stop,' and 'distant' signals, both in function and appearance, has already been discussed. At diverging junctions it is the usual practice to instal a separate 'stop' signal for each route. The arms are grouped together on short posts, or 'dolls,' of varying heights, the tallest posts relating to the most important routes. At large centres of traffic the number of doll posts is such as to require structures of considerable size for their support, particularly where a number of running lines has to be spanned. In addition to the main running signals, the working of which has already been described, shorter arms with spectacle glasses of reduced size are used for the regulation of shunting movements, and, at large stations, for instructing drivers to draw ahead cautiously past a 'stop' signal at danger. These special arrangements are made to facilitate traffic working, as, for example, where a driver is instructed to draw into a long platform road, part of which is already occupied by another train.

(b) *Point Operation.*—The points at a diverging junction are termed 'facing,' and on passenger lines the blade on the closed switch side must be bolted against the stock rail before the signals can be lowered for the passage of a train. Fig. 3 illustrates a typical facing point layout. There are two rods from the signal box, one to operate the bolt, and the other to throw the switches themselves. The bolt lever stands normally pulled over in the cabin, so that the plunger is inserted and the points are locked. Before the points can be thrown the bolt lever must be restored to normal; this action moves the rod connected to the bell crank on the third sleeper from the left in fig. 3, and through the agency of the locking bar oscillates the rocker shaft and withdraws the plunger from one of the notches in the lock stretcher bar. The points can then be thrown, after which the bolt lever is pulled again and the plunger enters the second notch in the lock stretcher bar.

The locking bar is a member usually between 40 and 50 ft. long, of angle or tee section, and supported on a series of rocking arms. The level of the bar is normally just below the flanges of the wheels and in that position the arms are lying at an angle of approximately 45° to the vertical. As shown in fig. 3 the plunger is out, and the arms are towards the left. When the bolt lever is pulled the movement of the bell crank pulls the bar from the left to the right and the rocking arms move through an angle of 90°. In so doing the bar itself is moved from flange level up to rail level in passing from one extreme position to the other. This feature provides an important safeguard in that the bar cannot be raised when a vehicle is standing on, or a train is passing over, the particular piece of track. As the bar cannot be moved a signalman is prevented from inadvertently throwing the points under a train. The position of both switch blades, and the position of the locking plunger must be proved correct by the detector before the signal wires can be pulled. Fig. 3 shows two wires for signals; the lock notches in the detector blades are so arranged that only the wire to the signal relating to the right hand road is freed in the position shown. At converging junctions the points are termed 'trailing,' and there is no need for the switch blades to be locked.

(c) *Signal Box Equipment.*—The cross-section of a typical signal box interior is shown in fig. 4. The levers for point and signal operation are grouped in a frame supported on the cabin structure. Connections for both points and signals are shown. The points are operated direct from rodding coupled to the lever tail, whereas the signals may be operated by wire attached directly to the lever tail, or through the agency of a gain-stroke wheel as illustrated. This latter arrangement is used for signals that are at a very long distance from the box. In such cases a wire regulator is used to compensate for expansion or contraction of the wire due to atmospheric conditions. For signals located at shorter distances from the box a turnbuckle in the wire connection is usually all that is necessary for compensation. The necessary adjustments are made, as required, by the linesman. In the case of point operation automatic compensators are fitted in all lengths of rodding exceeding about 120 ft.

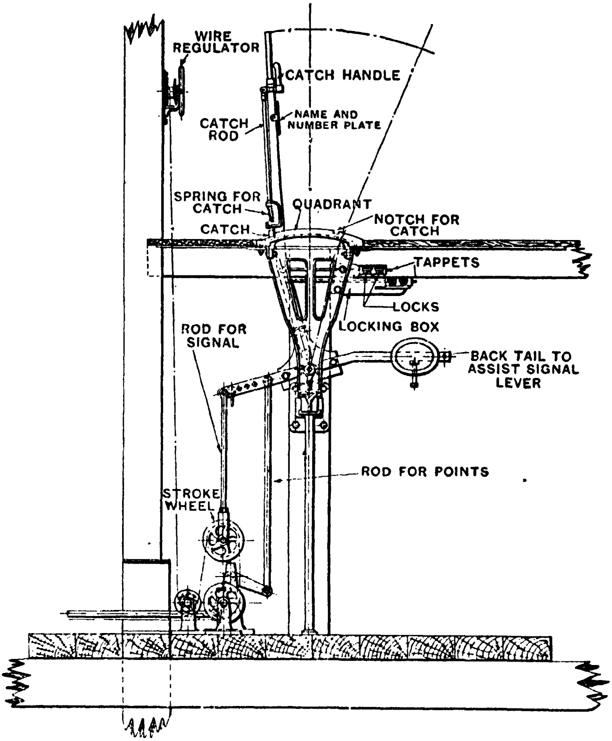


FIG. 4.—Interior of Signal Box.

To avoid any chance of signalmen pulling wrong signal levers, or setting points so that trains following the routes so established would collide, it is necessary for all signal and point levers to

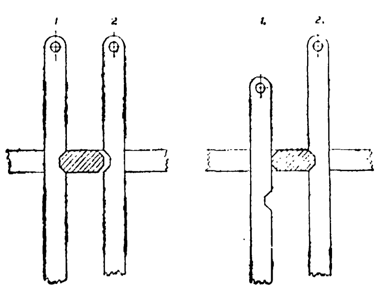


FIG. 5.—Principles of Tappet Locking.

be interlocked with one another. The locking plungers, or tappets, which are attached direct to the levers in the apparatus shown in fig. 4, engage with locking dogs sliding in troughs running the length of the lever frame. The tappets are notched to suit the bevelled ends of the locking dogs, so that when a lever is pulled any dog which is engaged with the particular tappet is forced out of engagement and slides along the longitudinal trough, and its opposite end maybe engages with the tappet of another lever. The principle of tappet locking is illustrated in fig. 5. In designing an interlocking mechanism all the traffic movements in a station or junction must be considered, and a chart prepared showing the conditions for every signal and point lever in the frame. With a junction signal, for example, a study of the layout will show that before this can be pulled to the clear position certain points must be lying in their normal position, and others in the reverse; all signals relating to conflicting routes must be locked at danger by the pulling of this junction signal lever. From the table of locking thus prepared the designer can work out the interlocking mechanism.

Application of Power to Railway Signalling.

(*a*) *Electrical Apparatus in Mechanical Territory.*—In the earliest British installations of power signalling, completed upwards of fifty years ago, power was used solely for the operation of points and semaphore signals. By thus relieving the signalmen of heavy manual work more units could be controlled by one man, and lever operation was quicker. The systems then introduced, and still used to-day, fall into two broad classes: one in which the power is supplied by electric motors and the other using compressed-air cylinders with electrically-controlled valves. The early installations provided no additional safeguards in operation, since no track circuiting was included. Installed at large centres of traffic, such as Bolton, Newcastle, Glasgow Central (Caledonian Railway) and elsewhere, these early plants were 'all-power'; while modern developments in this direction have been rapid, perhaps even more interesting has been the steady infiltration of power signalling practice into what would normally be regarded still as manual signalling territory. To-day, modern signalling practice in Great Britain is becoming largely a blend of mechanical and electrical methods, except where the density of traffic warrants the use of 'all-power' plants—either electro-pneumatic, or all-electric.

To meet the needs of high speed traffic the tendency is for 'distant' signals to be placed farther from the cabin. In the Western Region it is now usual for semaphore 'distants' to be operated by electric motors; elsewhere, in locations where 'distants' have been re-sited, opportunity has been taken to replace the semaphores by day colour light signals. Thus not only has the heavy pull always associated with wire-worked 'distant' signals been eliminated, but in most cases the sighting has been greatly improved from the locomotive driver's point of view. Improvements in the operation of points have developed concurrently. The limit for rod-worked points is 350 yds. from the signal box, and in earlier days it was usual to instal separate cabins for operating the outlying crossovers and siding connections. In localities where traffic conditions do not justify full power working, the remote control of outlying points is now conveniently arranged by using electric point motors, either battery-fed or supplied at 110 volt D.C. by hand generator. Such outlying points can thus be operated from the main signal box in the district, affording that unity of control which is desirable for smooth regulation of traffic working. Remote control systems for point operation proved extremely valuable during the war of 1939–45, when additional facilities were required on many lines carrying war traffic to the Channel ports. Some typical examples of their application were described in *The Engineer* for July 3, 1942.

Remotely controlled points, if not actually out of sight of the signal box, are far beyond the distance at which the man can observe the passage of trains, and track circuiting is essential to hold the points locked while vehicles are passing. There is a tendency to associate track circuiting primarily with all-power installations, and sections of purely automatic signalling—two applications which will be discussed in greater detail later. But its value in manual territory is also very great, for detecting the presence of trains or vehicles on sections of line not easily visible from the cabin concerned, and in providing track circuit locking for points. One of the most extensive early applications of track circuiting in conjunction with mechanicalsignalling was made on the former Midland Railway (now the London Midland Region) after the disaster near Hawes Junction in 1910, caused at a very busy time by a signalman forgetting the whereabouts of two light engines, and lowering the signals for an express to pass over the very track where the two engines were standing. (See *The Engineer*, June 12, 1942.) On the former S.R. the electrification of the main lines from London to the south coast was accompanied by some extensive resignalling works, wherein many of the larger intermediate and terminal stations have been equipped with complete track circuiting, while mechanical operation of the signals and points has been retained. Bognor Regis provides a typical example. The traffic, though considerable, did not warrant the higher cost of complete power working with day colour light signals, but with track-circuiting throughout the station yard and its approaches all the operating safeguards of a modern all-power interlocking are provided. The full-size levers for mechanical operation of the semaphore signals and points are equipped with electric locks, so that points cannot be moved if a train is standing upon, or passing over them, neither can signals be cleared if the track ahead is occupied by another train.

(*b*) *Track Circuiting.*—With examples such as the foregoing in mind, together with less extensive applications in mechanical territory, and its absolute necessity in complete power schemes,

the track circuit may well be called the basis of all modern signalling. In fact, hardly any signalling scheme to-day, even in mechanical territory, is installed without the inclusion of some track circuits. The principle of operation is illustrated in fig. 6. The section of line forming the track circuit is insulated at the rail joints from the adjoining sections, and current is fed to the running rails. If the section is clear, the relay is energised, but if a train or vehicle is standing, or passing, current from the battery takes the path of least resistance, through the wheels and axles, and the relay is shunted. The diagram in fig. 6 shows a track circuit in its simplest form, fed by direct current, though from a study of it some of the major considerations in track circuit design will be appreciated.

Conditions experienced in the performance of track circuits are liable to wide variation. When a train, or vehicle, is standing on a track circuit, current flows through a series of parallel paths from rail to rail; the combined resistance of these paths varies considerably, being theoretically

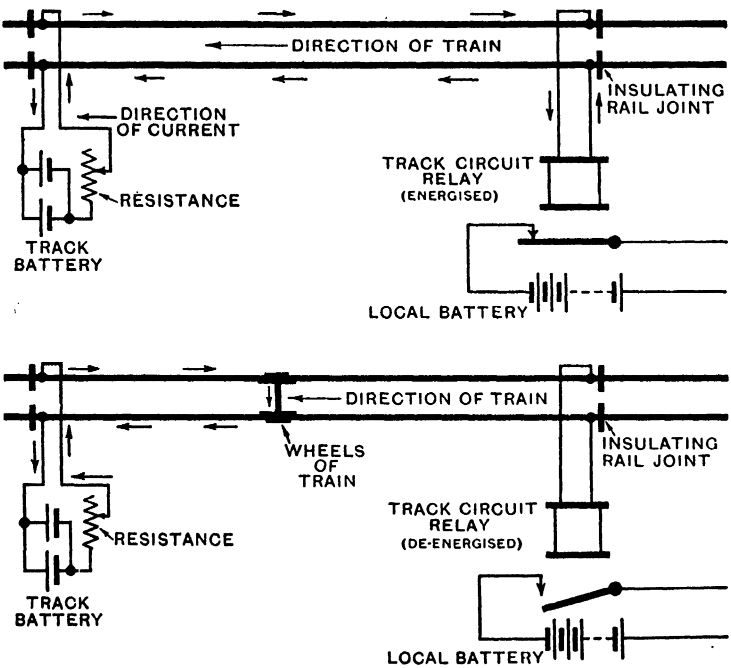


FIG. 6.—Principle of Track Circuit.

highest, of course, for a single pair of wheels and lowest for a long train. Thus it is possible that with a single vehicle—a railcar for example—the shunt resistance would be so high as to prevent the track circuit relay from becoming de-energised. Again, a certain amount of leakage always takes place through the ballast; such leakage varies with the nature of the ballast itself, but in any one locality track circuits may be particularly susceptible to changes in the weather. After heavy rain track, unless very well drained, is apt to hold moisture to a certain extent, and this will, of course, increase the leakage of current from rail to rail. In tunnels where the adverse combination of a wet formation, and deposits of soot and cinders may cause unusually high ballast leakage, it is sometimes very difficult to secure good track circuit performance. The problem of design largely centres upon the value of the current to be fed to the rails so that the relay shall be shunted under any circumstances when the track is occupied, and shall remain energised when the track is clear, even under the worst conditions of leakage through the ballast.

The diagram in fig. 6 illustrates a battery-fed D.C. track circuit. It is, however, not desirable to use D.C. track circuits on electrified roads, even though the traction return is carried in a fourth

rail owing to the possibility of interference from stray currents. In such conditions A.C. track circuits are used. Where the third rail system is used the track circuit insulations would interfere with the traction return current, and to provide a path for this latter impedance bonds are installed at the block joints. The principle of the impedance bond is shown in fig. 7. A coil con-

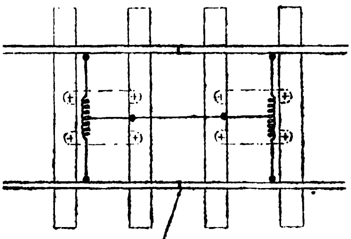


FIG. 7.—Impedance Bond Layout.

sisting of a few turns of heavy copper strip is wound on a laminated iron core, and is connected across the running rails; the A.C. track circuit current tending to flow from rail to rail is prevented by the impedance of the coil, whereas the resistance to the passage of the D.C. traction current is very low. The mid-points of the coils of adjacent bonds are connected as shown so as to provide a path for the traction return current.

ALL-POWER INTERLOCKINGS.

(*a*) *Systems of Operation.*—Developments in electrical engineering practice in recent years have led to a distinct breakaway from traditional interlocking methods in a number of noteworthy British signalling installations put into service shortly before the outbreak of war in 1939. There are now three distinct systems of power interlocking in use in Great Britain:

1. Power frame with mechanical interlocking between levers.
2. Power frame with electric interlocking between levers.
3. Control panel with non-interlocked thumb switches and relay interlocking.

The first type is no longer being used for new works except on the railways of the London Transport Executive. In extensive installations, particularly where a large number of shunting movements required to be signalled, the mechanical interlocking mechanisms grew so complicated, and so expensive to alter when modified traffic arrangements compelled a degree of resignalling, that the system of all-electric interlocking was developed. Instead of a mechanical interlocking mechanism, electro-magnets, additional to those already in use on power frames for track circuit and indication locking, were used for providing the interlocking between levers. An alteration could be made very simply with this type of frame, by changing a few contact bands and carrying out the necessary modification to the wiring. From the signalman's point of view the operation of an all-electric frame is the same as with one of the earlier types of power frame, in that if a signal or a particular pair of points cannot be used owing to prevalent traffic conditions the lever cannot be pulled. A cross-section of a typical signal box with this type of frame is shown in fig. 8.

It is in this respect that the modern control panel constitutes so radical a departure from previous standard practice. The interlocking between signals and points is still provided, as indeed it is essential it should be, but entirely by means of relays; the controlling thumb-switches are free to be moved at any time, and if the particular function which the signalman intends to operate happens to be locked the signal or point machine does not respond to the movement of the thumb switch. Experience with the large and busy relay interlockings in the North Eastern Area of the L.N.E.R. (now the North Eastern Region of British Railways) at Thirsk, Leeds, Hull and Northallerton has shown that no operational disadvantages have arisen from the use of these control panels, and the system may now be considered as well established in British railway practice. One of the most important features associated with relay interlocking is the compactness of the control machine; generally it may be said that a power-frame affording the same facilities will be 2½ to 3 times longer than a control panel. With manual work reduced to no more than the operation of thumb switches the signalman can devote almost his whole attention to the direction of traffic. At Northallerton, for example, one man controls the working over no less than 129 routes.

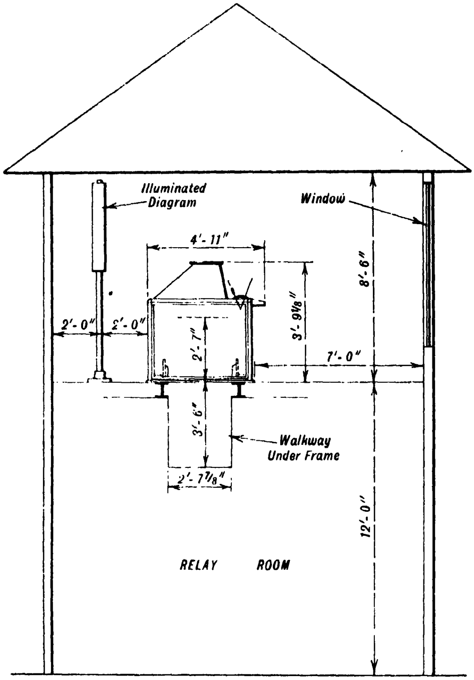


FIG. 8.—Typical Signal Box.

(b) *Control of Colour Light Signals.*—In modern all-power interlockings, whether controlled from power frames or a panel, the use of day colour light signals has become standard practice on all British railways. In a number of installations the so-called 'Searchlight' type of signal has been used, in which there is a single light unit, and the indications displayed—red, yellow, or green—are provided by the interposition of coloured roundels between the lamp and the lens, the roundels being carried on the vane spindle of a three-position relay housed in the signal. Generally, however, the multiple-lens type of signal is favoured, and a description of this is given later. Purely automatic signals are controlled by the relays of track circuits concerned, but in the case of signals operated from a power frame, or other form of control machine, a safety feature is included to ensure that after the passage of the train the signal lever is put to danger, before the signal can be cleared a second time. This fundamental principle of power signalling is illustrated in fig. 9. The essence of the arrangement is the 'stick relay,' which is energised by a circuit passing through a contact on the track relay made when that relay is energised, and a contact made when the signal lever is normal. When the lever is reversed, and the direct feed to the stick relay coil is broken, that relay is held energised, or 'stuck,' as it is termed, by a feed over its own front contact. When the train passes the signal and enters tract circuit 'A' the feed to the stick relay is cut, and owing to the nature of the circuit it cannot be energised again until the signal lever is restored to normal.

(c) *Illuminated Diagrams.*—In all modern schemes of power signalling the illuminated cabin diagram plays a very important part. With the control of large areas concentrated in one signal box, considerable sections are necessarily out of sight of the signalman. But even with the lines in the immediate vicinity of the box it is essential to provide the man with exact information as to the occupancy of all the track circuits. Where a power frame is used the illuminated diagram is a comprehensive track circuit indicator and little more; but in the latest relay interlockings its functions have been increased. In certain cases the thumb-switches have been mouoted on their correct geographical positions on the diagram, and in other instances where route-switching

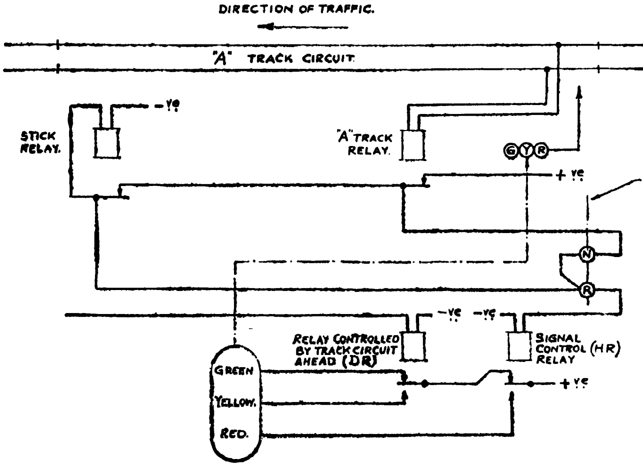


FIG. 9.—Stick Circuit for Power Signal Control.

is used additional lights are provided to show the signalman the routes which have been set up and signalled. The thumb switches are mounted on a separate keyboard below the diagram, and when a particular route has been set up its extent is shown on the diagram in a series of white lights. Thus the signalman is provided with immediate evidence of whether the various point motors and signals have responded to the turning of the thumb switch he has operated.

Apparatus for Power Signalling.

(a) *Point Operating Gear.*—Any mechanism for power operation of points, whether all-electric, or electro-pneumatic, must include all the functions of a mechanical layout as illustrated in fig. 3, p. 598. Track circuiting, of course, obviates the need for a facing point lock bar, and the actions of the bolt lever and the point lever are usually combined in one mechanism. The general tendency nowadays is to mount all apparatus in the 'six-foot,' rather than the 'four-foot' way, and modern types of all-electric point machine include the electric motor, facing point lock, point detector and motor control gear within a single casing, which can be mounted in the middle of the 'six-foot' way. Equally comprehensive mechanisms operated by a compressed air cylinder, are used in electro-pneumatic installations. In large and complicated layouts, owing to the presence of crossings, slip roads, and fixed structures it is not always possible to mount point machines adjacent to the switches to be operated, and then recourse has to be made to combined facing point locks and electric detectors mounted in the 'four-foot,' while the points themselves and the facing point lock plungers are operated by rodding connected to point machines fixed in the nearest convenient place.

Where electric point machines are relatively near to the signal box they are controlled by direct wires from contacts on the signal levers. In more distant locations, to avoid the need for carrying the heavy currents over, maybe, a mile of line wires, contactors are installed adjacent to the point machines, and, as the currents necessary to operate them are small, losses in transmission over these longer distances are avoided. The system of all-electric operation most generally favoured nowadays is 110 volts D.C., supplied from storage batteries in the cabin, keeping the local authority's supply, fed through suitable rectifiers, as a standby. In electro-pneumatic installations modern practice is to use a pressure of about 80 lb. per sq. in. The method of control varies. On the railways of the London Transport Executive, air is maintained in the cylinders after the completion of the stroke, the feed and exhaust being controlled by an electro-magnetically operated needle valve. In the latest installations put into commission before the war of 1939–45 in the North-Eastern Area of the L.N.E.R. a 'cut-off' type of valve is used, in which the air supply is cut off as soon as the points are thrown and locked.

(*b*) *Colour Light Signals.*—The light-unit of a modern colour light signal is shown in fig. 10. With a standard lamp, double-filament (See B.S.I. Spec. 469), such a lens combination gives a concentrated beam of light, and the indications can be clearly seen up to a mile away in brilliant sunshine. The 'spread' of the beam is very small, not more than 2½°, and this feature introduces certain problems in design and installation. When outside the beam the indication displayed by the signal is practically invisible in daylight, and to meet the frequent case of multiple-unit electric trains drawing close up to such signals, side lights incorporating lenses 1¾ ins. diameter are provided at a convenient angle for observation from the motorman's cabin.

In earlier installations of colour light signals, the subsidiary units, for controlling shunting, and draw-ahead movements, were also multi-lens day colour light signals. though having a short range and lenses of considerably smaller diameter. More recently, however, subsidiary signals displaying indications totally different from those of the main running signals have been installed.

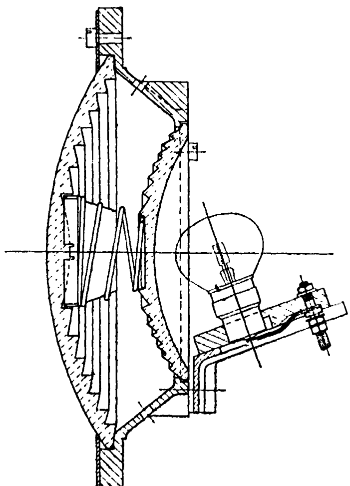


FIG. 10.—Lens Arrangement of a Colour Light Signal.

In the Southern Region solenoid-operated disc signals are now used ; to provide the same indication by night and day the discs are flood-lighted, thus contrasting to the discs in mechanical territory which have spectacle glasses in the discs and display reduced-size coloured lights at night. On the former L.N.E.R. and L.M. & S.R., what are termed 'position light' signals were adopted for subsidiary purposes. In the case of ground shunt signals two white lights displayed horizontally indicate 'stop,' and two white lights displayed diagonally indicate 'proceed.' 'Draw-ahead' signals are normally extinguished ; when the main signal is 'red,' and a train is required to draw forward, two white lights are displayed diagonally. With all forms of colour-light, or position light signal it is usually necessary to reduce the voltage at night ; otherwise the indications would be too brilliant and would cause dazzle.

(c) *Route Indicators.*—While it is generally desirable with any form of signalling to reduce as much as possible the number of semaphore arms, or coloured lights displayed, the long distance from which colour-light signals can be sighted makes this desideratum particularly necessary with modern power signalling. In the approach to large stations speed is relatively low and indicators showing the platform to which a train is routed have been successfully employed for many years. There are various forms in regular use ranging from the electrically-driven roller-blind type, as installed at King's Cross Station, to the theatre-sign indicators used in recent installations on the former S.R. The principle of the latter type is illustrated in fig. 11, and with this the indication is the same by day and night. Although such indicators show up well, the letter or figure displayed could not be read by the driver approaching a high-speed junction—at any rate not in time to take the necessary action—and a different form of indicator is essential. To meet such requirements the illuminated directional indicator has been developed, the principle

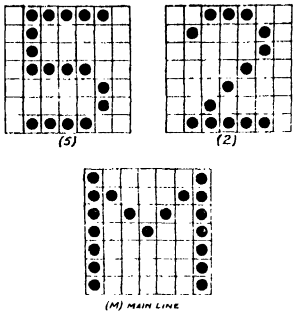


FIG. 11.—Theatre Sign Indicator.

of which is illustrated in fig. 12. When a train is signalled along the main line, and no reduction in speed is required the directional sign is not lighted at all; for diverging movements the line of white lights corresponding to the route to be followed is illuminated, and appearing above the coloured lights of the signal itself gives a striking indication to the driver of an approaching train.

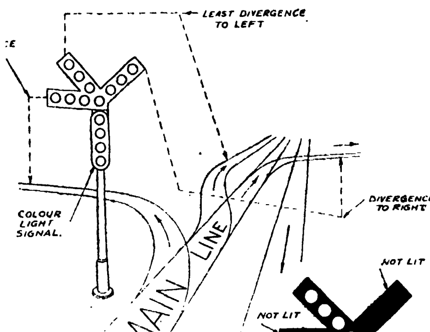


FIG. 12.—Principle of Illuminated Directional Sign Indication.

(*d*) *Structures for Colour Light Signals.*—The concentrated nature of the light beam from a day colour light signal introduces certain unusual problems in the design of structures carrying groups of signals, or signals associated with route indicators or the junction directional signs just described. The signals require to be mounted as close as possible to the running road to which they refer, and in busy areas this often necessitates the use of cantilevers with a considerable overhang. Such structures must be designed for the minimum deflection at the extremity, for any appreciable swing, under wind pressure, might deflect the beam of light out of the drivers' line of vision. It is thus usual to make the towers extremely stiff, a point that is often rendered more difficult by restricted space between running roads. Apart from wind pressure, and the dead loads, such structures are liable to be subjected to a type of loading which it is not possible to calculate with any accuracy—namely, the suddenly-applied upward loads on the underside of the landing due to the blast from the locomotive exhausts. Instances have been recorded where these upward loads have been of sufficient magnitude to deflect the beam of colour light signals upward out of the drivers' line of vision. A further point in the design of structures for colour light signals is that they should be capable of easy and rapid handling in erection; usually very little time is available between trains, even on Sundays, when such erection work is usually done.

(*e*) *Relays.*—In modern signalling relays are used for so great a variety of purposes, that the merest outline of their design and uses is all that it is possible to give in the space available. Where D.C. track circuits are used the track relays are generally similar in design to those used for line purposes, signal control and such like, differing only in the voltage on the coils. Performance characteristics for track and line relays are laid down in B.S.I. Specs. 452 and 475 respectively. For indicating the lie of points, and thus affording the necessary 'proving' for inclusion in signal control circuits polarised D.C. relays are used. When no current is applied to the coils this type of relay takes up a central position, while the normal and reverse positions are assumed when the coils are energised according to the direction of flow of the direct current. A further type of D.C. relay, now in increasing use, is the small single-coil instrument roughly of the same proportions as those used in automatic telephone switching. These have been used in a number of relay interlocking and remote control schemes.

In A.C. signalling schemes the vane type of relay is in general use. An aluminium vane takes the place of the conventional armature in a D.C. relay, moving vertically between the pole faces of a horse-shoe shaped core; the form of construction used gives rise to eddy currents in both the pole faces and the vane when alternating current is applied to the coils, and the resultant reaction between the vane and the pole causes the vane to move upwards and rotate the horizontal spindle on which it is mounted. This movement, transmitted through a suitable link mechanism, is utilised for the opening and closing of the relay contacts. For line relays a single core is used, and the entire control current passes through the coils mounted on this core; this type is known as a single element relay. For A.C. track circuits two-element relays are used in which the greater part of the power necessary to produce the required torque on the spindle is provided by a local supply, thus leaving only a very small proportion to be furnished by the current flowing through the track circuit. Efficient performance of the relay is thereby ensured, on the lowest possible track circuit current. As in D.C. installations, three position relays are used also in A.C. territory for point indication, and these are of the two-element type.

The importance of providing for easy maintenance of signalling relays is reflected in the present tendency to use detachable terminal boards. In large interlockings each relay will have a number of wire connections, and should it be necessary to remove a relay for attention, or routine examination, the uncoupling of so many wires takes a considerable time; and after replacement has been done the re-connected wires must be tested out. On the railways of the London Transport Executive it has for some years now been standard practice to use relays with detachable terminal boards. With this arrangement no wiring need be disconnected at all in order to replace a relay; manipulation of simple screw fastenings enables the terminal board to be lifted clear, and the relay mechanism may be removed and another substituted in a very short time.

Remote Control.

The remote control of signalling apparatus offers almost limitless possibilities for the co-ordination of traffic working in a wide area, with, at the same time, a considerable saving in man-power. The operation of points and semaphore signals by high voltage D.C. supplied from hand generators has already been discussed, though this method is limited largely to the working of isolated functions which would otherwise be beyond the range of control of the nearest manual signal box. There are other cases where it may be desirable from a traffic point of view for an entire interlocking to be remotely controlled. For this purpose an interesting system was developed by the London Passenger Transport Board, now London Transport Executive.

A signal box is fully equipped with a power-frame similar to the Board's standard type, complete with all the usual mechanical interlocking between levers, and the electric magnets for track and indication locking. But instead of being operated by a man, the entire machine is remote-controlled. Each of the levers in the locking frame is moved by two small compressed air cylinders, governed by electro-pneumatic valves. The circuits for controlling the operation of these small cylinders have no effect upon the safe working of the signalling as every safeguard is provided by standard apparatus between the remotely-controlled locking frame and the signals and point mechanisms concerned. Thus the remote control circuits can employ telephone type relays wiring and cable, rendering this part of the system relatively cheap. One of the principal advantages claimed for this arrangement is the reduction in cost over that of a single large interlocking frame in an extensive layout, since the small remotely-controlled frames can be situated near to the signals concerned, and the signalling cables kept short. For a fully detailed description of this system see *Proceedings of the Institution of Railway Signal Engineers*, 1942.

On the main line railways, the use of power signalling apparatus, and the means not merely for controlling it but of indicating its performance, has enabled many remote control schemes to be successfully operated. The 14-mile stretch of the London-Edinburgh main line between Northallerton and Darlington has only one intermediate signal box; some of the outlying points and crossover roads are two miles from the controlling signal box, and although many of the signals work automatically the whole line is under the surveillance of one or other of the three signal boxes concerned—Northallerton, Darlington and the intermediate junction at Eryholme. The location of trains, including those on the automatic sections, is indicated on the illuminated cabin diagrams, and in the event of a stop being required at an automatic signal displaying the danger indication the trainmen are required to telephone the signal box ahead for instructions. There is always

the chance that such a stop may not be due to occupancy of the road ahead, but due to some failure of the signalling apparatus. With the traffic position displayed on his illuminated diagram a signalman on being telephoned can judge the reason for the train being stopped and can advise the trainmen accordingly.

There is now theoretically no limit to the distance from which points and signals can be operated; the practicable limit is set by the cost of cable, which beyond a certain point will outweigh the savings effected by remote control. In recent years, however, the entire complexion of the problem has been changed by the development of the coded system, first introduced in the U.S.A. for regulation of traffic working on long stretches of single line. It is now widely known as Centralised Traffic Control, or C.T.C., and although there has so far been only one example in this country, the possibilities are great, and it is already being extensively installed on the New Zealand Government Railways. Instead of direct wire transmission from the control machine, or power-frame, to each individual signal, or pair of points, the entire control is carried out on two line wires. Not only one station, but a complete section, or subdivision of the line may be controlled from a central office. The limitation with this system lies in the use of only two line wires. Only one code can be transmitted at a time; for each code sent out from the control machine there is a corresponding return code, proving that the particular signal or points have responded. The time taken for the complete transmission of a code, outward or return, is approximately 3½ seconds. To the operational and proving codes, must be added the numerous incoming codes showing the changing occupancy of the track circuits. An analysis of traffic movements, together with the number of signal and point operations, will therefore show the extent of the area which can be brought under the control of one machine, as represented by the capacity of the two line wires to handle all the codes likely to be transmitted. The electrical principles on which C.T.C. is based were described in *The Engineer* for September 10, 1943. By utilising the principles of carrier telephony it is now possible to superimpose other codes on the two lines of a C.T.C. system. If the line extending through a controlled area be divided into two sections, the section nearest the control machine will be operated by ordinary C.T.C. codes; on this same section of the two line wires are superimposed the higher frequency carrier codes for the more distant section. At the division point the carrier codes are changed into ordinary D.C. codes for the control of points and signals on the section furthest from the control machine.

Automatic Train Control.

The numerous safeguards so far mentioned, lever interlocking, 'lock and block,' track circuiting and so on, are all designed to safeguard against errors in working on the part of signalmen; it is to provide against possible misinterpretation of the signals, misjudgment of distance, and such like, on the part of locomotive drivers that a series of safety features, which may be grouped together under the general heading of Automatic Train Control, have been introduced. In studying the subject, the speed and brake power of the trains must be considered no less carefully than the wide variation of traffic conditions existing in different parts of the country. On the intensely used railways of the London Transport Executive, and certain other electrified suburban lines, the speeds are relatively so high and the interval between successive trains so short, that the chance of an overrun at a signal displaying the danger indication is unthinkable. There must be no question of 'warning' a motorman in such circumstances; control must be taken completely and instantly out of his hands, and the train stopped in the shortest possible distance. From the practical point of view the problem is made simpler by the maximum speed and brake power of the trains being alike. A train stop mechanism is installed, co-acting automatically with each stop signal, in which a stop arm is raised when the signal is at danger; in the case of an overrun this stop arm is struck by the hanging lever of a trip cock on the train. When a train is thus 'tripped' an emergency application of the brakes is made, which the motorman is powerless either to cancel or forestall. On the railways of the L.T.E. the train stops are electro-pneumatic, while on the lines of the former L.M.S.R. in the London suburban area they are all-electric.

This necessarily drastic system would not be practicable in main line railway working. In the maintenance of high speed express passenger services the 'distant' is the important signal. Unless a driver has an adequate sight of this warning signal he cannot be expected to run with confidence, and the effects on punctuality caused by foggy weather are only too familiar. The G.W.R. system of automatic train control was originally designed to lessen the difficulties occasioned by fog, but it has proved a most valuable aid to efficient train operation in any circumstances, and is now installed over the Western Region main line network. It provides an audible indication, in the locomotive cab, of the aspect displayed by each distant signal along the line. If the wayside signal is clear, a short loud ring is given on a bell in the cab, and no action is required on the driver's part. If the distant signal is displaying the 'caution' indication, the cab signalling apparatus causes a siren to start up, the blast of which continues until the driver operates an acknowledging switch. A system of automatic train control, giving similar indications, was being developed on the L.M.S.R. and L.N.E.R., though the details of operation, as will be shown later, are quite different.

In these British systems of A.T.C. the operating of the acknowledging device forestalls the automatic application of the brake, and enables the driver to retain control of the train. While it might at first be thought that the provision of such a device goes some way towards nullifying the whole advantage of the automatic control, it must be recalled that the number of accidents in Great Britain due to drivers overrunning or misreading signals are astonishingly few. The purpose of A.T.C. in main line working is not to take control out of the driver's hands, but primarily

to provide him with confirmation of the aspect displayed by the distant signal, in the form of an audible indication in the cab. To introduce a purely automatic stop, on the same principle as that used on the L.T.E., would be to cause widespread delay, particularly in the operation of freight trains. Drivers of heavy trains on sighting a distant signal in the 'caution' position almost invariably take steps to reduce speed to dead slow, and then they draw slowly forward, by which time the section ahead has probably cleared. By this practice a very large number of dead stops are avoided.

In the Great Western system, connection between the apparatus on the locomotive and the wayside signals is made through a contact ramp fixed in the 'four-foot' way. The act of passing over a ramp raises a contact shoe on the locomotive, and this mechanical action takes place irrespective of the indication displayed by the distant signal. The shoe is fitted with circuit breaker contacts, which are normally made; the raising of the shoe when passing over a ramp opens the circuit to a valve magnet. When the distant signal is at 'caution' no current is picked up when the locomotive passes over the ramp, and the de-energising of the valve magnet (due to the opening of the contact shoe circuit breaker) opens the valve and causes the siren to sound. The driver can restore the circuit, and thus silence the siren, by operating his acknowledging lever. If he did not acknowledge promptly, not only would the siren continue to sound, but the reduction in vacuum, due to air entering the train pipe through the valve, would apply the brakes. If the distant signal is in the clear position current is applied to the ramp, through a signal arm circuit breaker, and this current is picked up by the locomotive contact shoe when passing over; it provides an alternative feed to the valve magnet to prevent the sounding of the siren, and supplies also a feed to ring the bell. In a test stop, when a locomotive and train together weighing 430½ tons passed a distant signal at danger, the siren was not acknowledged and steam was kept full on. This test took place on a length of straight and level track, and from an initial speed of 69 m.p.h. the train was brought to rest in 900 yds. from the ramp.

This G.W.R. system was designed to operate in semaphore signalled territory. It can, of course, be applied to colour light signalling, though on stretches where four indication signals were contemplated the necessity arose for some distinction to be made between the preliminary warning, as provided by the 'double yellow' indication of the wayside signals, and the second warning given by the single yellow. With some ingenuity the standard system was very simply adapted to give a double blast on the audible cab signal to correspond with the 'double yellow,' leaving the sounding of the siren to accompany the approach to a single yellow, as at present when approaching a semaphore distant signal in the caution position. The result was obtained by polarising the current applied to the ramp, one polarity being used in conjunction with the 'green' indication, and opposite polarity for the double yellow, in which latter case an electro-pneumatically controlled horn was sounded immediately prior to the sounding of the ordinary siren. A full description of this arrangement, together with particulars of a test run at 97 m.p.h. was published in *The Engineer* for October 17, 1947.

In the system installed on the former L.M.S.R. Southend line and at one time projected on the Edinburgh and Glasgow section of the L.N.E.R., the contact ramp is replaced by an inductive pick-up. On the lines so far equipped the audible signals received in the cab are a short blast of the siren for 'clear,' and a continuous blast for 'caution'; this latter is silenced by the driver operating the acknowledging plunger. Developments are in progress, however, with this inductive system which in later installations will bring the audible cab signals almost exactly into line with those provided by the G.W.R. The subject of the automatic train control was discussed at length in *The Engineer* (July 12, 19, 26, August 2 and 9, 1940), when existing British systems were compared with those in use overseas.

An experimental installation of continuously-controlled cab signals has been made between New Barnet and Potters Bar on the main line from King's Cross to the North. In many ways this installation follows the practice standardised on the Pennsylvania Railroad, but certain adaptations have been made to suit the arrangement to British practice (see *The Engineer*, November 14, 1947). The section of line is track circuited throughout, with track circuits of the coded type, and the coded currents flowing in the running rails are conveyed to the receiving apparatus on the locomotive through an inductor mounted on the policeman irons. The principle of coded track circuits was discussed at some length in *The Engineer* for September 24, 1943, but a brief summary follows for ready reference.

Whereas ordinary track circuits of the non-coded type depend for their operation on a supply of power being applied continuously at one end of a section of insulated rail, in the coded track circuit the supply of power to the rails is interrupted by code transmitters located at the feed end of each track circuit. At the other end of the track circuit the normal type of relay is replaced by what is termed a code following relay, which is alternatively energised and de-energised in synchronisation with the current impulses applied at the feed end of the track circuit. With this arrangement the current feed with the running rails is interrupted at different periods according to the occupancy, or otherwise of the road ahead, and codes are picked up by the inductor apparatus and used for the control of the cab signal.

Control of Marshalling Yards.

Marshalling yard operation has been studied extensively in recent years, with a three-fold view: to increase safety of working, both from the viewpoint of damage to merchandise and injury to personnel; the more efficient use of motive power; the quicker clearance of traffic. The actual layout of marshalling yards is not primarily a signalling matter, though in such yards the work of

the signal engineer is more intimately bound up with the actual running of vehicles than perhaps anywhere else, save on such intensely used lines as the London tube railways. The use of the hump type of yard introduces wagon speeds up to about 20 m.p.h., as they approach the classification roads; but it is not so much the speed as the short distance between successive wagons, or groups of wagons coupled together, that makes necessary the quickest possible operation of points. The interval is so short that it would be unreasonable to expect a man watching from the control tower always to judge the correct moment for moving the points, and the control is therefore arranged automatically, by the passage of the wagons themselves over critically-sited track circuits.

When a train enters the reception sidings of a marshalling yard, prior to classification, a list is compiled of the sidings into which each wagon must be switched after shunting over the hump. Where two or more successive wagons are routed to the same siding they are coupled together, and known generally as a 'cut.' The list of 'cuts' is sent to the man in charge of point operation, and he must arrange for the route to be set for each. By utilising the principles of relay interlocking the routes can be pre-set, or 'stored' as it is termed, so that as soon as one wagon, or cut, has passed over the first pair of diverging points route-setting for the next is begun immediately; the 'storage' circuits are so arranged that the various stages in the setting up of a route take place only after the preceding 'cut' has cleared the fouling points. As yet no standard practice for route setting has been evolved. As with most new developments the earliest installations embody successive new ideas. A description of British systems of working mechanised yards, and the part these played in the handling of wartime traffic appeared in *The Engineer* for July 10, 1942.

In adapting power signalling apparatus for automatic point control in marshalling yards certain fundamental changes were made in design in order to secure the fastest possible operation. In so doing, two safety features which it is imperative to include on passenger lines were dispensed with. The first of these concerns the track circuits. In ordinary signalling the track circuits are normally energised, and the relay is shunted by the entry of a train, or vehicle on to the particular section; in hump yards the track circuits are normally de-energised. D.C. relays pick up more quickly than they release, and by using this characteristic the point control circuits depending upon the occupation of a particular track circuit can be initiated more rapidly. The saving does not amount to more than a split-second, but in the limited time available for throwing the points any saving is worth having. The second important change is that no facing point locks are used; in cases of electro-pneumatic working the cylinders are coupled direct to the first stretcher bar, and the control valves are diaphragm operated, again to giveth utmost speed in working.

B.S.I. Specifications relating to Railway Signalling.

For ready reference a list is given below of the British Standards Institution specifications at present issued relating to mechanical and power signalling apparatus and wiring diagrams.

Apparatus.	*B.S.I. Spec. No.*
Cartridge fuses	714
Colours for signal glasses	623
Electric lamps	469
Lenses (plano-convex)	624
Mechanical signalling apparatus	689
Electric point operating machines	581
Line Relays	
A.C. Single element (2 position)	557
A.C. 2-element, 3 position	561
D.C. tractive armature neutral	475
D.C. tractive armature neutral-polarised	519
Relays, thermal type, time-element	635
Track Relays	
A.C. 2-element 2-position	520
D.C. tractive armature	452
Terminals for power signalling	442
General.	
Glossary of terms	719
Symbols used in plans, etc.	
Schematic	376, Part 1
Wiring and written circuits	376, Part 2

SECTION XXXIII

GAS AND GASWORKS

(pp. 615–656.)

(Contributed by Harold Moys, B.Sc. (Eng.), A.M.I.C.E.,
M.Inst.Gas E.)

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Parliamentary.

The public supply of gas in this country is governed by Acts for particular undertakings and by Acts applying to the whole industry. The latter class include the Gasworks Clauses Acts of 1847 and 1871, the Sale of Gas Act of 1859, the Gas and Water Works Facilities Acts of 1870 and 1873, the Gas Regulation Act of 1920, the Gas Undertakings Acts of 1929, 1932 and 1934. London companies are, in addition, under the regulations of the Metropolitan Gas Act (1860). Under the Public Health Act of 1875 the local authority may become its own gas supplier or may purchase an existing undertaking by agreement. In addition to the Acts named above the supply of gas in Scotland is governed by the Gas Acts of 1864 and the Burghs Gas Supply Acts of 1876 and 1893. Thus gas works are controlled by—

(*a*) Statutory companies.
(*b*) Non-statutory companies.
(*c*) Local authorities.

The supply of gas is, for the most part, in the hands of the local authorities in Scotland and the northern counties of England, particularly Yorkshire and Lancashire, but in the Home and Southern counties private ownership is the rule.

The Gas Regulation Act of 1920 is divided into four sections:—

(*a*) Price and quality of gas.
(*b*) Testing of gas.
(*c*) Forfeiture and Penalties.
(*d*) Power to make Special Orders.

Section (*a*).—The Board of Trade is authorised to make orders allowing undertakings applying to it to charge for gas on the thermal basis, to adjust the standard or maximum price (on the heat unit basis) to meet the additional expenses of manufacture and distribution arising from post-war conditions and to revise these charges should it be shown by the local authority or the undertaking that the profits are excessive or unjustly small. The undertaking is given freedom to choose the calorific value of the gas it will distribute, provided that it advertises the value and adjusts at its own cost all burners and appliances to use the gas with safety and efficiency. The gas must be free from sulphuretted hydrogen and shall be supplied at not less than the permissible pressure, *i.e.* 2 ins. water pressure in mains or services of 2 ins. diameter or upwards.

The Board of Trade were authorised to prescribe the limitations of inert constituents and carbon monoxide content but a Departmental committee recommended no such restrictions (1922).

Section (*b*).—This section deals with the Testing of Gas and the appointment of the Gas Referees, Chief Gas Examiner and the local gas examiners; the duties of the Referees, the appeals to the Chief Gas Examiner and the creation of the Gas Fund.

Section (*c*).—The penalties and forfeitures for non-compliance with any lawful prescriptions of the Gas Referees and failure to maintain the gas within the normal range of calorific power, purity and pressure.

Section (*d*).—Under this section the Board of Trade is enabled to do by Special Order practically everything which had previously been done by Act of Parliament or provisional order made under the Gas and Water Works Facilities Act, 1870.

The Gas Undertakings Act of 1929.

This Act was the result of a compromise between the parliamentary authorities and the gas industry. The latter had complained that it was working under obsolete legislation which tended to cripple its freedom of action and that the Government by creating the Electricity Commissioners and guaranteeing interest on a large amount of capital and by generally encouraging electrical development was supporting a newer industry to the detriment of the older one. The Gas Undertaking Act of 1929 was the outcome, and although satisfactory so far as it went, it was only regarded as a first step towards greater freedom. The principal powers of this Act are as follows:—

1. Without any further authorisation a company can raise by way of a loan an amount up to 50 per cent. of the ordinary and preference capital including premiums.

2. Subject to the consent of the local authority the Board of Trade can authorise a company to raise by way of loans a sum equal to 100 per cent. of the ordinary and preference capital. Under certain conditions the Board of Trade may grant this power even if the local authority objects.

3. In the case of the old reserve fund or the more modern special purposes fund the permissible total can be increased to include the prescribed proportion of loan capital.

4. Restrictions on the manufacture of residuals were removed.

5. The Board of Trade can authorise the supply of gas to premises outside the limits of supply of any specific undertaking.

The Gas Undertakings Act, 1932.

This Act permits any undertaking which has supplied more than 250 million cub. ft. of gas in a year to invest (*a*) if a local authority up to one-tenth of the capital expended in certain defined companies or local authorites, (*b*) if a company, up to one-twentieth of its authorised share and loan capital or with the approval of the Board of Trade up to one-tenth.

The Gas Undertakings Act, 1934.

This, the most recent general legislation for gas undertakings, is extremely important as it alters the text of the main Act and introduces several new principles into gas law. It is subdivided under five headings:—

(*a*) Capital and Renewal Funds.
(*b*) Charges for Gas.
(*c*) Non-Statutory Undertakers and Undertakers other than Thermal Unit Undertakers.
(*d*) Amendments of Principal Act.
(*e*) Miscellaneous and General.

(*a*) *Capital and Renewal Funds.*—The Board of Trade is given power to authorise the issue of share capital and debentures by public subscriptions provided it is satisfied that the terms are the best obtainable. The payment of underwriting commission is sanctioned. Permission is given to create redeemable preference stock and debentures within the authorised capital. The contribution to the Renewal Fund shall now be limited to ½ per cent. of the capital of th : undertaking and the amount shall not exceed 5 per cent.

(*b*) *Charges for Gas.*—Any thermal unit undertaking can give notice in writing to the local authority of the price or prices, if differential prices for different areas have been previously enacted, at which they are prepared to supply gas to persons who do not enter into a special contract. A copy of this notice must be displayed in all showrooms and offices. This price is the published price and is the price to be used in the calculation of dividend which already appertains to the undertaking. The undertakers can then enter into Special Contracts with consumers upon any agreed terms ' provided that the terms of every contract under this section in the like circumstances and for the same purposes shall be alike.'

(*c*) *Non-Statutory Undertakers and Undertakers other than Thermal Unit Undertakers.*—Any non-statutory undertaking supplying more than 30 millions cub. ft. of gas in a year must make application to the Board of Trade who can make an order conferring upon the undertaking the power and restrictions applying to the statutory undertaking. Non-statutory undertakings supplying between 20–30 millions cub. ft. of gas per annum who fail to maintain a satisfactory supply of gas shall be served with a notice by the Board of Trade requiring them to seek statutory powers. The gas supplied by non-statutory or non-thermal unit undertakers must be free from sulphuretted hydrogen and be supplied at adequate pressure (2 ins. W.G.). This provision does not apply to undertakers supplying less than 20 million cub. ft. per annum.

(*d*) *Amendments of Principal Act.*—The office of chief gas examiner was abolished on January 1, 1935, and appeal is now to be made to the Board of Trade. The office of Gas Referees was abolished on January 1, 1939, and their function is performed by the Board of Trade

who have power to appoint up to three persons to advise them. The gas examiners will be appointed by the local authorities or quarter sessions upon the application of five or more consumers. When the average calorific value of the gas supplied during one quarter is less than the declared value the excess revenue if less than one-fifth of a penny per therm, shall be paid into the Gas Fund (in addition to the usual contribution) or if more than one-fifth of a penny per therm shall be allowed to the consumers as a credit at the end of the next quarter.

A minimum of six tests is specified for the calculation of the average calorific value.

(e) *Miscellaneous and General.*—Only a few of the more important revisions made under this section are given below.

Closer co-operation possible between undertakers with regard to bulk supplies and joint use of research establishments.

All consumers must give 24 hours' notice before quitting premises or accept responsibility for the gas consumed up to the next ordinary meter reading.

The undertaking can refuse a supply of gas to a consumer who has previously quitted premises without payment until such debts are discharged.

Additional powers of entry for the inspection of factories.

A consumer who is connected up to the mains merely as a standby is not entitled to receive a supply unless he agrees to make certain minimum payments.

Consumers must fit and maintain back pressure valves and antifluctuators, if deemed necessary.

The Kettering Clause is inserted in the general Act 'It is not lawful for any local authority to insert in any instrument for the sale or letting of premises a provision restricting the right of any occupier to take a supply of gas.'

A non-statutory company operates as a result of established practice or under the goodwill of the local authority. Its position is in many respects similar to that of a statutory company, but before opening roads for the laying of mains permission must be obtained from the district or county authority.

Emergency War Regulations affecting Gas Undertakings.

The laws affecting gas undertakings during the War Emergency were many.

Orders made specifically for the Gas Industry, the Gas Supply (War Damage) Order, 1940, the Gas Supply War Damage (No. 2) Order, 1940, protected the undertaking from the usual penalties for the failure of normal supply or maintenance of declared calorific value. Additional Orders under the Defence (General) Regulations, 1939, have helped the undertaking to alter its declared calorific value in a shorter period than hitherto and have given them additional powers to meet the present conditions.

The industry is affected by orders dealing with the control of timber, steel, non-ferrous metals, fertilizers, and war risks insurance, and the scheduling of protected industries.

Since 1942 the powers formerly vested in the Board of Trade have been transferred to the Minister of Fuel and Power.

Statistics of Gas Undertakings in Great Britain.

The following figures are taken from White Papers issued by the Ministry of Fuel and Power.

Number of Undertakings, Materials Used, Gas Made and Total Gas Available.

	1938.	1940.	1942.	1944.	1946.	1947.
Number of undertakings:						
Statutory: Local Authorities	298	284	275	274	274	275
Companies .	405	409	406	406	406	402
Other Companies . .	402	380	371	367	366	361
Total . . .	1,105	1,073	1,052	1,047	1,046	1,038
Materials used for making gas:						
Coal (thousand tons) .	19,128	17,983	20,634	20,777	22,646	22,716
Gas oil (thousand gallons) .	32,707	40,180	52,858	96,116	132,220	172,958
Total gas made (million cu. ft.)	319,549	306,557	343,063	363,889	418,855	432,413
Gas bought from coke ovens (million cu. ft.) . .	29,621	36,765	46,880	49,081	53,088	53,984
Total gas available:						
Million cu. ft. . . .	349,170	343,322	389,943	412,970	471,943	486,397
Million therms . .	1,659	1,579	1,813	1,924	2,240	2,310

Gas Consumers—Consumption per Consumer.

The following statistics are taken from the Ministry of Fuel and Power Statistical Digest, 1946 and 1947.

STATISTICS FOR YEAR, 1947.

	Number of Consumers.	Million Therms Sold.	Therms per Consumer.
Mainly domestic: Prepayment	7,634,980	918·9	120·3
Credit	3,287,171	472·2	143·7
Total	10,922,151	1,391·1	127·4
Industrial	105,094	416·2	3,960·4
Commercial (shops, offices, etc.)	511,041	202·4	396·1
Local Authority establishments	54,293	43·9	807·6
Government Departments	14,389	9·4	656·2
Total	579,723	255·7	441·1
Service Departments	5,690	6·3	1,111·9
Public Lighting	—	25·8	—
Total	11,612,658	2,095·1	180·4

The fuel used for gas production in the Gas Industry during 1947 was:—

Coal Carbonisation.

	Tons.
In horizontal retorts	9,381,508
„ continuous vertical retorts	10,892,957
„ intermittent vertical chambers	1,547,623
„ inclined retorts, static vertical retorts and coke ovens	589,634
	22,411,722

It will be seen that the continuous vertical retort is most widely used and is gradually replacing inclined and horizontal retorts.

Carbonising Plant.

There are four main types of carbonising plant in general use in this country, namely:

Horizontal retorts;
Continuous vertical retorts;
Intermittent chamber ovens; and
Coke ovens.

In all settings except the continuous vertical retort, the charging is intermittent, the coal being carbonised for a fixed time.

The following factors influence the type of plant selected:

Type and grade of coal to be used.
Calorific value of gas to be made.
Type of coke required.
The market for tar.
Size of the installation.
Ground space available.
Local amenities.
Degree of flexibility in operation required.

With horizontal retorts, intermittent chambers and coke ovens the widest range of coals is available for carbonisation, but the choice with continuous settings is more limited due to the difficulty in dealing with strongly swelling coals.

It is generally found difficult to produce a high grade gas, *i.e.* above 520 B.Th.U's. per cub. ft., in continuous vertical retorts and to produce an economic low grade gas in horizontal retorts or coke ovens. Intermittent vertical chambers are the most flexible in this respect.

The coke produced from continuous vertical plants is very free burning but somewhat friable, whilst that from intermittent vertical chambers and coke ovens is dense and somewhat difficult to ignite; this latter is excellent for steam raising purposes.

The tar produced from coke ovens and horizontal retorts is superior to that from continuous vertical retorts and can easily be treated to comply with the B.S.S. for road tar.

Coke ovens are only employed in large installations. With the other systems the charging and discharging machinery necessary with horizontal retort and intermittent vertical chamber installations renders their capital cost high in small units. In such cases the continuous vertical retort would prove attractive.

The table below, taken from F. B. Richards' paper, shows the output in therms per sq. ft. for plants actually installed.

Type of Installation.	Total output Therms/day.	Total ground space occupied Sq. ft.	Therms/day per sq. ft. of ground space.
Horizontal retorts	34,560	23,400	1·48
Continuous vertical retorts	50,388	14,212	3·55
	42,028	13,230	3·18
	19,890	7,326	2·72
	1,210	950	1·27
Intermittent vertical chambers	45,060	18,914	2·37
	20,780	9,388	2·21
	1,920	1,830	1·05
Coke ovens	84,500	62,200	1·35

On the grounds of local amenities the claims of the continuous vertical retort are greater than those of the other types although the intermittent vertical chambers can be improved by the installation of dry cooling plants for coke quenching.

Flexibility of output can be obtained with all plants, but widely ranging calorific values are most easily obtained with intermittent vertical chambers.

Proportions of Horizontal Retort Benches.

The following normal dimensions are given as exemplifying the usual proportions of various types of retort houses and benches:—

(*a*) Single retort bench, 10-ft. retorts:—

	ft.	ins.
Width of bench	12	6
At back of bench	1	6
In front of bench	18	0
Total width inside houses	32	0

The height of the retort house to the eaves would be 20 ft., and the height of the retort bench from ground level 11 ft.

(*b*) Through retort bench, hand charging and discharging:—

	ft.	ins.
Width of bench	20	0
Each side of bench, 20 ft.	40	0
Total width of house	60	0

The height of the retort house to eaves, if house of the stage type, would be about 35 ft.; if of the subway type 8 to 10 ft. less. The height of the retort bench from base of producers to top of brickwork about 20 ft., if retorts set in three tiers.

(*c*) Through bench, mechanical charging and discharging:—

	ft.	ins.
Width of bench	20	0
Charging side of bench	23	0
Discharging side	17	0
Total width of house	60	0

The height of the bench from base of producers to top of brickwork would be about 25 ft. if retorts are set in five tiers. The height of the house (stage type) to eaves would be about 40 ft. if continuous coal hoppers overhead were included. As an approximate rule for ascertaining the height of the bench from charging floor to top of brickwork, allow 29 ins. for each tier of retorts. The width of the main setting arch will be, for two rows of retorts, from 7 ft. 9 ins. to 8 ft. 9 ins.; for three rows of retorts, 8 ft. 6 ins. to 9 ft.

Vertical Retorts.

Modern vertical retorts are of three types. (*a*) Continuous; (*b*) Intermittent; (*c*) Static.

Continuous Vertical Retorts.

Woodall-Duckham System.—The retorts are now built in five standard sizes which vary only in the dimensions of the major axis, the minor axis being the same in each case. The coal carbonised and the average gas output per day of the different sizes of retorts are as follows:—

Mean Major Axis.	Coal Carbonised.	Gas Output.
	Tons.	Cu. ft. per day.
44 inches . . .	3 to 5½	51,000 to 59,000
53 " . . .	3½ " 6½	59,000 " 94,000
62 " . . .	4½ " 7½	72,000 " 109,000
82 " . . .	5½ " 9½	90,000 " 142,000
103 " . . .	6½ " 12	115,000 " 180,000

Glover-West System.—In this system the retorts are elliptical in shape, having a taper towards the bottom. At the base of the retort is fitted a large steaming chamber and this has allowed the amount of steam admitted to the retort to be increased with the consequent increased yield of gas per ton of coal carbonised. The retorts are set in units of 2, 4 or 8 and they are made in three sizes, the particulars in connection with which are given in the following table:—

	Type 1.	Type 2.	Type 3.
Major and minor axis of ellipse at top	33 ins. × 10 ins.	40 ins. × 10 ins.	50 ins. × 10 ins.
Major and minor axis of ellipse 21 ft. from top	39 ins. × 18½ ins.	46 ins. × 18½ ins.	56 ins. × 18½ ins.
Total length of retort . . .	25 ft.	25 ft.	25 ft.
Capacity per day in cubic feet of gas	60,000	75,000	90,000

Drake's System.—The 'Drakes' Vertical Retort is made in varying sizes to cope with individual requirements, the minimum size being a nominal 3-ton retort, having a major axis of 36 ins. and a minor axis of 18 ins.

All the retorts are rectangular in shape and have a uniform taper from top to bottom on both major and minor axes.

The retorts can be grouped in single or double units or in units of four, each separate unit having individual producer gas and air supplies and waste gas outlets.

Intermittent Vertical Chambers.

Continuous vertical retorts are generally considered unsuitable for the carbonisation of highly expanding coals. Where such coals have to be used intermittent vertical chambers would be considered. Compared with horizontal retorts they have these advantages. (1) No costly machinery for retort charging and discharging. (2) Bulk carbonisation. (3) Increased flexibility due to the suitability of the charge for steaming.

The capacity of the chambers varies, but a normal figure is 3½ tons, the chamber having a minor axis of 8 ins. at top and 12 ins. at bottom, major axis of 9 ft. 8 ins. at top, 10 ft. 0 in. at bottom and a height of 21 ft.

Settings have been designed with built-in producers and recuperators or with outside producers.

It is usual to provide gas off-takes at the top and bottom of the chamber, both off-takes being closed during charging and discharging. The top off-take is fitted with a saucer valve so that, when open, the pull of the collecting main is transmitted to the retort. The bottom off-take is sealed to a depth of about 2 ins. so that it is used only to reduce the pressure at the bottom of the chamber during peak pressure conditions. Due to the high temperature of the whole of the charge, steaming during the last two hours is extremely efficient. The tests on the intermittent vertical chambers at Croydon carried out by the Joint Research Committee of the Institution of Gas Engineers and Leeds University, 1931–32, show that the thermal output was 78·6 therms per ton and a dry coke yield of 11·3 cwts. The calorific value of the gas was 549 B.Th.U./cub. ft.

Static Vertical Retorts.

The best-known plant of this type is the 'Westvertical.' In this retort the charge remains static during the carbonising period. The system is, however, completely enclosed and steaming is continuous. The carbonising chambers so far constructed have been 8 ft. 4 ins. wide on the major axis and about 20 ft. deep in the carbonising zone The chamber is extended downwards through the intermediate zone to the cooling zone, the charge being dropped after carbonisation

into the zones for steaming and cooling. The residual heat of the coke is restored to the system, the coke is discharged cool and dry and the intermediate zone is sufficiently hot for the producton of water gas. It is suggested that hydrogenation of the tar vapours is effected and that the overall thermal efficiency is improved.

The static vertical retort has also been developed by the Woodall-Duckham companies and a full scale model plant was installed at the Croydon Gas Works. Like the 'Westvertical' plant the static vertical retort consists of an intermittently operated two-stage vertical carbonising retort divided into two superposed portions, the upper one heated by external flues forms the carbonising zone, and the lower forms a zone in which the coke to a certain extent is cooled. From the published figures available for this test plant it would appear that the thermal yield is considerably improved over those for both intermittent vertical chambers and continuous vertical retorts.

Whilst the number of commercial installations of the 'Westvertical' and Woodhall-Duckham static vertical retort is not large it would seem that this method of carbonisation has many attractive features.

Admission of Steam.

The process of admitting steam to the base of the retorts on continuous and intermittent vertical retort installations has now become standard practice. This process has also been applied with varying success to horizontal retorts. Blue water gas is generated *in situ* and besides increasing the yield of gas and the therms per ton of coal carbonised, some noticeable augmentation of the liquor and tar yields has resulted. The effect of steaming is well illustrated by the following results obtained at the Fuel Research Station at East Greenwich with Mitchell Main gas coal:—

	Cu. ft. of Gas per Ton of Coal.	Calorific Value B.Th.U. (gross).	Therms per Ton of Coal.
Without steaming	13,100	544	71·26
Steaming 20·53 per cent.	22,580	460	103·86
" 30·03 "	35,610	432	110·63

Dilution of Coal Gas with Producer Gas.

Coal gas produced in continuous vertical retorts and intermittent vertical chambers can be diluted to about 400 B.Th.U. per cub. ft. by water gas produced *in situ*. For small works without water gas plants dilution of the high grade gas from horizontal retorts is best effected by the admixture of producer gas. This can be obtained from the ordinary built-in producers or from an independent producer. Various devices have been designed for controlling automatically the rate of admission of the diluent so as to give a steady calorific value. The advantages derived from the use of producer gas dilution are:—

(1) Improved working in retort house due to the absence of overpulling.
(2) Decreased wear and tear in retort houses.
(3) Increased thermal yield.
(4) Decreased cost per therm of mixed gas.
(5) More stable gas for use on the district.

Owing to the acute shortage of plant resulting from the war period, many experiments have been tried to increase the capacity of existing plants. As the result of one such experiment it has been found that running continuous vertical retorts on high coal throughputs with reduced steaming and diluting the high grade coal gas by producer gas gives a greater output than when producing the correct type of gas by steaming in continuous vertical retorts.

Heating of Retorts.

All modern settings on gasworks are heated by producer gas made either in a small built-in producer, a battery of step grate or similar hand-clinkered producers, or in mechanically operated producers.

At coke oven plants not situated at gasworks the rich coal gas or blast furnace gases are used for heating purposes.

The fuel consumptions that may be expected with various types of setting are:—

	Lbs. of coke per 100 lbs. of coal carbonised.
Horizontal retort setting with recuperators	14–17
Continuous vertical retort without recuperators	13–15
Intermittent " " " "	16
" " " with "	14
Coke ovens with regenerators	12

The provision of recuperators or regenerators depends very much on local conditions. If they are fitted, the interest on the extra capital cost involved in recuperators and additional steam raising plant has to be offset against the saving in fuel, less the cost of raising the steam. When fuel costs were low it was generally economic not to fit recuperators to continuous and intermittent vertical installations. During, and since, the war conditions have changed considerably, and it is now unusual not to fit recuperators on intermittent vertical retort installations. It is still usual not to adopt recuperators on continuous vertical retorts, as already much preheating of the secondary air is done by circulation around bottom castings and main seating joists.

In general, horizontal retorts are heated by built-in producers fitted with step grates, one producer serving one setting. On large installations of continuous and intermittent vertical retorts it is the practice to construct outside producers in batteries. The variations in quality and quantity of producer gas are thereby minimised.

Until recently mechanical producers had not been extensively used on gasworks for supplying producer gas to settings. The horizontal retort house at the Partington Works of the Manchester Corporation has used cold producer gas for many years. Some difficulties with gum and tar deposits do occur with cold gas, and recent experiments with hot gas (400° C.) have been made on continuous vertical retort installations. The advantages claimed for the adaption of mechanical producers to gasworks' use are :—

(1) Provision of ample supplies of producer gas under all conditions and at exact pressure required.

(2) Steady quality and heating valve of gas as compared with the fluctuations on small built-in producers frequently inadequate in depth and area.

(3) Provision of dust-free gas leading to long life of refractories and less leakage.

(4) Elimination of poor working conditions on feeding and clinkering producers of horizontal installations.

(5) Ability to produce ample supplies of gas from coke of varying qualities.

(6) Ability to use the smaller grade of coke at present difficult to market.

Producer Grate and Chimney Areas.

Step grate producers are now most commonly employed for heating retorts. The rate of gasification with a modern design varies between 10–20 lbs. of coke per sq. ft. per hour. The higher figure is obtained under forced draught conditions with steam admission.

With mechanical producers high rate of gasification can be obtained. Using graded cokes the following rates were obtained during a recent test :—

Using breeze containing 30 per cent. below $\frac{1}{8}$ in., 40 per cent. $\frac{1}{8}$ in. to $\frac{1}{4}$ in., 30 per cent. over $\frac{1}{4}$ in., the average rate of gasification was 15 lbs. of breeze per sq. ft. per hour.

Using coke nuts containing 10 per cent. below $\frac{1}{2}$ in., 60 per cent. $\frac{1}{2}$ in. to 1 in. and 30 per cent. over 1 in., the average rate of gasification was 23 lbs. of coke per sq. ft. per hour.

These figures are capable of considerable improvement where the producers employ mechanical means to ensure that the top of the bed is level and that rat-holing is minimised. This last trouble is most acute where the material consists of a wide mixture of grades.

For horizontal retort installations brick-built chimneys are most common. The height varies between 50 to 70 ft. The chief considerations to bear in mind are heights of adjoining buildings, avoidance of down draught and shifting winds. The rectangular chimney with an external batter is commonly employed. The batter varies between the limits 1 in 36 to 1 in 40. Retort house chimneys are usually braced for their entire height ; the general method being to run angle irons up each corner, the angles being held in position by tie bands at intervals of 6 ft. An inspection door should be left at the base for cleaning purposes. As regards materials of construction, stock bricks may be employed for external courses, but a lining of fire bricks in the interior is essential. Very frequently the whole chimney is constructed in fire brick, in which case no separate lining is necessary. The height of the chimney is calculated from the formula :—

$$D = H\left(\frac{7 \cdot 6}{t_1 + 460} - \frac{7 \cdot 9}{t_2 + 460}\right).$$

D = Draught in inches water gauge.
H = Height of chimney in feet.
t_1 = Air temperature °F.
t_2 = Mean gas temperature °F.
7·6 = Weight of 100 cu. ft. of air at 60° F. in lbs.
7·9 = „ „ „ „ „ waste gas at 60° F. in lbs.

Mean gas temperature = temperature of the gases entering the chimney less 2° F. for each 3 ft. of chimney.

The draught required by most horizontal settings is about 0·8 ins. water gauge.

The velocity of gases up the chimney should not exceed 15 ft. per second at working temperature, and the area of the chimney should be calculated using this factor.

The gas velocity that can be set up by a chimney varies as the square root of the height.

Composition of Producer Gas.

The following is the average composition under good conditions :—

	Per cent. by volume.
Carbon monoxide	25
Carbon dioxide	5
Hydrogen	6
Methane	1
Nitrogen	63

The analysis of the waste gases emitted from the retort setting should approximate to the following :—

	Per cent. by volume.
Carbon dioxide	19
Oxygen	0·4
Nitrogen	80·6

(The presence of carbon monoxide in the waste gases is indicative of a deficiency of secondary air, and is wasteful in firing. The oxygen, if above 0·5 per cent., denotes an excess of secondary air.)

Gasworks Temperatures.

	°F.
Retorts	1,800–2,100
Ascension pipes	200–300
Foul main (inlet)	140
Condenser inlet)	125
,, (outlet)	78–85
Washers and Scrubbers (outlet)	60
Purifiers (inlet)	60
,, (outlet)	65
Holders	60

The combustion chamber temperatures are varied according to the class of material used in the construction of the setting.

Thus with an all silica setting it would be possible to maintain temperatures between 1,420° and 1,480° C.

With siliceous retorts and silica combustion chambers, temperatures up to 1,400° C. are worked.

Before the introduction of the regenerative system, one of the chief causes of the loss of heat arose from the escape of the waste gases at unnecessarily high temperatures. In such cases the normal temperature at which the gases are allowed to leave the settings is in the neighbourhood of 1,700° to 1,800° F., but by the transference of the sensible heat from these gases to the inflowing secondary air the temperature of the waste gases has, in the modern regenerator, been reduced to 600°–800° F. before their escape is permitted. In the ordinary way, the secondary air approaches a temperature of 1,800° F. before combustion.

The introduction of cold producer gas made in mechanical producers external to the retort house necessitates a design of recuperator such that the secondary air and producer gas can both be preheated to conserve heat, but more especially to permit the necessary combustion chamber temperature of 1,400° C. to be attained.

Waste Heat Boilers.

Even with the installation of regenerators for preheating the secondary air it is found that the waste gases still leave the settings at a high temperature (600°–700° C.), and it is economic to install waste heat boilers for the transfer of this heat into steam.

The horizontal fire tube boiler is commonly used for this purpose, and the gases are drawn through the tubes at high velocity by a mechanically-propelled fan. The fan aids the control of combustion chamber temperatures by giving a steady 'pull' on the waste gas main.

Pressure up to 300 lbs. per sq. in. is now being employed. Depending on the temperature of the waste gases the evaporation on the boilers varies between 2 to 6 lbs. of steam per 1 lb. of fuel fed to the producers.

Horizontal Retort Benches.

The modern horizontal retort house consists of a number of beds each containing 8 to 10 retorts. It is now common to arrange that no more than 10 beds shall be built in one unit braced together so that this unit can in modern works be heated up, worked and cooled down as a unit. The wracking of settings, owing to uneven expansion and contraction, is thereby avoided.

The retorts are generally D shaped, though the elliptical retort gives a better distribution of heat to the coal and reduces the free space in the retort (*i.e.* over the coal and under the crown of the retort). Lately, experiments with narrow rectangular retorts have been tried. The rapidity of the heat transfer leads to greater output for each sq. ft. of ground area.

Most horizontal retorts are now built with silica segments shaped to form the D or ellipse. Such segments would be about 18 ins. long. Siliceous and fireclay retorts are extruded and can be obtained in lengths up to 11 ft. 0 in. The temperatures to which such materials can be heated are lower than the almost pure silica.

Retort Bench Fittings.

The following may be taken as giving the approximate size of ascension pipes necessary with horizontal retorts:—

Works making under 10 million cu. ft. per annum	**5-in. pipes.**
" " between 10 and 200 million cu. ft. per annum	**6-in. "**
" " above 200 million cu. ft. per annum	**7-in. "**

The above refer to retorts having an ascension pipe at both ends. If a single pipe is used it should not be less than 8 ins. in diameter.

Joints for Ascension Pipes.

These must on no account be made with lead. Many engineers will employ none but a rust joint, although the labour entailed in cutting out is considerable and the number of replacements is high. To prepare the material for these joints cast-iron borings should be damped down with gas liquor; the mixture is then rammed well down into the joint space. When the bench is at work the heat soon hardens off the joint. A serviceable joint may be made from the following:—

Ewell brick-dust	**1 part.**
Slaked lime	**2 parts.**

This is rammed into the joint dry, no moistening being necessary.

An excellent joint which combines the hardness and durability of the rust joint with easy cutting out is: The socket is packed with two turns of suitable size asbestos rope, filled with moist fireclay to within half an inch of the top of the socket and finished off with borings to form a rust joint.

For the joints between the cast-iron mouthpiece and the fireclay retort the following may be used: 1 part of cast-iron borings to 2½ parts of fireclay. This mixture is then damped down with a solution of sal ammoniac. The amount of sal ammoniac required is about 1 lb. to every cwt. of fireclay and borings. Ammoniacal liquor may be used in lieu of the sal ammoniac, but the latter is to be preferred as a more complete rusting action is thereby set up.

Hydraulic Mains.

These are now of two types:—

(*a*) Distinct mains, one to every retort bench on either or both sides.

(*b*) Continuous mains, one main extended over a number of settings.

The latter type have their advantages as regards initial cost, but distinct mains are otherwise to be preferred owing to the greater ease with which they are adjusted. In addition, by their use a single setting may be readily shut down and isolated.

The gas off-take from the hydraulics should be arranged from the cover of the box rather than from the side, as is so frequently found.

Owing to the difficulties involved in maintaining the same seal in the various sections of the hydraulic main, which causes either loss of gas due to heavy seals or in-leakage of air due to light seals, other systems of gas collection have been adopted.

With the 'Anti dip' system each dip pipe is fitted with a sleeve which can be raised or lowered from the outside. The retort is discharged with the sleeve down, *i.e.* retort sealed from the hydraulic main. As soon as the charge is laid the sleeves are raised and the retort will then be in direct communication with the gas space in the hydraulic main. Every retort is under the same conditions.

Another development is the use of the scrubber standpipe. The retort mouthpieces are directly connected to a standpipe which is increased in area in proportion to the volume of gas

passing to ensure maximum scrubbing effect and the maintenance of equal pressure on all retorts. At the top of the standpipe is a scrubbing chamber with baffle plates. The foot of the standpipe is connected to a steel box with provision for collecting foreign matter which might be pushed in from the mouthpieces. The tar and liquor flow to separating tanks in the cellar and the liquor is recirculated. It is claimed for this system that it is possible to control accurately the pressure conditions in the retort to within $\frac{1}{50}$ of an inch water gauge; that there is always a free passage for the gas from the retorts and that the irksome labour of cleaning ascension pipes is eliminated.

Retort House Foul Main.

The size of the foul main must depend upon the combined areas of all pipes leading into it. Generally speaking, the following sizes are those prevailing and which may be safely applied :—

Capacity of Bench per 24 Hours.	Size of Foul Main. If one.	If two.
30,000–100,000 cu. ft.	6 ins.–9 ins.	—
200,000–500,000 cu. ft.	12 ins.–15 ins.	—
1 million–1½ million cu. ft.	18 ins.	14 ins.
2 million–3 million cu. ft.	24 ins.	18 ins.

Another rule is that which states that the foul main (if it delivers all the gas) should be 125 per cent. of the area of the main works connections.

The modern practice is to construct the foul main of steel plates either welded or riveted together instead of cast-iron pipes. When steel is used, however, some form of expansion joint should be introduced, for the main is liable to undergo some considerable expansion when the settings are fired from cold. With the cast-iron mains, the lead joints at the spigots will be sufficient to cope with any change of length and flange joints should therefore be avoided, unless some form of expansion joint is inserted.

Tar and Liquor Removal.

The method employed for removing the tar and liquor condensed out of the gas in the hydraulic main is of extreme importance, for if an efficient principle is not employed endless trouble results, in the shape of pitched-up hydraulics, stopped ascension pipes and tar mains, and considerable loss of ammonia. The tar-tower system has been regarded as one of the best methods and is a considerable improvement on the older systems. The objection to the tar tower is that it is not automatic in action requiring periodical 'running off,' during which process there is every possibility of temporarily unsealing the dip pipes.

Retort House Governors.

The retort house governor is an essential auxiliary of the modern retort house. It differs from the station governor in that, whilst the latter is employed with the object of reducing pressure the retort house governor is interposed for the purpose of reducing the intensity of vacuum created by the exhauster. Thus the governor ensures a constant vacuum in the hydraulic main whatever the volume of gas coming away from the coal charges may be. With the bell type of governor the variations of vacuum are transmitted from the bell direct to the valve. Owing to the large size of bell and valve needed for large installations some sluggishness may be experienced.

In such cases it is common to use relay governors, *e.g.* Askania, Arca, Bryan Donkin, etc., in which the pressure changes operate a sensitive mechanism which uses a power-operated circuit using oil, water or gas to move a butterfly valve in the foul main.

The best place for the governor is on the top of the retort bench, as close up to the end of each section of the foul main as possible, for it is not so sensitive when interposed at the inlet or outlet of the condenser as is occasionally found. By-passes should be fitted to the governor, for it will require cleaning out every four months.

CONDENSERS.

The process of condensation is best divided into two sections: (1) Gas cooled to about 80–90° F. in either an atmospheric or water cooled condenser; (2) Gas cooled to about 60° F. in a water cooled condenser. The plant should be arranged so that the exhausters are placed between the two sections of condensers, as this layout ensures that the temperature of the gas at the inlet to the ammonia scrubbing plant is low and should enable the ammonia slip into the purifiers to be kept below 5 grains per 100 cub. ft. without the addition of water.

The table below shows the amount of moisture liberated and the heat to be removed per 1,000,000 cub. ft. (60° F. and 30 in. Bar) when saturated gas is cooled down to 60° F.

Gas Temp.	Vol. of Moist Gas.	Water deposited in cooling to 60° F.	Heat liberated in cooling to 60° F.
70	1,027,000	34·7 galls.	$0{\cdot}6 \times 10^6$ B.Th.U.
80	1,057,000	82·7 ,,	$1{\cdot}3 \times 10^6$,,
90	1,091,000	147 ,,	$2{\cdot}2 \times 10^6$,,
100	1,131,000	237 ,,	$3{\cdot}4 \times 10^6$,,
110	1,179,000	357 ,,	$4{\cdot}9 \times 10^6$,,
120	1,238,000	521 ,,	$6{\cdot}7 \times 10^6$,,
130	1,312,000	742 ,,	$9{\cdot}3 \times 10^6$,,
140	1,409,000	1,050 ,,	$12{\cdot}5 \times 10^6$,,
150	1,534,000	1,479 ,,	$17{\cdot}3 \times 10^6$,,
160	1,724,000	2,120 ,,	$24{\cdot}2 \times 10^6$,,
170	2,002,000	3,099 ,,	$34{\cdot}5 \times 10^6$,,
180	2,460,000	4,745 ,,	$50{\cdot}9 \times 10^6$,,

As can be seen the amount of heat to be removed from the gas rises rapidly with the temperature, and particular attention should be paid to the consideration of condensers if intermittent or continuous vertical plants are installed and steaming of the charge is practised.

The following areas to be allowed for condensing plant represent average practice :—

Atmospheric Condensers.

For air cooled annular condensers allow 5–10 sq. ft. per 1,000 cub. ft. per day.

Water Cooled Condensers.

The modern water cooled condensers are all of the water tube type. Horizontal water tube condensers are normally provided with surfaces of 1·5–2·5 sq. ft. per 1,000 cub. ft. per day. For a modern type of vertical water tube condenser the tube surface provided varies from 2·0–3·0 sq. ft. per 1,000 cub. ft. per day.

The design of water cooled condensers depends on the temperature and moisture content of the gas and the temperature and cost of the available water supply. The larger the cooling area the smaller the water supply needed. It is therefore necessary for each installation to be designed to meet local conditions.

EXHAUSTERS.

The type of exhauster used on gasworks depends largely on the size of that works.

For the smallest size the 2-, 3- or 4-bladed rotary exhauster is commonly used. It is driven by a slow-speed steam engine and gives remarkably trouble-free running provided it is given reasonable attention.

The following formula is a useful guide to the output of 4-bladed exhausters of medium size :—

$$Q = 65D^2LN\,;$$

where

Q = cub. ft. passed by the exhauster per hour.
D = inside diameter of outer drum in ft.
L = length of drum in ft.
N = number of revolutions per minute.

This gives the gross capacity. To arrive at the nett capacity, about 20 per cent. must be deducted (for slip) from the figure obtained.

The speed at which the exhauster operates should be gauged not by a consideration of the revolutions performed per minute but by the peripheral speed of the tips of the blades. This should be approximately constant for all sizes of exhausters and should lie within the limits of 650–850 ft. per minute.

The lubrication of bladed exhausters requires the most careful attention. The best type of lubricant is a mixture of refined creosote and a mineral oil in the proportions of one part of creosote to three parts of mineral oil. A vegetable oil should not be used as this tends to thicken the tar inside the drum.

On the medium size works there is a tendency to abandon the bladed rotary exhauster and to adopt the Roots blower or Connersville exhauster.

This consists of two cast iron impellers formed similar to figure eights rotating in opposite directions alternately trapping pockets of gas and discharging a definite volume at each revolution. The absence of frictional bearing surfaces within the exhauster obviates any necessity for internal lubrication and reduces the power consumption.

Since small high speed steam engines can be used with these exhausters, considerable economies, compared with the bladed type, can be expected in

(a) Ground space occupied.
(b) Steam consumption.

On the largest works and on coke oven plants the high speed turbo exhauster working on the fan principle and providing large clearances is commonly employed.

An advantage of this type is that it has some considerable volumetric capacity per unit of floor space occupied. The construction consists of an outer case provided internally with two or more impellers which rotate on a common axis but in distinct cells. The gas is passed from cell to cell in series until it is discharged from the machine at the requisite pressure, this being dependent upon the size and speed of the impellers. This machine has considerable power in removing tar fog and it can be used for the removal of tar from hot gas.

On the larger works it is usual to maintain a vacuum of about 4 ins. W.G. at the exhauster inlet. At the outlet the pressure to be overcome varies in accordance with the resistance offered by the apparatus following and the quantity of gas passing at that time. As a general rule the total back pressure may vary between 8 to 10 ins. W.G. on a small country works to as much as 50 ins. W.G. on the largest plants, but under average conditions the following pressures will approximate to those found at the different portions of the apparatus :—

Retorts	Level gauge.
Hydraulic main	½in.–1 in. vacuum
Exhauster inlet	4 ins. ,,
,, outlet	30 ins. pressure.
Tar extractor outlet	25 ins. ,,
Washers outlet	24 ins. ,,
Scrubbers outlet	22 ins. ,,
Purifiers outlet	9 ins ,,
Station meter outlet	8·5 ins. ,,
Holder inlet and outlet	8·5 ins. ,,

It will be seen from the above that the forcing of the gas through the dry purification material and the raising of the holder bell forms by far the greatest portion of the work against back-pressure which the exhauster is called upon to perform.

TAR EXTRACTION.

This is done (a) by the use of Livesey type washers in which the gas is forced in fine streams through a layer of a few inches of ammoniacal liquor operating on the wire drawing and bubbling principle; (b) by the use of such an apparatus as the Pelouze and Audouin extractor in which the gas is split into fine streams and impinges at high speed on steel plates. The tar particles, due to inertia, make contact with the plate and the tar flows away as a liquid. (c) By electrical precipitators in which the gas passing through a strong electric field is subjected to a silent or corona discharge. The static rectifier system using a direct current of 40,000–50,000 volts as a rectified sine wave gives excellent results even on hot coal gas or crude water gas. The efficiencies of tar removal with (c) are normally over 99·5 per cent.

WASHERS AND SCRUBBERS.

The capacity of the wet purification plant depends very largely upon the type of apparatus employed, whether mechanical or merely static. The static washer, which was being gradually superseded by the mechanical washer scrubbers, has again become prominent due to the improvement in packing material and the compactness of the circulating pumps. The Livesey type of washer is excellent as a tar extractor and a means of working up the liquor strength when placed immediately prior to some other form of final scrubber, but it is not sufficient on its own to reduce the ammonia content of the gas to the limit of 5 grains per 100 cub. ft. of gas which is common practice.

The Horizontal Mechanical Washer.—In this the gas passes through a number of compartments in each of which it has to pass over packing material fixed to a slowly rotating shaft which dips it in the washing medium.

The Vertical Mechanical Washer.—The gas rises through a cylindrical chamber and has to pass a series of contacts set up by liquid sprayed from members mounted on a rapidly rotating vertical axis.

Tower Scrubber.—The gas ascends a comparatively long tower and is met by a descending stream of liquid which is spread over solid packing elements to give even distribution and maximum contact between liquid and gas.

Static Washers.—These consist of a series of small chambers containing packing material through which the gas passes in series whilst the liquor is pumped over the packing material by means of circulating pumps. With all types of washers and scrubbers counter flow of gas and liquid should be employed and the temperatures of both should be kept as low as possible. The table below, taken from a paper read before the Institution of Gas Engineers in June 1929, shows the effect of temperature upon the possible concentration of ammonia in liquor.

Temperature ° F.	50	60	70	80	90
	Grains of NH_3 per 100 cu. ft. of gas.				
Ammonia in gas in equilibrium with:—					
1 oz. liquor . .	2	3	5	9	17
2 ,, ,, . .	3	6	10	20	35
3 ,, ,, . .	6	10	16	30	55
4 ,, ,, . .	7	13	20	39	69

It is possible to eliminate water from the scrubbing plant by obtaining a supply of liquor weak in free ammonia either from the retort house circulating system or the first section of condensers and by using this instead of water in conjunction with gas which has been cooled to a low temperature. (The normal amount of water added varies between 10–13 galls. per ton of coal carbonised.)

DRY PURIFICATION PLANT.

The capacity of the dry purifiers depends upon the extent to which purification is carried. Nowadays it is customary to remove sulphuretted hydrogen alone, but if lime purification (for the removal of carbon bisulphide) is resorted to, it is advisable to employ rather larger vessels. It is always as well, however, that the vessels should be too large rather than too small, for the first cost increases but slowly as compared with the capacity and purification charges undergo considerable reduction as the unit capacity becomes greater. For oxide boxes the following rule may be taken :—

For a set of four purifiers allow FOR EACH BOX an area of 0·5 sq. ft. per 1,000 cub. ft. of gas to be dealt with per maximum day. If lime is to be used increase this allowance to 0·6 or 0·65–0·7 if no catch boxes are available (Meade). Hunts' rule for the total amount of purification space required, *i.e.*, the combined area of all boxes, says that 20–30 sq. ft. per ton of coal carbonised per maximum day should be allowed.

It may be taken that the purifying capacity of one ton of oxide of iron is two million cub. ft. of gas before it is finally spent.

In order that the oxide of iron in the dry purifiers may be retained in an alkaline condition it is not advisable to remove the whole of the ammonia from the gas. The usual practice is to allow between 1–5 grains of ammonia per 100 cub. ft. of gas to remain.

Purifiers are invariably square or rectangular in shape. Cast iron plates bolted together are generally employed, but many installations built in reinforced concrete have been made. Some welded steel purifiers have also been constructed. As regards the necessary thickness of cast iron plates the following figures may be taken as indicative of the general practice :—

Purifiers up to 10 ft. square	. . . .	$\frac{5}{8}$ in. plates.
,, ,, 20 ,, ,,	. . . . .	$\frac{11}{16}$ in. ,,
,, above 20 ,, ,,	. . . . .	$\frac{3}{4}$ in. ,,

The flanges are usually cast $\frac{1}{16}$ in. to $\frac{1}{8}$ in. thicker than the body. The smallest vessels are usually 3 ft. deep while the largest sizes vary from 6–8 ft. in depth. The following rule is suggested for ascertaining the required depth :—

$$\text{Depth in ft.} = \sqrt[4]{\text{Total superficial area}} \times 1\cdot 1$$

The depth as given may be taken to the nearest foot. For smaller purifiers, *i.e.* up to 300 sq. ft. in area multiply by 1·3 instead of 1·1.

It is advisable to stiffen the sides of purifier boxes by the casting of vertical rib stiffeners at about 2 ft. 6 ins. centres on the plates.

TOWER PURIFIERS.

The latest developments have been in the direction of reducing the ground space occupied by an installation. One method is to use very deep boxes with oxide in eight tiers each 16 ins. deep, arranged so that the flow of gas can be frequently reversed. Another attempt at reducing the superficial area of the plant has been tried at the Wandsworth and District Gas Company's main works in the 'Tower' purifiers. These consist of six towers each of which has an outer

shell 22 ft. in diameter and 52 ft. in height, closed at the top and bottom. The top cover is removable. Each tower contains 12 superimposed containers filled with oxide; each of these is polygonal and holds two tiers of oxide supported on wooden grids. Each container consists of a central tube and two oxide containing trays. The gas is introduced between the layers of oxide from a centre feed and passes through a single layer of oxide to the annular space between the trays and the body of the tower and thence to the outlet. It is claimed that this method required only one-third of the area covered by the old ground level boxes and less than two-thirds the area covered by deep overhead boxes.

The capital cost is comparatively low and the handling charges should be less than normal due to the increased use of machinery. For this reason also the time during which a tower is out of action is small.

Purifier Valves.

The modern practice is to construct new purifiers without catch boxes. It is customary to work 'backward rotation,' *i.e.* to alter the sequence of boxes thus :—

1234 4123 3412 2341 1234

the time between changes depending on the condition of the oxide in the boxes, the amount of gas passed and its sulphuretted hydrogen content. This time varies from hours to about five days. It is essential that the changes shall be made rapidly and easily and that the valves shall remain gastight since the Gas Referees test paper is darkened if the H_2S content of the gas is more than 1 part in 500,000 parts.

For small installations the centre valve working on the principle of the plug cock is employed.

For larger units the connections can be brought to one central box divided into inlet and outlet sections. The gas is controlled by disc valves. The discs may provide iron to iron connections or may be packed with white metal or greasy hemp packing. The best known of this centralised type is the Weck valve. The Milbourne valve employs disc valves but has not the central box.

The hydraulic valve employing either U-shaped connections or a box with a midfeather has been used for many years and has been employed on many modern large installations.

On all types of purifiers it is possible to instal the ordinary double-faced or single-faced gas valves instead of using the types given above. Care should be given to maintenance of all valves as faulty seatings will quickly allow foul gas to go forward to the holders.

Purifier Connections.

The diameter of the connecting mains is most readily and accurately computed from the following formula :—

$$\text{Diameter of pipe in ins.} = \sqrt{\text{area of each box in sq. ft.}}$$

For small purifiers (up to 60 sq. ft. area) the result can be taken as it stands, but for medium size boxes deduct 16 per cent. and for large vessels 25 per cent.

Modern purifiers are of the dry lute type which does not blow under temporary additional pressure and is not subject to freezing in the winter.

The cover is constructed of steel or wrought iron plate about ¼ in. thick, supported by some form of trussing. The joint between the cover and the box is made by inserting a strip of rubber or tallowed hemp between the parallel flanges, the two flanges being afterwards pulled together by means of special fasteners or ordinary bolts.

With modern purifiers it is customary not to have the boxes at ground level so that discharging can be quickly done by unloading the oxide through convenient openings in the bottom of the box on to the ground. The rate of recharging can be made equally rapid if a storage floor is provided over the box or rapid mechanical elevators and conveyors are available.

The effect of the elevation of the boxes has been the chilling and subsequent lack of activity of the oxide in the box.

The purifier preheater has been designed to minimise this trouble. In general it consists o a nest of steam-heated tubes around which the gases circulate before purification.

Insulation of the lids by blankets filled with glass wool or fibre glass has proved of advantage in maintaining the boxes in an active state.

Station Meters.

Station meters are for measuring the production of gas or the sale of gas in bulk. They may be divided into three classes.

(1) Positive displacement meters.
(2) Semi-positive meters.
(3) Inferential meters.

Positive Displacement Meters.

The drum type is widely used as a station meter. It is extremely accurate over its whole range. In this meter the water level is fixed just about the centre and gas enters through the hollow axis in the four compartments of the drum, filling each in turn. The pressure of the gas causes the

drum to rotate and operate the integrating device. The Holmes 'B.M.' meter is also of the wet type, using oil or water as a sealing fluid. The method of metering is the filling and emptying of three or more chambers which form part of a roughly spherical drum. This drum is supported on a central pivot and runs on an outer circular rail. The drum itself does not revolve but its axis performs a circle and thereby gives the drive to the counter mechanism. This meter is being increasingly used for metering the gas consumption of large industrial users.

Semi-Positive Meters.

The best known of this class—the 'Holmes-Connersville'—is similar in construction to the exhauster of the same name. Two impellers, figure 8 in section, are geared together and supported on bearings within a cast iron casing. The gas entering the top exerts a pressure on the impellers causing them to rotate. In each revolution the gas trapped between the impeller and the casing is allowed to pass. Since the impellers do not touch the casing there is some slip. The meters have an accuracy within 1 per cent. The space covered by the meter is small and the pressure loss across the meter about ½" W.G.

Inferential Meters.

These are the rotary fan type, the thermal meter or differential pressure meter.

STORAGE CAPACITY.

The holder capacity necessary is somewhat dependent upon whether or not the works is provided with a water gas plant. If such a plant is available the storage allowed for need not exceed 18 hours maximum make, but if there is no water gas to fall back upon provision should be made for 21 to 24 hours maximum production. In years gone by it was customary to provide considerably greater storage chiefly on account of the heavy fluctuations in output. The levelling up of the hourly load-curve has, however, had a marked effect in the direction of diminishing the quantity of gas kept in hand.

PROPORTIONS OF GASHOLDERS.

Single lift height = 0·3 to 0·4 of diameter.

Multiple lift height = 0·6 to 1·0 of diameter.

In computing the capacity of a holder the volume enclosed by the crown is neglected, as the gas contained therein is not available for distribution. Accordingly

$$\text{Capacity in cub. ft.} = 0{\cdot}7854 \times \text{diameter in ft.}^2 \times \text{height in ft.}$$

With multiple lift holders the capacity of each lift should be estimated separately by applying the above formula to each section.

DEPTH OF LIFTS.

Each lift will be approximately of equal depth.

RISE OF CROWN.

The greater the rise the stronger will be the crown. The dome should conform to a segment of a sphere. Normally, the rise will range between $\frac{1}{15}$th to $\frac{1}{20}$th of the diameter, but when trussing is employed it may be less than when the plain crown only is used.

The cubic capacity of the crown space may be found from the following :—

$$\text{Volume in cub. ft.} = {\cdot}5236R\left(R^2+\frac{3D^2}{4}\right),$$

where,

R = rise of crown in ft.;

D = diameter of top lift in ft.

The depth of each lift should never be less than one-seventh of its diameter, otherwise locking or tilting may occur. In general, the proportion varies between one-fourth to one-fifth of the diameter.

The crown sheeting of gasholders varies from 14 gauge to 10 gauge (about $\frac{5}{64}$ in. to ⅛ in.); but much depends upon the size of the holder.

The side sheeting varies from 12 gauge to 10 gauge, but curb or cup and grip rows are thicker, ¼ in. to ⅜ in. Top lift sheets are usually thicker than those of succeeding lifts, often about 8 gauge.

The thinner sheets of the bell are generally riveted up cold with ¼ in. rivets at a pitch of 1 in. to 1¼ in. All rivets above ⅜ in. must be driven hot.

GASHOLDER TANKS.

Steel tanks have many advantages over the masonry variety, and cast-iron as a material for this purpose may now be looked upon as obsolete.

The thickness of plate required may be calculated as follows :—

$$T = \frac{PD}{2fe},$$

where

T = thickness of the plate in ins.

D = diameter of the tank in ins. ;

f = allowable working stress (usually $5\frac{1}{2}$ tons per sq. in.) ;

e = the efficiency of the vertical riveted seams. This will range up to 90 per cent. for large holders. For small holders it will vary between 75 and 85 per cent. ;

P = the pressure due to the water in lbs. per sq. in.

The latter may readily be found from the following :—

$$P = WD,$$

where,

W = the weight of a cub. ft. of water = 62·4 lbs. ;

D = depth below the surface of the section under consideration.

As regards the apportionment of the cost of gasholders, the following figures represent normal conditions :—

	Per Cent. of Total Outlay.
Bell, complete	35 to 45
Guide framing	18 to 23
Tank (steel)	35 to 45

Calculation of Pressure Thrown.

The pressure thrown is dependent upon the weight of the floating portion and its diameter. It may be found as follows :—

$$\text{Maximum pressure (ins. of water)} = \frac{549W}{D^2},$$

where,

W = weight of bell and water in lutes, in tons

D = diameter of holder in ft.

When dealing with a multiple lift holder take the mean diameter.

Waterless Gas Holders.

These holders comprise a vertical continuous shell of circular or polygonal cross-section having its two ends closed respectively by a bottom plate and a roof, the latter being usually fixed rigidly to the shell. A movable piston guided by wooden rollers slides up and down inside the shell, rising and falling to the change in contents of the holder. Additional pressure is obtained by concrete weights affixed to the piston deck. Waterless holders varying in capacity from 50,000 to 20 million cub. ft. are in use.

The various types of waterless holders differ mainly with regard to the method of sealing the piston and the shell. The 'Klonne' holder effects the seal by means of rubber impregnated cotton fabric interspersed and covered top and bottom with leather. The pad is lubricated by a special lubricant introduced through cups fitted to it at intervals. The 'Bamag-Meguin' holder employs a leather tube filled with oil from a reservoir placed in the centre of the piston of the holder. In the 'M.A.N.' holder the seal between the piston and inner shell is effected by an annular trough containing a quantity of prepared tar equal to not less than twice the pressure of the gas underneath the piston. Contact with the shell surface is by steel rubbing bars kept in position with counterweighted levers. This tar seal prevents gas leakage, lubricates the piston and preserves the interior surface of the shell.

The following advantages are claimed for the Waterless holder :—

1. Constant pressure.
2. Less weight than with water tank holder and consequently cheaper foundations.
3. Easy to extend while in use and more rapid construction and erection.
4. Eminently suitable for storage of dehydrated gas.
5. No risk of freezing up in cold weather.
6. No immersion of plates in water, reduced corrosion and lower painting costs.

Pressure Gasholders.

With the extension of the supply of gas to thinly-populated districts the problem of economical distribution of gas has arisen. One solution is to have a small pressure gasholder filled during

off-peak periods and which will feed back into the distributing system during the periods of heavy loads.

Pressure holders are normally built in units from 10.000 to 250,000 cub. ft. releasable capacity. Pressures range from 50 to 70 lbs. per sq. in. but higher pressures can be employed. As the cost is almost independent of the number of units, one can be installed initially and additions can be made later.

The following points are claimed for pressure gasholders :

(1) They are small and compact and can be cylindrical or spherical to suit the site.

(2) Can be used in high pressure distributing systems or to supply peak demands in low pressure systems.

(3) They are of heavy plate construction and those of the cylindrical shape can easily be protected against war conditions.

(4) Low painting costs ; low foundation costs ; no risk of freezing and no moving parts.

(5) Installations can be built to be entirely automatic in operation.

Inspection of Gasholders.

The examination of gasholders was rendered compulsory by the Factories Act, 1937.

Clause 33 refers specifically to gasholders :—

(1) Every gasholder should be of sound construction and shall be properly maintained.

(2) Every gasholder shall be examined externally by a competent person at least once in every period of two years and a record containing the prescribed particulars of every such examination shall be entered in or attached to the General Register.

(3) In the case of any gasholder of which any lift has been in use for more than twenty years, the internal state of the sheeting shall, within two years of the coming into operation of this section and thereafter at least once in every period of ten years, be examined by a competent person by cutting samples from the crown and sides of the holder or by other sufficient means, and all samples so cut and a report on every such examination, signed by the person making it, shall be kept available for inspection.

(4) A record signed by the occupier of the factory or by a responsible official authorised in that behalf, showing the date of construction as nearly as it can be ascertained of the oldest lift of every gasholder in the factory shall be kept available for inspection.

(5) Where there is more than one gasholder in the factory, every gasholder shall be marked in a conspicuous place with a distinguishing number or letter.

(6) No gasholder shall be repaired or demolished except under the direct supervision of a person who, by his training and experience and his knowledge of the necessary precautions against risks of explosion and of persons being overcome by gas, is competent to supervise such work.

(7) In this section the expression 'gasholder' means a water-sealed gasholder which has a storage capacity of not less than 5,000 cub. ft.

The Fourth Report of the Gasholder Committee of the Institution of Gas Engineers (*Transactions* 1937–39), gives the particulars to be noted at each examination to conform with the requirements of the Factories Act. Inspection of gasholders is undertaken by contractors or by the Insurance Companies specialising on engineering business.

COAL STORAGE.

At the present time it is necessary for gas works to keep in hand much larger stocks than were thought necessary some years ago. Whilst it is better to store coal under cover if possible, no serious deterioration takes place if it is stored in the open and the problem of coal handling is simpler as such devices as the drag line scraper can be used. Due to the oxidation of coal with a consequent rise in temperature the selection of coal for storage should be made with care.

Some coals are more prone to spontaneous combustion than others. The coal heaps should, if possible, be kept below 15 ft. in height, but 20 ft. should not be exceeded.

To enable the temperatures at the base of the coal heaps to be taken weekly, wrought iron pipes, 2 in. diamenter, with closed spiked ends, should be placed in the heaps before storage.

A thermometer can be lowered down these tubes. If it is found that the temperatures are rising above 100° F. the coal at these points should be moved and if possible carbonised.

It appears from researches carried out on the effects of storage on coal (*Trans. Inst. Gas Eng.*, Communication 201, by Jamieson and Skilling) that whilst the volumetric yield is but little altered, the calorific value and hence the thermal yield were reduced. The quality of the coke was also poorer.

Coke Storage.

One ton of coke occupies 85 cub. ft. (average).

For the convenience of dealing with stocks of other materials used on gasworks the following figures are given :—

One ton of broken coal occupies	40 cub. ft.
,, ,, cannel coal occupies	35 to 45 cub. ft. (varies considerably)
,, ,, anthracite coal occupies	35 to 39 ,,
,, ,, breeze occupies	60 to 65 ,,
,, ,, lime occupies	42 ,,
,, ,, oxide of iron (Dutch)	45 ,,
,, ,, ditto (prepared)	41 ,,
,, ,, sulphate of ammonia	47 to 48 ,,
,, ,, ashes	50 ,, (about).
,, ,, Portland cement	23 to 26 ,,
,, ,, loose earth	28 to 30 ,,

In gas undertakings the revenue from coke and breeze represents 25 per cent. of the total income and increasing care is given to the preparation of this solid smokeless fuel. In normal times the coal to be carbonised is carefully selected for its gasmaking and coking qualities. Durham and Somerset coals produce dense blocky cokes—these are ideal for steam raising and central heating plants. Yorkshire and Lancashire coals are less strongly coking and produce a more open coke which is more suited to use in the domestic grate. Coals from the Notts and Derby coalfields produce a less dense coke but the proportion of breeze is higher. Scottish coals are generally of poor coking power.

Preparation of Coke.

The demands of the local market for coke will bear very largely on the coals to be carbonised and more particularly on the type of plant in which such carbonisation will be effected. Static carbonisation produces coke which is less reactive but denser than that produced in continouus vertical retorts.

From the point of discharge from the retort until the placing of the coke in the consumers' store care must be taken to prevent the following :—

1. Combustion of coke due to insufficient quenching.

2. Excessive dust formation if insufficiently quenched. (A moisture content of about 5 per cent. is required to effect this.)

3. The degradation of the fuel by rough handling. The value of 1 ton of coke is at least three times the value of 1 ton of breeze.

Coke is therefore transported by rubber belt conveyors, screened with the minimum of violent movement and lowered into storage hoppers by inclined or spiral shoots. It is usual to store coke in the sizes in which it is sold. The types of screens used are :—

1. *Static.*—These require ample headroom and are generally unsatisfactory.

2. *Rotary Screen.*—The cylindrical screen is placed at an angle of 5°–10° and the coke descends. The efficiency of the screen is limited as so much of the mesh is inactive at any given time.

3. *Conveying Screen.*—These jigging screens convey and screen at the same time. The upkeep is high but they have the advantage of delivering into the correct hoppers without the use of belt conveyors.

4. *High Speed Vibration Screens.*—These have one, two or three decks and are most efficient.

With all screens handling coke, high upkeep costs must be expected, due to the abrasive nature of the material handled.

The storage facilities provided in bunkers should be adequate to store the coke made over the week end.

If a large amount of coke has to be stored in the summer in order to meet the winter demand, consideration should be given to stocking graded coke in its various sizes. Dumping should be by specially designed plant which 'places' the coke on the heap and reclamation by a Smith-Harmer loader which not only picks up the coke but also debreezes it as well.

The sizes of coke recommended by the Institution of Gas Engineers Committee of Enquiry on Coke Quality (1942) were :—

Name.	No.	Sizes : Square Mesh Screen.
Large or unbroken .	1	No upper limit. Over 1¾ ins.
2 ins. × 3 ins. . .	1a	Within limits 3½ in. to 1¾ ins.
Broken	2	,, ,, 2 ins. to 1 in.
Coke boiler nuts . .	3	,, ,, 1¼ ins. to ½ in.
Forge coke . .	4	,, the limits ¾ in. to ⅜ in.
Screened breeze . .	5	Unwashed fuel approximately ⅜ in. to ⅛ in. from which fines up to ⅛ in. have been removed.

It is also recommended that cokes be known by numbers.

Products of Distillation.

One ton of average gas coal will yield (approximately) the following products :—

Gas	14,500 cub. ft., or approximately 72·5 therms.
Coke	12 to 13 cwts., excluding breeze.
Breeze	1 cwt.
Tar	9 to 11 gallons.
Motor Spirit	2 to 3 gallons.
Ammoniacal liquor	32 to 40 gallons at 8 ozs.
Sulphate of ammonia	24 to 28 lbs.
Sulphur	5 to 7 lbs.
Cyanogen (HCN)	4 lbs.
Virgin liquor	10 to 12 gallons.

The yield of gas as given above is the normal quantity of straight coal gas obtained, but it should be borne in mind that the modern vertical retort in which 'steaming' is employed will give a yield of from 85 to over 100 therms per ton of coal.

(The sulphate of ammonia is obtained from the ammoniacal liquor, and not in addition to it. The virgin liquor is included in the figure given for total yield of ammoniacal liquor. The figure given for sulphur is that which may be recovered during purification of the gas, and does not include the quantity remaining behind in the coke.)

The approximate distribution of the elements of the original coal, and the manner in which each exists after the process of carbonisation, is illustrated in the following table :—

One ton of coal, consisting of :—

Carbon, 1,792 lbs., distributed as—	Lbs.
Coke	1,385
Breeze	107
In gas, as CO and CO_2	30
In gas, as hydrocarbons	130
In tar	122
As scurf, cyanide, etc.	18
Hydrogen, 123 lbs., distributed—	Lbs.
In gas and tar	98
As ammonia	1·0
As water	18·5
In coke	5
Oxygen, 197 lbs., distributed—	
In gas, as CO, CO_2, and combined in tar	49
As water	148
Nitrogen, 34 lbs., distributed—	
In gas (free)	11·8
In coke	14·6
In tar (combined)	1·0
As ammonia	5·8
As cyanogen	0·8
Sulphur, 18 lbs., distributed—	
As gaseous impurities and in liquor	6
In coke	12
Ash remaining in coke and breeze	75
Total . . .	2,240

Cost of Manufacturing Coal Gas.

The cost of manufacturing town's gas depends primarly upon the capacity of the works, its geographical position relative to suitable coal-fields, and the character of the plant in use. It must be borne in mind, too, that the actual cost of production (*i.e.* cost into gasholder) represents only less than one-half of the ultimate selling price, the remainder being accounted for by the cost of distribution, overhead charges, dividends, etc.

Gasworks Conveyors and Elevators.

The formula given below may be taken as giving a useful guide as to the capacity of bucket conveyors :—

$$\text{Capacity} = \frac{S \times B \times W}{96{,}768 \times P} \text{ tons per hour,}$$

where,

B = capacity of buckets in cub. ins. ;
S = speed of conveyor in ft. per minute ;
W = weight per cub. ft. of substance carried ;
P = pitch of buckets in ft.

The horse-power required to operate elevators and conveyors may be readily arrived at by adopting a coefficient of friction in accordance with the conditions under which the machine is working.

Let M = the weight of material carried in lbs.
W = the weight of the moving parts in lbs.
K = the coefficient of friction.
S = the speed of the conveyor or elevator in ft. per minute,

then, for

(*a*) Vertical elevator:

$$\text{H.P.} = \frac{M \times S}{33{,}000} + \frac{K \times S(M+W)}{33{,}000};$$

(*b*) Inclined elevator:

Let $a°$ = the angle of inclination with the horizontal, then,

$$\text{H.P.} = \frac{M \times S \times \sin a°}{33{,}000} + \frac{K \times S(M+W)\cos a°}{33{,}000};$$

(*c*) Horizontal conveyor:

$$\text{H.P.} = \frac{K \times S(M+W)}{33{,}000}.$$

As regards the value of K, the following figures may be taken:

0·15 for metal sliding on metal :
0·07 to 0·08 „ „ (well lubricated) ;
0·25 for wood sliding on wood.

Purification of Coal Gas.

The purification of coal gas is carried out in two stages, namely, by means of wet purification in the washers and scrubbers and afterwards in the dry purifiers. During the wet purification of the gas the alkaline ammonia is made use of for removing a portion of the acidic impurities, sulphuretted hydrogen and carbon dioxide. The two last-named are, therefore, absorbed with the formation of certain salts, which are themselves soluble in water. The resultant solution consists of water containing a certain proportion of these salts, and is known as ammoniacal liquor. In crude coal gas the ammonia content is usually in the neighbourhood of 1·5 per cent. by volume, and of this amount from about one-third to one-half will have been deposited prior to the gas reaching the wet purification plant. Hunt has given the following figures for the distribution of ammonia :—

	Per cent. of total.
Ammonia removed by condensation	42·7
„ „ first scrubber . . .	43·3
„ „ second scrubber . . .	14·0

When gas containing carbon dioxide and sulphuretted hydrogen comes in contact with a solution of ammonia both acid and normal salts are formed. Thus we have :

(*a*) $2NH_3 + H_2O + CO_2 = (NH_4)_2CO_3$ (normal) (*c*) $2NH_4OH + SH_2 = (NH_4)_2S$ (normal) $+ 2H_2O$.
(*b*) $NH_3 + H_2O + CO_2 = NH_4 HCO_3$ (acid) (*d*) $NH_4OH + SH_2 = NH_4 HS$ (acid) $+ H_2O$.

Composition of Gas Liquor.

The ammonia present in gas liquor is combined in two distinct ways. In the first place it is present as easily decomposable compounds such as carbonates and sulphides, and secondly, as

more stable compounds which will only undergo decomposition by heating in the presence of a caustic alkali. In general, the final mixture of gas liquor obtained consists of from 75 to 80 per cent. of the more unstable salts, i.e. 'free' ammonia, whereas the remaining 20 to 25 per cent. is made up of the more stable salts, i.e. 'fixed' ammonia. The possible constituents of gas liquor are enumerated below, although it must be borne in mind that any one sample of liquor will seldom contain all of the compounds given. In fact, some of those included are only met with on rare occasions.

Water, containing in solution:—

Ammonium carbonate	'Free' ammonia.	Ammonium chloride	'Fixed' ammonia.
„ bicarbonate		„ sulphocyanide	
„ carbamate		„ sulphite	
„ sulphide		„ thiosulphate	
„ hydrosulphide		„ sulphate	
„ cyanide		„ ferrocyanide	
„ polysulphide		„ acetate	

Also phenols, amines, and other nitrogenous bodies.

The amount of ammonia in gas liquor is in reality quite small, and varies between 1·5 and 2 per cent. In general, it will be apportioned as follows:—

	Per cent.
'Free' ammonia	1·2 to 1·6
'Fixed' ammonia . . .	0·3 to 0·4

The following figures relate to sulphuretted hydrogen and carbon dioxide:—

	Per cent.
Carbon dioxide in gas liquor .	1·3 to 2
Sulphuretted hydrogen . .	0·1 to 0·6

The distribution of the total ammonia in gas liquor varies, of course, to some considerable extent. The following table, however, refers to the average solution:—

	Per cent. of total ammonia.
Ammonium sulphides	6
„ carbonates	74
„ chloride	15
„ sulphocyanide	1·75
„ thiosulphate, sulphite, etc.	3
„ sulphate	0·25

The most effective manner of operating the washing and scrubbing plant is to ensure that the gas on first entering the apparatus is treated with the strongest liquor available. In each succeeding vessel the washing liquor employed becomes gradually weaker and weaker until, in the final scrubber, clean water should be employed. Where circumstances permit, it is preferable that the first washer should be operated with the liquor from the hydraulic main, for in this way both CO_2 and SH_2 may be considerably reduced.

The amount of water which must be added for the purpose of wet purification seems to vary to some considerable extent in accordance with the conditions prevailing.

The writer has found with Durham coal that the ammonia may be completely removed in ordinary coke filled tower scrubbers by adding 10–13 gallons per ton of coal carbonised. When mechanical washer scrubbers are in use the quantity of water required will be considerably less. If the gas is cooled after the exhauster and liquor having a free strength of under '1 oz.' is used, it should be possible to eliminate water from the washing plant.

Ammonia Lost.

The quantity of ammonia lost in the direction of that permitted to go forward from the final scrubber may be conveniently calculated as so many gallons of 10 oz. liquor.

$$\left.\begin{array}{l}\text{Quantity of 10-oz. liquor}\\ \text{lost per week (in gallons)}\end{array}\right\} = \frac{\text{Total grains of ammonia passing forward from scrubber per week.}}{1{,}522}$$

Impurities in the Gas at various stages.

Outlet of condensers—	Grains per 100 cub. ft.
Ammonia	200 to 300
Sulphuretted hydrogen	600 „ 900
Carbon dioxide	900 „ 1,400
Sulphur compounds	20 „ 45
Hydrocyanic acid	60 „ 80
Naphthalene	20 „ 25

Inlet of purifiers—	Grains per 100 cub. ft.
Ammonia	nil to 1·5
Sulphuretted hydrogen	450 „ 700
Sulphur compounds	20 „ 45
Hydrocyanic acid	50 „ 70
Carbon dioxide	750 „ 1,150

Outlet of purifiers—	Grains per 100 cub. ft.
Ammonia	nil
Sulphuretted hydrogen	nil
Carbon dioxide	750 to 1,150
Sulphur compounds	20 „ 45
Hydrocyanic acid	nil „ 10

(The last three named above may all be removed, or partially removed, by the employment of special processes, but in the majority of gasworks such processes are not resorted to.)

Removal of Sulphuretted Hydrogen.

Sulphuretted hydrogen is now almost universally removed by means of oxide of iron in one of its forms. Natural ferric oxide (bog-ore) is still largely employed for the purpose, although in recent years various artificial preparations have found a good deal of favour, chiefly owing to their lower cost. The natural bog ore is chiefly obtained from deposits in Holland, Belgium, and Ireland. In the dried state its composition varies between the following limits :—

	Per cent.
Hydrated ferric oxide (Fe_2O_3, H_2O)	60 to 65
Organic matter	15 „ 25
Silica	3 „ 6
Alumina	1

The active material in bog-ore and artificial preparations is not actually the iron oxide, but the oxide plus a certain quantity of water of combination, i.e. a hydrated ferric oxide. In addition to this moisture the material as used should contain a further quantity of water, usually from 20 to 30 per cent. The artificial oxides are employed with a similar degree of moisture. The made-up or artificial material, if of good quality, is usually rather more active than the natural product, owing to a rather higher content of ferric hydrate, which may reach so much as 70 to 75 per cent.

Reactions during the Removal of Sulphuretted Hydrogen.

The absorption of H_2S by the active oxide occurs in two distinct ways :—

$$(a)\ Fe_2O_3, H_2O + 3H_2S = Fe_2S_3 + 4H_2O.$$
$$(b)\ Fe_2O_3, H_2O + 3H_2S = 2FeS + S + 4H_2O.$$

In the first instance ferric sulphide is formed, whilst secondly the result is ferrous sulphide and some free sulphur.

When spent and inactive the material is removed from the purifier and subjected to the process of revivification, or exposure to the air, when the sulphides are oxidised to oxides and the material is ready for further use. The following are the revivification reactions :—

$$(a)\ 2Fe_2S_3 + 3O_2 + 2H_2O = 2Fe_2O_3, H_2O + 3S_2.$$
$$(b)\ 4FeS + 3O_2 + 2H_2O = 2Fe_2O_3, H_2O + 2S_2.$$

Oxide is generally laid in the purifiers on grids to a depth of from 8 to 10 ins. In some works, however, it is customary to employ layers as deep as 48 ins. In this case hurdle grids would be used to help break up the oxide.

A space of at least 2 ins. should be left between the top of one layer and the base of the sieves or grids of the layer immediately above. Oxide, as it becomes fouled increases in bulk, hence this precaution. Partial revivification *in situ* is nearly always arranged for in present-day systems. This is effected by admitting a small measured quantity of air to the purifiers along with the gas. The air may be taken from a small compressor and forced into the box or may be injected by means of steam. In other cases it is sucked in with the gas through a pipe attached to the main on the inlet side of the exhausters. When air is admitted at some point prior to the washers and scrubbers it should not be overlooked that undesirable reactions may be set up in the wet purification plant.

It is generally understood that the material in any one purifier cannot simultaneously extract sulphuretted hydrogen and undergo revivification. The assumption is not entirely accurate, but the process of sulphiding takes place at a very much more rapid rate than does the oxidation effect.

If the capacity of the purification plant is not ample the full effect of the admitted air will not be realised. The volume of oxygen required to revivify is given by the equation—

$$2H_2S + O_2 = 2S + 2H_2O.$$

Thus 0·5 per cent. of oxygen is sufficient to oxidise to sulphur 1 per cent. of sulphuretted hydrogen. As the H_2S content of the gas does not often exceed 1½ per cent. by volume 0·75 per cent. of oxygen or 3·7 per cent. of air is all that is required to produce this action in the purifiers. To obtain the full effect of the air, excess over that required theoretically must be admitted. On account of the reduction in calorific value due to the dilution of inert gases it is not generally desirable to add more than 3 per cent. air.

It is essential that the amount of air admitted to the gas stream shall be carefully controlled and apparatus is installed at some works for the purpose. The amount of air required is decided by chemical analysis and the system of governing applied maintains the same proportion of air whatever the fluctuations in make.

With continuous vertical retorts admission of excessive air will cause the calorific value of the gas to fall; the calorific value of the gas in the retort house will be raised by reduction in steaming and hence the volume of gas and the thermal yield from the plant will fall. Thus, with a continuous vertical retort house supplying to the district gas having a calorific value of 460 B.Th.U./cub. ft., if the unpurified gas contained 0·9 per cent. H_2S by volume and 4 per cent. air is admitted it can be shown that the reduction of air to 3 per cent. will allow for an increased thermal yield of 1·8 per cent.

Removal of Naphthalene.

If the gas is not treated for the removal of naphthalene on the works, it will in general contain from 6–12 grains per 100 cub. ft. Butterfield has pointed out that a million cub. ft. of gas cooled from 68°–50° F. may deposit 14 lbs. of naphthalene. The introduction of carburetted water gas plants has to a certain degree mitigated the naphthalene evil although its effect in this direction is somewhat erratic. Carburation with the aid of solvents was introduced by Botley, who injected, by means of a special spray, about 4 gallons of thoroughly atomised paraffin per million cub. ft. of gas. Botley found that petroleum oil was more suitable for the purpose than spirits such as benzol.

Where oil-washing plants for the extraction of benzol are installed the reduction in the naphthalene content of the gas will be automatic if the oil in circulation is maintained in good condition by frequent make up of clean oil or continuous tapping of the fractions containing the naphthalene.

Paraffin, water gas tar, anthracene oil and gas oil have all proved effective in stripping the naphthalene from the gas if an efficient washer is provided. The rate of oil circulation is quite low—20 gallons per million cub. ft. of gas washed.

The Recovery of Crude Benzol.

The 'stripping' of gas for the recovery of the aromatic series of hydrocarbons is now the regular practice in the majority of gas works. The coal gas, either in the semi-crude state (*i.e.* after the wet purification plant) or preferably when completely purified, is subjected to washing with a solvent, gas oil, green oil or spindle oil. The gas is washed with the solvent in a mechanical or static washer the normal rate of circulation being 100 gallons per ton of coal carbonised. The 'benzolised' oil which may contain up to 5 per cent. of benzene hydrocarbons is then taken to the still in which it is subjected to steam distillation. The benzol vapours with steam and other naphtha fractions are passed through a dephelegmator (this is a tubular heat interchanger), wherein the temperature being lowered by preheating the incoming oil the naphtha fraction is thrown down and is collected for sale or re-use. The remaining vapours pass up the fractionating column and the quality of the spirit is controlled by a water-cooled analyser. Only spirit of a low boiling point can pass through as vapour, the remainder being thrown back in a liquid state for further fractionation. The vapours are condensed and the water removed. The yield of benzol depends upon the rate of oil circulation and washing efficiency, but it should be possible to recover 3 gallons of crude spirit per ton of coal carbonised. The crude spirit will vary in composition, but in general 65–70 per cent. will distil over up to 120° C. and 80 per cent. up to 160° C. The portion which distils over will consist of about 80 per cent. of benzol, 15 per cent. of toluol and the remainder xylol.

The efficiency of benzol extraction depends on: (*a*) The completeness of stripping in the distillation plant; (*b*) Quality of the oil (freedom from water and naphthalene); (*c*) Temperature of the cooled oil. This should not exceed 80° F. and should be maintained 5° F. above the temperature of gas; (*d*) Quantity of the oil in circulation. The more oil is used the greater the efficiency; (*e*) Absorptive capacity of the wash oil used.

Spindle oil is used on many works in preference to gas oil, as no sludging of the oil is experienced. With gas oil the steam distillation gradually gets rid of the lighter fractions and with a sudden fall in temperature setting of the oil takes place. The yield of benzol with spindle oil as the absorber is slightly lower than with gas oil. Due to its higher viscosity some alteration to the design of plant may be required if maximum circulations are always worked.

The crude benzol is treated for the reduction of gum-forming hydrocarbons and sulphur impurities by washing with caustic soda and sulphuric acid prior to redistillation. The product obtained is well within the limits of National Benzole Association's Specification.

The Active Carbon Process.

The advantages claimed for this method are: (1) The benzol is recovered direct and is free from wash oil and impurities derived therefrom. The cost of refining is reduced. (2) A higher efficiency of recovery of the benzol from the gas. (3) Smaller heat and power requirements. The disadvantages of the system are: (1) The process is intermittent. (2) Naphthalene content of the gas must be low and may entail the use of oil washing plant for its extraction.

The gas carrying the adsorbable hydrocarbon vapours is brought into contact with activated carbon which retains the benzol and certain other constituents of the gas, *e.g.* unsaturated hydrocarbons, carbon disulphide. Owing to the liberation of heat cooling water is circulated. When saturated the adsorber is cut off from the gas stream and closed and then direct steam applied. The benzol vapours are collected and the process restarted. The plant at the Beckton works of the Gas Light & Coke Co. is designed to treat 75 million cub. ft. of gas in 24 hours and to produce 16,000 gallons of benzol.

Many plants for small and medium size works have recently been installed. It is claimed that with new or reactivated carbon the efficiency of benzol recovery may attain 99 per cent., and for sulphur removal 65 per cent.

These smaller size plants are designed to be automatic, the cycle of operations being controlled from a series of cams rotated at constant speeds. Some chemical supervision on the plant is required to see that, as the carbon ages, the necessary alterations in operating cycle are made. The consumption of steam on this type of plant is comparable with the ordinary plant employing wash oil and distillation under normal conditions. It is, however, in excess of that type of plant employing vacuum distillation and normal rates of oil circulation.

Dehydration of Gas.

Of the gas distributed in Great Britain one-third or 110,000 million cub. ft. is treated for the removal of water vapour. The advantages claimed for this are: (1) A saving in the cost of pumping syphons. (2) The prevention of water troubles in consumers' pipes and fittings. (3) The prevention of internal corrosion of mains and services with consequent stoppages. (4) Prevention of corrosion in meters.

The two methods employed on a commercial scale use either calcium chloride or glycerine as a dehydrating agent. Glycerine has a vapour pressure lower than water and the solution of calcium chloride in water has a low vapour pressure and is able to absorb moisture from gases having a greater partial pressure of water vapour.

The gas is washed with either medium in an efficient washer, the dewpoint of the outlet gas being the factor to be controlled.

With the calcium chloride process some of the diluted solution is tapped off and run over an evaporator and back to the main supply through a cooler. The strength of the washing solution is thus kept correct. With the glycerine process the glycerine is treated under a vacuum for the removal of the water vapour absorbed.

The cost of each process is very low.

Trouble has been experienced with gum deposit in apparatus on the district of some undertakings that have gas drying plants. This has been traced to the action of the very small quantities of oxides of nitrogen upon the unsaturated hydrocarbons in the gas. It is found that, where gas drying is not practised or where it takes place after storage in a holder where condensation can and does occur, no such trouble is experienced. With the present degree of knowledge it is recommended either (1) to instal gas dehydrating plants on the outlet of holders, or (2) to arrange to pass gas through a holder even if the main storage is for 'dry' gas.

Reduction of Organic Sulphur Compounds in Town Gas.

After the complete removal of sulphuretted hydrogen from gas, sulphur is still present as organic compounds. A considerable research into the problem of their elimination is yielding success and it is hoped that in the near future many commercial plants will be erected. The scope for gas, both in the domestic and industrial fields, will thereby be extended, and maintenance costs of apparatus reduced.

The organic sulphur compounds in purified (not debenzolised) coal gas are:—

Carbon disulphide . . .	15 grains per 100 cub. ft.
Carbon oxysulphide . . .	6 ,, ,, ,, ,,
Thiophen	9 ,, ,, ,, ,,
Mercaptan	2 ,, ,, ,, ,,
Total	32 ,, ,, ,, ,,

These figures are an average, but it is correct to say that with continuous vertical retorts a lower sulphur content is experienced than in gas from static carbonisation at a higher temperature. (It should be noted that the sulphuretted hydrogen contents of the two unpurified gases are in the reverse order.)

Very few plants designed expressly for the reduction of sulphur compounds are in use at present, but the extensive removal of such compounds is incidental in two methods of benzol recovery.

(*a*) ***Oil Washing.***—This method has already been described on p. 638. When dealing with coal gas it is found that a circulation of 8 galls. of solvent (gas oil) per 1,000 cub. ft. should suffice for the absorption of benzene and of all the thiophen and its higher homologues, whilst for the complete extraction of carbon disulphide a circulation of 32 galls. of solvent per 1,000 cub. ft. would be needed. Thus the normal oil-washing process with a circulation of 8 galls. of oil per 1,000 cub. ft. reduces the sulphur compounds by 40 per cent. whilst extracting between 2½ to 3 galls. of benzol per ton of coal carbonised. The use of the increased oil circulation, *i.e.* 32 galls. of solvent per 1,000 cub. ft. of gas washed results in excessive steam consumptions unless the distillation plant is specially designed. The Gas Light & Coke Company and Messrs W. C. Holmes have produced a plant employing this high oil circulation. The distillation is carried out at a temperature of only 170° F. in a partial vacuum. The steam consumption is similar to that of a normal plant. The efficiencies of benzol and sulphur removal are 90 per cent. and 75 per cent. respectively.

(*b*) *Active Carbon Benzol Absorption Process.*—This again is designed for benzol extraction. It is more efficient than the normal oil washing process in the removal of sulphur compounds from the gas. In this process, which is described in greater detail under 'Benzol,' the process of adsorption is carried out in vessels for periods of about 8 hours. During the early parts of the adsorption process the whole of both the thiophen and the carbon disulphide is removed, but as saturation of the carbon is approached the more volatile compounds (*e.g.* carbon disulphide) are displaced. Over a period of six months continuous working of the Beckton plant of the Gas Light & Coke Company, the sulphur content of the gas was reduced from 30·5 grains per 100 cub. ft. to 6·1 grains.

(*c*) *Catalytic Removal of Organic Sulphur Compounds from Coal Gas.*—A full scale plant for the catalytic removal of organic sulphur compounds from coal gas has been at work at the Harrow works of the Gas Light & Coke Company since 1938. The plant is fully described in a paper published in the *Gas Journal*, dated July 12, 1944.

The gas is brought into contact with the nickel subsulphide catalyst on china clay pellets after being treated for the removal of sulphuretted hydrogen and napthalene and being preheated in a heat interchanger. The temperature of the gas entering the catalyst vessels is maintained by burning some of the hydrogen in the gas by the addition of air. The gas then passes out through the heat interchanger, through a vertical washer cooler, wherein it is washed with a soda ash solution and a catch purifier box for the elimination of H_2S.

The table below shows the behaviour of the individual sulphur compounds in the Harrow plant.

ORGANIC SULPHUR IN GRAINS PER 100 CUB. FT.

		Mean Temperatures of Gas Leaving Catalyst.		
		280° C.	310° C.	360° C.
As Carbon disulphide	inlet	15·1	14·7	16·3
,, ,, ,,	outlet	5·0	6·4	0·9
,, ,, oxysulphide	inlet	8·3	6·0	6·1
,, ,, ,,	outlet	4·5	4·0	2·5
,, Thiophen	inlet	10·5	8·8	10·2
,, ,,	outlet	10·4	7·4	10·2
,, Mercaptan	inlet	1·2	3·9	1·1
,, ,,	outlet	0·5	1·3	0·0
Total	inlet	35·1	33·4	33·7
,,	outlet	20·0	19·1	13·6

When this gas is treated in an active carbon benzol absorption plant the sulphur compounds are further reduced to under 5 grains per 100 cub. ft.

The authors of the paper estimate that the cost in 1939 of a plant of 2 million cub. ft. of gas capacity would have been £7,500.

Running costs daily would have been:—

Interest and depreciation	394 pence per day.
Therms in H_2 consumed at 4*d*. per therm	458 ,, ,, ,,
Soda ash (0·73*d*. per lb.)	131 ,, ,, ,,
Labour and supervision	120 ,, ,, ,,
Power and light	33 ,, ,, ,,
	1,136 ,, ,, ,,

As the thermal output is 8,400 therms per day the cost of running the plant is 0·135 pence per therm. This figure includes the regeneration of the catalyst but represents working with a good load factor.

Use of Plant for Treating Furnace Atmospheric Gases.

Where there are special demands for a gas with a low content of total sulphur, as for example in the production of special furnace atmospheres, the catalytic process has proved very satisfactory for small scale operations. Some 40 units of this type have been in operation in various parts of the country for periods up to four years and the catalyst has frequently operated over this time without renewal or replacement. These small plants are fitted with external heating.

Contamination of Town Gas in Storage.

It is frequently found that gas freed from sulphuretted hydrogen stored in a water-sealed holder becomes contaminated due to the production of sulphides by the bacterial reduction of sulphates. The trouble is most acute in the autumn when the atmospheric temperature is lower than the gas holder water temperature. The water on the surface is chilled, sinks to the bottom and that at the bottom rises to the surface.

A simple and effective cure is aeration of the water. The water should be taken from the surface inside the tank, be passed over an aerating tray and returned to the space between the tank side and the outer lift. This scheme has the objections of the smell of the holder water and the fact that the sulphides are not removed but merely returned to sulphates upon which the bacteria feed. Chemical oxidants used are calcium hypochlorite, chlorine and hydrogen-peroxide. All these oxidant processes require accurate control to ensure that the holder water is not rendered unduly corrosive towards steelwork.

Precipitants such as ferrous sulphate, copper sulphate, zinc oxide and zinc acetate can be used to remove from the water in insoluble form either the sulphides or sulphates from which they are formed. The introduction of sulphates such as ferrous sulphate or copper sulphate is undesirable and zinc compounds are best employed (1) since zinc being electro positive to iron is unlikely to affect the rate of corrosive attack ; (2) zinc sulphide is not decomposed by coal gas ; (3) the addition of the oxide or acetate does not involve the introduction of undesirable acid radicles.

The use of bactericides to kill the bacteria would be the best solution of the problem but to date no satisfactory bactericide has been found. Naphthalene prevents the growth of bacteria but must be present in a concentration high enough to impart a considerable amount of naphthalene to the gas. Phenol must be employed in impracticably high concentrations.

Tar Distillation.

(Contributed by Harold Moore, M.Sc., A.I.C., F.C.S.)

The yield of coal tar depends very largely upon the volatile content of the coal; but, in addition, it is considerably affected by the design of the retort. Retorts in which heating is conducted gradually, and which ensure the rapid removal of the volatile products from the heat zone, yield the largest quantities of tar. The nature of the tar depends, to a slight extent, upon the type of coal used, but is much more seriously affected by the carbonising conditions. A rapid rise in temperature, with a high maximum temperature, tends to produce tar high in specific gravity, high in viscosity, rich in free carbon, and mainly consisting of aromatic hydrocarbons. Gradual heating, low maximum temperature, and rapid removal of the volatile products result in brown mobile tars, comparatively devoid of free carbon and rich in aliphatic hydrocarbons.

Horizontal retorts subject the coal to a very rapid heating with a high maximum temperature, whilst the volatile products have to pass along the heated retort walls, whereby they undergo cracking. The result is a viscous tar, rich in free carbon. Horizontal retort tars frequently contain over 20 per cent. They possess a specific gravity of about 1·2. In continuous vertical retorts the coal is slowly passed from the cool zone into a hotter zone, undergoing gradual heating, whilst the volatiles are removed from the cooler end of the retort, and, therefore, are not so liable to undergo cracking. The resulting tar possesses a specific gravity of about 1·08 and about 1 to 8 per cent. of free carbon. Inclined retorts give products of intermediate composition; whilst coke ovens yield tars of very varied composition, depending upon the design of the ovens.

Special processes for the low-temperature carbonisation of coal, such as the Coalite and Premier Tarless systems, yield brown low-gravity tars in which free carbon is almost entirely absent. Gas works and coke ovens supply nearly all the tar produced in this country. Tar produced in gas works is accompanied by ammoniacal liquor, and requires settling out previously to

distillation. The tar in distillation is usually divided into the following fractions in English distilleries :—

	Temperature.	Approx. Yield.
1. Crude naphtha	Up to 110° C.	3 per cent.
2. Light oil	110° to 200° C.	7 ,,
3. Crude carbolic oil . . .	200° to 240° C.	7 ,,
4. Creosote oil	240° to 270° C.	15 ,,
5. Anthracene oil	270° to finish	10 ,,
6. Residuum of pitch . . .	—	58 ,,

The finishing-point of the distillation is decided by the quality of pitch desired. The yields of the above products vary, of course, with the nature of the tars, horizontal retort tars yielding a residue of about 65 per cent. of pitch, whilst vertical retort tars give only about 50 per cent. of pitch. The benzene and toluene are mostly present in the crude naphtha fraction, the yield of benzene being about 2 gallons to the ton of tar. They are separated by distillation combined with washing with dilute alkali, which removes phenols, cresols, and other acidic bodies, followed by washing with dulite acid, which removes pyridine, quinoline, and other basic substances, and are finally refined by treatment with strong sulphuric acid, which removes certain sulphur-containing compounds and a portion of the unsaturated hydrocarbons originally present in the tar. Similar processes of refining are adopted for other fractions, though the strong acid treatment is usually applied only to the benzene, toluene, and solvent naphtha-containing fractions. The most common grades of these three distillates are sold under the names of 90*s*. benzol, 90*s*. toloul, and 90*s*. xylol. The 90*s*. benzol distils 90 per cent. of its volume at 100° C. by the tar distillers' retort test; 90*s*. toluol yields 90 per cent. at 120° C. by the same test; whilst 90*s*. solvent yields 90 per cent. at 160° C. The solvent fraction consists mainly of the three isomeric xylols. Naphthalene and anthracene are removed from the higher boiling fractions (naphthalene mainly from the carbolic oil fraction, anthracene from anthracene oil) by cooling and decantation, or by cooling and filtration.

The limiting temperatures of the fractions are liable to alteration, depending upon market conditions, whilst the distillation is frequently stopped at an earlier stage than that indicated in the above table, in order to yield a soft residue known as treated or refined tar, which is used for tar spraying on the roads. A soft finished residue is also frequently sold as pitch-creosote mixture, and used for the fuel market.

Benzol is mainly used for motor fuel, together with a considerable quantity of the toluol, though smaller amounts are employed in the synthetic chemical industry. Carbolic acid (phenol) and cresylic acid (cresols), obtained by alkali washing, both find their main market for the manufacture of disinfectants. Creosite oil is used for timber preservation and as fuel. The use of creosote as fuel is somewhat intermittent, depending upon the price of petroleum. Pitch is mainly used for road repairing, but also for many other purposes, a considerable amount being consumed for electric insulation.

WATER GAS.

Water gas plants give a high output of gas at calorific values between 205 and 600 B.Th.U. per cub. ft. within a few hours of starting up. These plants are normally used as auxiliaries to coal gas plants to provide for sudden peak loads caused by abnormal weather conditions. The reactions occurring in the generator of the water gas plant are as follows :—

$$(a)\ C + 2H_2O = CO_2 + 2H_2 - 33{,}170 \text{ B.Th.U.}$$
$$(b)\ C + H_2O = CO + H_2 - 52{,}900 \text{ B.Th.U.}$$

The carbon dioxide produced in the lower portion of the fire according to the equation (*a*) is then reduced to carbon monoxide on passing through the remainder of the fuel bed—

$$(c)\ CO_2 + C = 2CO - 72{,}000 \text{ B.Th.U.}$$

The extent to which the reaction (*b*) takes place depends largely upon the temperature prevailing in the lower part of the fuel bed. As the above reactions are of an endothermic nature the heat of the fuel bed is gradually absorbed, accordingly this is replaced at intervals by the process of 'blowing.' The reaction occurring during the 'blow' is of an exothermic nature being simply as follows :—

$$C + O_2 = CO_2 + 172{,}560 \text{ B.Th.U.}$$

The modern water gas plant consists of: (1) Generator fitted with a self-clinkering grate and annular boiler. (2) Carburettor fitted with a creeper attachment to the air main. (3) Superheater. (4) Waste heat boiler for cooling of waste gases.

The cycle worked on such a plant would be about 3 minutes' duration, of which 45 per cent. is used for the 'blow.' The 'down run' has now been superseded by the 'back run,' in which the

steam is admitted to the top of the superheater and travels backward through the vessels carrying heat back to the bottom of the fire. The 'back run' process has resulted in reduced coke consumption and increased gas production.

The modern plant is entirely automatic and the cycle of operations can be adjusted to meet the conditions of each works. The whole subject has been thoroughly investigated and reported upon by the Joint Research Committee of the Institution of Gas Engineers. (See *Trans. Inst. Gas E.*, 1930, 1932, 1934, 1935.)

A brief summary of their conclusions is given below :—

1. The charges for labour and capital are so important that when using C.W.G. (Carburetted Water Gas) as a coal gas auxiliary the total costs of manufacture are at a minimum when a plant is operated at high output. To obtain this

(a) the blast pressure should be such that with the maximum fuel bed temperatures allowed by clinker formation and wear and tear of grate the heat in the blow gases leaving the generator is no more than is required for carburetting ;

(b) the rate of steam supply to be as high as possible considering the quality of blue water gas made unless the rate of oil supply thus required leads to an excessive loss of efficiency.

2. The rates of air and steam admission are decided as shown above, the relative durations of the run and blow periods are fixed by the necessity of storing sufficient heat in the fuel bed during the blow to decompose the steam during the run. It is assumed that the fuel is of adequate density to withstand the blast pressure.

3. During periods when a low output of gas is required economies in raw materials and maintenance can be obtained as follows :—

(a) Clinker formation and hence wear and tear of grate are reduced by use of lower blast pressures.

(b) Improved decomposition of steam and oil and hence better quality gas result from slower rates of steam and oil admissions.

(c) When outside steam is expensive the plant can be made self supporting by increasing the heat in the blow gases.

4. A large proportion of the steam should be passed downwards through the fuel bed as there is then less tendency for heat to leave the generator in the blow gases, and there is also an improvement in quality of blue water gas.

5. With coke of normal grade nothing is gained by alteration in depth of fuel bed under steady conditions. The limits are set by

(a) A thin fuel bed increases the sensible heat loss in the blow gases leaving the generator ;

(b) too deep a fuel bed restricts unduly the air supply.

6. Coke of normal large grade (50 per cent. 3 in.–2 in.) gives a better performance than smaller grades which give a smaller output, lower thermal yield and poorer quality of blue water gas.

7. Slower rotation of the grate and reduced ejection of ashes improves the quality of blue water gas, the output and thermal efficiency as a greater proportion of steam can be admitted on the back run. If insufficient ash is extracted clinker troubles will follow.

8. Self-clinkering generators operate at a greater output than hand clinkered machines and at the same time give a greater thermal yield of blue water gas per 1,000 lbs. of coke and at a better quality.

9. The chief advantage of the generator boiler on a self-clinkering generator is the reduction of charges per therm C.W.G. until it can be established that a similar output can be obtained from a generator of the same size without a boiler. On outputs below peak the presence of the boiler is advantageous under circumstances in which it is economical in raw materials to maintain the plant self supporting in steam.

10. To obtain the highest output and the greater yield of blue water gas from the coke it is specially important to secure complete admixture of the blow gases and secondary air at the top of the carburettor.

For the manufacture of carburetted water gas having a calorific value of about 450 B.Th.U. per cub. ft., the materials required are as follows :—

Coke	32–40 lbs. per 1,000 cub. ft. of gas.
Oil	1·5 gallons of oil per 1,000 cub. ft. of gas.
Steam	0–30 lbs. per 1,000 cub. ft. of gas.

A set fitted with generator and waste heat boilers can be self-supporting in steam.

The cost of carburetted water gas is mainly influenced by the cost of the enriching oil, and for calculation the yield from 1 gallon of oil can be taken as 1·25 therms.

The impurities in water gas consist of carbon disulphide and sulphuretted hydrogen, which are present in the following quantities :—

	Grains per 100 cub. ft.
Sulphuretted hydrogen	110 to 120
Carbon disulphide	10 to 15

As regards water gas tar, S. Carter has given the following analysis :—

Specific gravity	1·0665
	Per cent.
Benzol	1·11
Naphtha	2·33
Creosote oil	21·35
Naphthalene	4·09
Anthracene oil	15·91
Pitch	56·70

SULPHATE OF AMMONIA.

Average neutral commercial salt contains :—

Ammonia	25·5 per cent.
Equal to	
Nitrogen	21·0 per cent.
Free acid	nil.
Moisture	0·5 per cent.

Materials required for the manufacture of 1 ton of sulphate of ammonia :—

Liquor, approximately	3,400 gallons of 8 oz. strength.
Acid ,,	18 cwts. at 80 per cent. strength.
Lime ,,	3 cwts.

The sulphate of ammonia recovered per ton of coal varies in this country between 24 and 28 lbs.

Equation for formation :—

$$2NH_3 + H_2SO_4 = (NH_4)_2SO_4.$$

Gallons of liquor × ounce strength × 0·0823 = lbs. of sulphate.

Concentrated Gas Liquor.

Due to the fall in the selling price of ammonium sulphate from £14 in 1924 to as low as £5 per ton in 1932 many of the smaller plants for its manufacture were run at a loss. Whilst the selling price for ammonium sulphate has to some extent improved, it is found more profitable for the smaller undertakings to distil the liquor and to manufacture concentrated gas liquor containing up to 20 per cent. ammonia by weight. The subject is dealt with fully by P. Parrish in a paper to the Institution of Gas Engineers (*Trans.* 1938–1939). He gives a cost statement for the production of concentrated gas liquor in a plant having a capacity of 4,000 tons of concentrated gas liquor containing 20 per cent. ammonia by weight per annum.

Steam.—To produce 1 ton of concentrated gas liquor containing 20 per cent. NH_3 by weight at 96 per cent. efficiency 13·9 tons of gas liquor needed

Steam consumption = 18 per cent. by weight on crude liquor handled.

$$\text{Steam needed} = \frac{18}{100} \times 13{\cdot}9 = 2{\cdot}52 \text{ tons.}$$

	Cost/Ton.
Supplied at 15 lbs. per sq. in. at 12 pence per ton	2*s.* 6*d.*
Supervision	9*d.*
Cooling water	3*d.*
Repairs and maintenance	6*d.*
Depreciation at 7½ per cent. and interest at 5 per cent.:—	
Capital cost £5,600.	
Annual charge £700	3*s.* 6*d.*
Overheads and contingent expenses	1*s.* 0*d.*
Total manufacturing costs per ton	8*s.* 6*d.*

$$\text{Cost per unit of ammonia} = \frac{8s.\ 6d. \text{ per ton}}{20 \text{ units}} = 5d.$$

USEFUL MEMORANDA.

A Calorie is the amount of heat required to raise 1 kilogramme of water 1° Cent.; correctly from 0° to 1° Cent. This is the *large* calorie, which is the unit employed on gas works.

1 British Thermal Unit	= 0·252 calories.
1 Calorie	= 3·968 B.Th.U.
1 Therm	= 100,000 B.Th.U.
Therms	= cub. ft. × calorific value per cu. ft. ÷ 100,000.
Cu. ft.	= therms × 100,000 ÷ calorific value per cu. ft.

Calorific Power of Various Substances.

	B.Th.U. per lb.		B.Th.U. per lb.
Coke-oven coke	14,436	Petrol, specific gravity 0·684	20,923
Peat, 5 per cent. water	9,900	Shale oil	18,217
Wood, pine	9,153	Blast-furnace oil	16,080
„ oak	8,316	Heavy tar oil	16,050
Petroleum, American	19,627	Coal gas tar	15,500
„ Texas	19,242	Alcohol, absolute	12,931
„ Russian	19,440		

(*Lewes.*)

Gasworks Terms.

A barrel of tar	25 gallons	A fodder of lead	19½ cwts.
A butt of liquor (Usually 8 oz. strength)	108 „	A hundred of deals	120 in number
A bushel of coal	80 lbs.	A load of lime (A bushel = 70 lbs.)	32 bushels
A bushel of coke	45 lbs.	A long thousand of slates	1,200 in number
A chaldron of coal (Varies according to locality)	25½ cwts.	A seam of glass	120 lbs.
A chaldron of coke (Varies according to locality)	12½ cwts.	A short ton (American)	2,000 lbs.
A cord of wood	128 cub. ft.	A tonne (French)	2,204·6 lbs.
		A ton of freight	40 cub. ft.
		A truckload of bricks	About 3,300 in number

Volumes of Oxygen and Air required for Combustion of One Volume of Combustible Gas.

Combustible Gas.	Oxygen.	Air containing 21 per cent. Oxygen.	Volume of Products of Combustion.
Hydrogen	0·5	2·38	1 vol. H_2O
Carbon Monoxide	0·5	2·38	1 vol. CO_2
Marsh Gas	2·0	9·52	1 vol. CO_2, 2 vols. H_2O
Ethylene	3·0	14·28	2 vols. CO_2, 2 vols. H_2O
		Incomplete Combustion.	
Marsh Gas	1·5	7·14	1 vol. CO, 2 vols. H_2O
Ethylene	2·0	9·25	2 vols. CO, 2 vols. H_2O

Flow of Gas in Mains.

Pole's Formula.

Let Q = the discharge of gas in cub. ft. per hour.
d = the diameter of pipe in inches.
p = pressure of gas in inches of water,
s = specific gravity of gas, air being 1.
l = length of pipe in yards.

then,

$$Q = 1350d^2\sqrt{\frac{p\,d}{s\,l}}\,;\; d = \sqrt[5]{\frac{Q^2 s\,l}{(1350)^2 P}}\,;\; p = \frac{Q^2 s\,l}{(1350)^2 d^5}\,;\; l = \frac{(1350)^2 d^5 p}{Q^2 s}.$$

Unwin's Formula for High Pressure Gas Mains.

Let D = diameter of pipe in ft.
S = specific gravity of gas (air being 1).
L = Length of pipe in ft.
P_1 = initial pressure in lbs. per sq. inch absolute.
P_2 = final pressure in lbs. per sq. inch absolute.
C = coefficient of friction $= 0{\cdot}0044\left(1+\frac{1}{7D}\right)$.
U_1 = initial velocity of gas in feet per second.
U_2 = final velocity of gas in feet per second.
W = area of pipe in sq. ft.
Q = discharge in cub. ft. per hour,

then,

$$Q = 1{,}323{,}250\sqrt{\frac{P_1^2 - P_2^2}{P}\cdot\frac{P_1^2}{P_2^2}\cdot\frac{D^5}{CSL}}\,;\; U_1 = 468\sqrt{\frac{D}{CSL}\cdot\frac{P_1^2 - P_2^2}{P_1^2}}\,;\; U_2 = \frac{P_1}{P_2}U_1.$$

Table of Constants for Friction in Gas Mains.

Diameter of Main in Ins.	Diameter of Main in Ft. = D.	Area of Main in Sq. Ft. = W.	Value of C = Coefficient of Friction $= 0\cdot0044\left(1 + \frac{1}{7D}\right)$
2	0·166	0·022	0·0082
3	0·250	0·049	0·007
4	0·333	0·087	0·0063
5	0·417	0·136	0·006
6	0·500	0·196	0·0057
7	0·583	0·267	0·00554
8	0·666	0·349	0·00537
9	0·750	0·442	0·00527
10	0·833	0·545	0·00515
12	1·000	0·785	0·005
14	1·166	1·069	0·00492
15	1·250	1·227	0·00488
16	1·333	1·396	0·00485
18	1·500	1·767	0·0048
20	1·666	2·182	0·00475
21	1·750	2·405	0·00473
24	2·000	3·142	0·0047

(Professor Unwin.)

Flow of Gas in Service Pipes.

To determine the correct size of service pipe to be installed or to determine the maximum rate of flow from a pipe, the table evolved by Lacey and given below should be used (*Trans. Inst. Gas E.* 1923).

This table gives the flows in cub. ft. per hour corresponding to $\frac{1}{10}$ in. loss of pressure between the ends of various lengths of straight pipes of nominal diameters from ⅛ in. to 3 ins. Specific gravity 0·50.

Length of Pipe (Ft.)	Nominal Diameter of Pipe (Ins.).								
	⅛	¼	⅜	½	¾	1	1¼	1½	2	
10	3	<u>14</u>	33	72	130	240				
20		7	<u>18</u>	49	88	165	340	530		
30			12	<u>38</u>	70	130	270	420	890	
40				29	60	110	235	360	770	2240
50				23	<u>52</u>	98	210	320	680	1980
60					47	89	190	290	620	1800
70					40	<u>81</u>	170	260	560	1630
80						75	160	240	520	1510
90						71	150	230	490	1420
100							<u>140</u>	215	460	1340
150							110	170	370	1070
200							96	<u>150</u>	310	900
250								130	280	795
300								115	250	710

The underlining of certain figures in the table indicates the greatest length of pipe of a particular diameter which in practice, it has been found desirable to lay.

Bends, toes and elbows can be allowed for by adding to the overall net length of pipe the lengths given in the table below. The gross length obtained can then be used in the preceeding table.

Nominal Diameter of Pipe.	Addition to be made to Over-all Length of a Pipe (in ft.) for Increased Resistance introduced by Fittings.		
Ins.	Elbows.	Tees.	90° Bends.
$\frac{3}{8}$ to 1	2	2	1
$1\frac{1}{4}$ to $1\frac{1}{2}$	3	3	$1\frac{1}{2}$
2	5	5	2
3	8	8	3

Use of Gas in Industry.

In recent years the amount of gas used for industrial purposes has steadily risen and it is estimated that, during 1943, 98,000 industrial consumers used 103,160 million cub. ft. of gas or 28 per cent. of the whole gas sold in this country. Some of the typical processes employing gas are given below:—

Industry or Trade.	Process.
Iron and Steel	Billet heating for rolling and forging. Foundry—Core and mould drying, ladle heating. Annealing. Rolling mills. Forging.
Non-ferrous Metals	Melting, annealing, forging, stamping, brazing.
Engineering (Automobile and Aircraft)	Every kind of heat treatment, brazing, stoving paint-stoving ('infra-red' process), running-in-engines.
Engineering (Electrical and Radio)	Manufacture of cable, wire, lamps, valves, instruments. Soldering.
Engineering (Light Alloys)	Melting, refining and every form of heat treatment.
Engineering (General)	Annealing, normalising, carburising, hardening, plate cutting.
Tube and Wire	Billet heating, annealing, patenting, galvanising bright annealing. Heating of solutions, tanks and lead pots.
Agriculture	Grain drying, dairy equipment.
Ceramic	Firing of earthenware, china, porcelain. Drying.
Chemical and Chemical Plant	Heating of stills and evaporators. Soap manufacture. Paint and varnish manufacture. Dye manufacture.
Cremation	Furnace heating.
Food, etc.	Large-scale bread and biscuit baking. Confectionery. Brewing.
Glass	Melting, annealing, working, finishing.
Plastics	Platen heating.
Printing	Type-metal melting.
Textile	Singeing of cloth and yarn.
Turkish Baths	Production of steam, vapour and hot air.

Gas Meters.

The original consumers' gas meters were of the wet type similar in operation to the Wet Station Meter.

The original 'lights' type dry meter has been improved in the 'Standard,' High Capacity, and later in the small High Capacity I.G.E. types. The improvements have been made in the design of diaphragm, valve gear and general layout. The table below gives the capacity and sizes of the various types of meter. The table shows, particularly in the larger sizes, the reduction in the space occupied by the later types.

Size or Denomination.			Capacity. Cub. ft./hr.	Height in ins. approx.	Base Dimensions.	Space occupied. Cub. ins. approx.
Lights.	Standard	High Capacity.				
3	—	—	18	15½	9¼ × 7	1,003
5	No. 1	30	30	17	12 × 8	1,630
—	No. 2	—	40	14	11½ × 8	1,290
—	—	50	50	$10\frac{11}{16}$	9 × 6¼	602
10	No. 3	—	60	16	13 × 9	1,870
—	—	100 I.G.E.D.I.	100	12	9⅜ × $7\frac{3}{16}$	809
10	No. 4	—	120	18	18 × 12	3,888
20	No. 4	—	120	24	18 × 12	5,184
—	—	125	125	15⅛	$12\frac{3}{16}$ × $8\frac{9}{16}$	1,580
30	—	—	180	26	18½ × 12½	6,010
30	No. 4A	—	180	21	18½ × 12	4,660
—	—	I.G.E.D.2	200	15⅛	$12\frac{3}{16}$ × $8\frac{9}{16}$	1,580
—	No. 5	—	210	25	21 × 15½	8,150
—	—	250	250	18⅝	15⅝ × 10½	3,050
50	—	—	300	32	23 × 15½	11,400
—	No. 5A	—	315	30	26 × 20	15,600
60	—	—	360	32	23 × 18	13,260
—	—	I.G.E.D.4	400	18¾	16½ × 10½	3,250
—	No. 6	—	420	31	25½ × 20	15,800
—	—	450	450	$18\frac{13}{16}$	16½ × 10½	3,260
80	—	—	480	40	29 × 22	25,520
—	No. 7	540	540	35	29 × 22	22,350
100	—	—	600	42	29 × 22	26,800
—	—	700	700	25¾	22 × 15	8,500
120	No. 8	—	720	47	34½ × 27½	44,520

A similar table for flange meters is given below.

Size or Denomination.			Capacity. Cub. ft./hr.	Height overall. Ins.	Base over Flanges.	Space occupied. Cub. ins. approx.
Lights.	Standard	High Capacity.				
150	9	—	900	52	42 × 29	63,300
200	10	1,200	1,200	56	44 × 30	73,800
250	—	—	1,500	58	45 × 35	91,300
—	11	—	1,800	71	51 × 40	144,800
—	—	1,800	1,800	37½	34½ × 23	29,800
300	—	—	1,800	63	48 × 40	121,000
400	—	—	2,400	74	56 × 46	191,000
500	—	—	3,000	76	57 × 48	208,000
—	12	—	3,000	82	63 × 54	279,000
—	—	3,000	3,000	45	42 × 27	51,000
600	—	—	3,600	80	63 × 50	252,000
—	—	4,500	4,500	53	46 × 31	76,000
—	—	6,000	6,000	57	49 × 31	86,000
—	—	9,000	9,000	65	56 × 36	131,000
—	—	15,000	15,000	71	61½ × 46	201,000
—	—	25,000	25,000	89	70 × 56	349,000
—	—	50,000	50,000	85½	87½ × 51	382,000

Gas Tests.

Gas Tests as Prescribed by the Gas Referees.

As gas is now sold on the therm basis the determination of calorific value is the most important of the official tests. In addition, tests are made for pressure and freedom from sulphuretted hydrogen.

Tests for Calorific Value.

The apparatus to be used for ascertaining the calorific value of the gas supplied by Gas Undertakers who have sold in the preceding year more than 100 million cub. ft. of gas must include a calorimeter for the production of a continuous record of the calorific value of the gas supplied. The instrument must be one of the following three:—

(1) The Boys Recording Calorimeter.
(2) Fairweather Recording Calorimeter.
(3) Cambridge Scientific Instrument Company's Thomas Recording Calorimeter.

The apparatus to be used for testing the calorific value of gas supplied by undertakers who have not sold 100 million cub. ft. of gas in the preceding year is a Boys Non-Recording Calorimeter. As an alternative to a recording instrument of the above types the Gas Referees may prescribe the provision of an inferential recorder used in conjunction with a Boys Non-Recording Calorimeter.

The types of inferential recorders approved are:—

The Sigma B.ThU. Recorder No. 3.
The Simmance Calorgraph type R.

The record of inferential recorders will be used only for adjusting the average calorific value ascertained from the results of testings made with the Boys non-recording calorimeter, according to the fluctuations of calorific value which have been shown by the inferential recorder to have taken place in the intervals between testings.

In cases in which it appears necessary to the Gas Referees a Recording Calorimeter or an inferential recorder, can be prescribed for testing places for Gas Undertakers who have not sold more than 100 million cub. ft. of gas in the preceding year.

Tests for Calorific Value with Non-Recording Calorimeter.

These are carried out by observations taken by means of a calorimeter illustrated in fig. 1, in which water flowing is heated by a double burner consuming about $\frac{1}{15}$ cu. ft. per minute. The calorimeter having been set up, the burners are lighted and the water flow is so adjusted that the difference between he inlet and outlet temperatures is about 20° C. The heating of the calorimeter is allowed to proceed for not less than 45 mins. While the hand of the meter is making four revolutions, four readings of the inlet thermometer and fifteen readings of the outlet thermometer are taken at intervals, during which period the discharged water is collected in a measuring vessel. The temperature of the surrounding air, of the gas in the meter, and of the effluent gas are also observed. The mean of the four readings of the inlet temperature is subtracted from the mean of the fifteen readings of the outlet temperature, and the difference is multiplied by 11·976 and by the number of litres of water collected. The product is divided by the tabular number.* The difference in degrees Fahrenheit of the temperature of the effluent gas and the surrounding air is multiplied by $\frac{3}{5}$, and this is added to the result previously found if the effluent gas is the warmer of the two, or subtracted if the effluent gas is cooler. The result is the gross calorific power of the gas in British thermal units per cu. ft. (*c*).

In addition to the foregoing, the amount of condensed water resulting from the combustion of the gas is measured. For this purpose the condensation water is led into a measuring vessel not less than 20 mins. after the calorimeter has been set up. The amount collected in not less than 30 mins. is measured, and the time of collection accurately noted. The number of cubic centimetres collected is multiplied by the number of seconds taken for four revolutions of the meter and by the constants 1·86 and 3·968 (*a*). The number of seconds during which the water is collected is multiplied by the tabular number (*b*). The first product (*a*) divided by the second (*b*) and subtracted from the gross calorific power (*c*) gives the net calorific power in British thermal units per cubic foot.

The meters used for these tests are of the 'wet' type, having a calibrated capacity of $\frac{1}{12}$ cu. ft. per revolution of the drum. The drum carries an index hand moving over a dial divided

* The tabular number depends upon the barometer reading and the temperature of the gas passing through the meter.

into 100 divisions, and the meters are provided with a thermometer for measuring the gas temperature.

Testing of Non-saturated Gas.

Undertakers supplying non-saturated gas are entitled to a modification of the ordinary method. Before stating the modifications the Gas Referees require evidence of the degree of dryness of the gas over the distributing system. The degree of dryness is determined by a hygrometer of the Gas Light & Coke Company's pattern, made by Scientific Supplies or Negretti and Zambra, and certified by the Gas Referees.

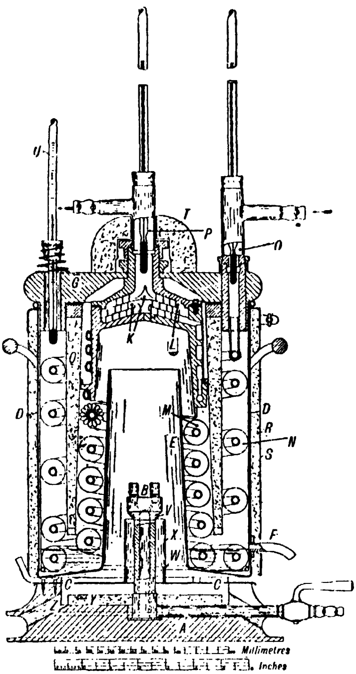


FIG. 1.

Test of Sulphuretted Hydrogen.

The gas is passed through a glass vessel in which are suspended six strips of bibulous paper moistened with a solution of 6·5 grams of acetate of lead crystals in 100 c.c. of water. The gas having been turned on full for three minutes, one of the strips is compared with a strip similarly moistened but not exposed to the gas. The gas is taken as exhibiting a trace of H_2S if the exposed is unmistakably darker than the unexposed strip.

Testing the Pressure of the Gas.

A test for pressure may be made at any time at any testing station and should be made on each occasion on which the Gas Examiner visits the testing place to make other testings, or to verify the readings of a recording calorimeter.

Tests are made with U gauges filled with water.

ACETYLENE.

(Contributed by the late Professor Vivian B. Lewes, F.C.S., F.I.C.)

Acetylene, which is a gaseous compound of hydrogen and carbon, represented by the formula C_2H_2, is a colourless gas with a density of 0·92. As ordinarily prepared, it has a strong and very penetrating odour, but when purified from minute traces of other gases, invariably present with it in its crude state, has a not unpleasant ethereal odour. It can be converted into a liquid at a pressure of 21·53 atmospheres, at a temperature of 0° C., and liquefies under ordinary atmospheric pressure when cooled to a temperature of − 82° C. The critical point of the gas is 37° C., and at this temperature the pressure has to be raised to 68 atmospheres for its liquefaction.

Acetylene is readily soluble in water, which at normal temperature and pressure takes up about its own volume of the gas; its solubility in other liquids is given in the following table:

100 Volumes of		Volumes of Acetylene.	100 Volumes of		Volumes of Acetylene.
Brine	absorb . .	5	Fusel oil	absorb . .	100
Water	,, . .	110	Benzene	,, . .	400
Alcohol	,, . .	600	Chloroform	,, . .	400
Paraffin	,, . .	150	Acetic acid	,, . .	600
Carbon disulphide	,, . .	100	Acetone	,, . .	2,500

Extended researches by Bistrow and Liebreich upon the toxic properties of acetylene show that it acts upon the blood in much the same way as carbon monoxide, but exhaustive experiments show that it is less poisonous than ordinary coal gas.

Acetylene forms compounds of an explosive character with copper and silver, and at first this was thought to be a great drawback to its general use; but experience has shown that these fears were unfounded, and that brass gas fittings, when gas-tight, can be used with perfect safety.

Acetylene, when heated, polymerises, like most hydrocarbons, i.e. it is partly or wholly split up, and the resulting gas is usually a mixture of acetylene itself and poorer gases, such as hydrogen, methane, etc., while tarry compounds and even solid carbon are formed, if the heating is carried far enough.

This is a matter of the utmost importance to acetylene users. For lighting purposes, it means that a poor light is given; for oxy-acetylene welders it means that the temperature at the blowpipe is reduced considerably, indeed in some cases it has proved impossible to make satisfactory welds, owing to the poor quality of the gas.

These troubles arise almost exclusively with water-to-carbide generators, which are often faultily designed, so that overheating is bound to occur.

Experiments lately carried out by the French Acetylene Office have shown that polymerisation starts in an ordinary commercial generator at about 130° C., *i.e.* at a much lower temperature than was originally believed to be the case. Unfortunately many generators sold in this country act by driving the water away from the carbide container, when a certain amount of gas has been generated, leaving the live carbide surrounded by a mass of moist lime sludge, which is again surrounded by a blanket of acetylene. Acetylene, however, is a far poorer conductor of heat than water. At the instance of the writer, the heat conductivity of acetylene was ascertained by the National Physical Laboratory, we believe for the first time. It was found to be 0·000043 c.g.s., or only about one thirty-second of that of water. If, therefore, decomposing carbide is surrounded by acetylene instead of by water, only about 3 per cent. of the heat is carried off in a given time, which could be carried off by water, and the writer has seen sludge become actually visibly red under such conditions.

It is therefore of the utmost importance that acetylene users should be careful to see that they select acetylene generators in which there is always an ample supply of water in constant contact with the carbide, or constantly surrounding the carbide containers. It may also be pointed out that the better the carbide is, the more quickly it is attacked by water and, in consequence, the greater the quantity of heat liberated in a given time by its decomposition.

A new type of acetylene generator which differs radically in its generating chamber from anything previously used, although of extremely simple construction, has been put on the market in this country. The system has been in considerable use for about two years, and has given complete satisfaction.

In contact with air acetylene inflames at a temperature of 480° C. and burns with a flame which, when properly spread out so as to obtain a large quantity of oxygen, emits more light than any other gaseous compound of carbon and hydrogen.

Gas.	Candle hours per 5 cub. ft.	Carcel hours per cub. metre.	Gas.	Candle hours per 5 cub. ft.	Carcel hours per cub. metre.
Methane .	5·2	3·5	Propane .	56·2	40·0
Paris gas .	13·2	9·6	Ethylene .	70·0	49·0
London gas .	16	11·5	Butylene .	123·0	86·0
Ethane . .	35·7	25·0	Acetylene .	240·0	168·0

Professor Clowes has shown that acetylene has a wider range of explosibility when mixed with air than any other combustible gas, the limits in percentages being as follows:

	Per cent.		Per cent.		Per cent.
Acetylene . .	3 to 82	Carbon monoxide	13 to 75	Methane . .	5 to 13
Hydrogen . .	5 to 72	Ethylene . .	4 to 22		

Acetylene is made commercially by acting upon a compound known as calcium carbide (CaC_2) with water, when acetylene gas is liberated and calcium hydrate (slaked lime, $Ca(HO)_2$) is formed.

Calcium carbide is made by fusing together very pure coke (or high-carbon anthracite coal) and very pure lime by the intense heat of the electric furnace. No other way of heating the mixture to the required temperature has yet been discovered, although many attempts have been made; for instance, by the use of the oxy-acetylene blowpipe.

Acetylene generators, in which the interaction of the calcium carbide and water to generate the gas takes place, may be either automatic or non-automatic. In the former class, devices are used to stop the generation of the gas automatically, by cutting off the supply of water or the supply of carbide, thus doing away with the need of a large store-holder for the gas, whilst in the non-automatic class the action is allowed to continue until all the carbide is decomposed, the gas being stored in a holder from which it is used as required.

In making acetylene, water may be brought in contact with the carbide either by adding it to the carbide, or by adding the carbide to water.

The overwhelming majority of acetylene generators in practically all countries are now constructed on the water-to-carbide system.

Of late years the automatic type of apparatus for the generation of acetylene has almost entirely superseded the non-automatic type, owing to the very large quantity of acetylene now being required for the oxy-acetylene welding process.

Experience has shown that automatic generators can be constructed with a high factor of efficiency, while the capital outlay per unit of gas generated is considerably less than in the case of non-automatic apparatus.

Owing also to the much smaller space occupied by the automatic type, it has also been found much more convenient for private use.

Until comparatively recent times, the pressure customary in acetylene generators was from 4 to 8 inches water-column, these limits having been found to give the most satisfactory results for lighting purposes. The advent of the oxy-acetylene welding blowpipe, however, soon caused experiments to be made with higher pressures, with the most favourable results. To-day the general tendency in most countries is to have the acetylene under slightly higher pressure than the oxygen in the welding blowpipe, it being claimed that less oxidation of the metal took place under these conditions and that there was less chance of the right proportion of the two gases being interfered with by accidental obstruction in the blowpipe, and, lastly, there was less tendency for the flame to flash back.

In consequence of this movement, the Governments of the United States, Canada, Switzerland, and Denmark now allow a maximum pressure of 1 atm. in acetylene generators for welding, and Germany is on the point of adopting this maximum pressure also. Very interesting details of tests have been published by the Government Bureau of Standards in the United States.

The oxy-acetylene blowpipe has been most successfully used for a considerable time for case-hardening various parts of machinery, and in particular the gear-wheels and other parts of motor cars. Lately a most important development has taken place in this direction, the blowpipe having come into the most extensive use, not only for cutting and repairing rails, but for applying an exceedingly hard surface, to a depth reaching even the high figure of ½ inch and more, to the entire rail system of electric tramways.

The improvement shown in the wearing properties of rails is shown by the following figures

	Wear of rails in 12 months. Untreated.	Treated.	No. of cars carried.
Bristol	0·045	0·0157	300,000
Liverpool	0·030	0·0125	200,000

The rails are hardened *in situ*, without otherwise interfering with the track in any way.

When acetylene first made its appearance, its use was confined almost entirely to lighting detached buildings, motor-cars, and cycles.

The advent of electric sets materially reduced the field for acetylene for lighting purposes, but recently there has been a decided revival in this direction, owing to the enormously increased construction of houses of the bungalow type, large numbers of which are situated so far from a centre of supply of coal-gas or electric current that the cost of laying the necessary supply pipes or cables makes the capital outlay too great, in view of the modest requirements per bungalow.

In such cases the very low capital outlay required for an acetylene outfit, its extreme simplicity, and the beautiful light it gives, have made this form of lighting extremely popular.

The use of the oxy-acetylene blowpipe has continued to increase during the past year; indeed these and other applications have so greatly extended the use of acetylene that whereas the consumption of carbide before the war amounted to about 28,000 tons, it rose in 1925 to 40,000 tons per annum. There was a set-back, owing to the difficulties in the coal industry, but at the time of writing a consumption of approximately 40,000 tons a year has again been reached.

The use of acetylene in other directions than as a combustible again made great strides in the past year. One of the most interesting new derivatives is the 'Meta' block, which has been placed on the market by a Swiss firm. This block resembles in appearance solidified methylated spirit, but in its properties it is greatly superior, inasmuch as the block does not liquefy when burning, nor evaporate even if exposed to the air for many months.

Acetylene as obtained from ordinary generators contains a certain amount of impurities. The use of the acetylene blowpipe for welding high-class steels has shown more and more clearly the necessity for removing these impurities from the gas, a purification which can easily be effected by the use of such purifying materials as catalysol, heratol, etc.

One pound of commercial calcium carbide, when acted upon by water, yields on an average about 4·5 cub. ft. of the gas.

Before the war the price of carbide was about £12 10*s*. per ton, packages included. The price is now about £16 to £18 in the United Kingdom, while it is still higher in many of the countries producing it.

Dissolved Acetylene.

By an Order in Council issued in 1897 the compression of free acetylene was made illegal, but further orders were issued in 1901 and 1912 exempting the process now known as 'dissolved acetylene.' This is acetylene thoroughly purified and washed, and compressed into steel cylinders previously filled with a porous mass and having a fixed quantity of acetone added. Acetone is a liquid hydrocarbon, having the property of absorbing 25 times its own volume of acetylene at one atmosphere and 15° C.

For each additional atmosphere of pressure, the same proportion of gas is absorbed, but an allowance has, of course, to be made not only for the volume occupied by the porous filling of the cylinder, but, in addition, in order to allow for expansion, so that it is customary to charge these cylinders with only 43 per cent. of acetone.

A cylinder will therefore in actual practice contain about 100 times its own volume at 10 atmospheres pressure and about 150 times at 15 atmospheres pressure.

In order to show the enormous capacity for storing light obtained in this manner, it may be mentioned that the standard oil-gas cylinder in use on railways has a capacity of 14 cub. ft. at atmospheric pressure. As the oil-gas is usually compressed to 12 atmospheres, this gives a gas volume of 168 cub. ft., which, if filled with oil-gas, will supply eight burners using 2 ft. per hour and burning eight hours per day, for about a day and a half. If this cylinder were filled with porous material and acetone in the manner described above, and charged with acetylene at 15 atmospheres, its capacity would be 2,100 cub. ft., or sufficient to supply eight acetylene burners of 1 cub. ft. each (giving a most brilliant light), for about thirty-three days (basing on eight hours' lighting per day), before the cylinders would require to be re-charged.

The possibility of obtaining acetylene in compressed form opens up a very wide field for the expansion of its utility.

The tests carried out by the Home Office Authorities before issuing the exemption order of 1901 were exhaustive, and furnished satisfactory proof of the safety attending the process of manufacture and the use of the cylinders.

Storage of Dissolved Acetylene.

Acetylene dissolved in acetone and stored, under a pressure not exceeding 225 lbs. per sq. in., in a vessel filled as completely as possible with some porous substance is not deemed an explosive, and can therefore be manufactured, stored, and used without risk of infringing the Explosives Act.

(See also 1919 Order in Council, 'Dissolved Acetylene.')

Porous Material.—Various types of filling are in use and appear to give satisfaction. The porosity of the material should not exceed 80 per cent. The material should completely fill the receptacle, and should contain no pockets or cavities.

The Solvent.—Acetone is the most suitable commercial solvent, but substitutes for acetone could be permitted. The solvent should not completely fill the porosity of the material under any rise of temperature likely to be met with in practice.

Pressure Allowed.—The pressure can be 225 lbs., provided the cylinders used are of the solid drawn type, the pressure to be measured at 60° F.

The Cylinders.—The cylinders should be made of mild steel containing not more than 0·25 per cent. of carbon and not more than 0·05 per cent. each of phosphorus and sulphur. The test pressure should be four times the working pressure, and should be maintained for not less than fifteen minutes. All cylinders should be annealed, but the use of fusible plugs, at present compulsory on acetylene cylinders, should be discontinued, as they are apt to become loose and, further, in the event of fire occurring, may actually lead to the fire being augmented.

It is not considered necessary to impose any conditions as regards the retesting of acetylene cylinders beyond requiring them to be thoroughly examined visually at intervals. The form of retesting to which oxygen and other cylinders are subjected cannot readily be applied to acetylene cylinders with their content of porous material and solvent.

Acetylene Lighting.

For the illumination of buildings, towns, and large areas by acetylene, generators of some suitable type are in general use.

Dissolved acetylene is used successfully from an economical and efficient point of view for many purposes where a constant supply of acetylene is required, and is particularly adaptable where space available for the apparatus is strictly limited.

In the matter of motor vehicle lighting it has been very extensively adopted. A cylinder containing acetylene absorbed in acetone under pressure and of convenient size is attached to the footboard or other suitable position on the car. Such a cylinder contains a sufficient supply of acetylene to meet the requirements for the lighting of the necessary lamps. The time for which the supply is available is variable according to the rate of consumption; the pressure of gas in the cylinder is reduced by an automatic governor or regulating valve, and the supply in the service pipes is maintained at a constant pressure to the lamps.

The standard automobile cylinders contain 20 cub. ft. and 40 cub. ft. of dissolved acetylene, and by the exchange system originated by the manufacturers it is possible for users to obtain refils at practically all important garages throughout the country.

Larger vehicles, such as motor omnibuses, mail vans, motor lorries, etc., may be equipped with outfits of larger capacity to meet the requirements of any particular case.

Another recent departure is the manufacture and circulation of cylinders containing dissolved acetylene of such capacity to meet the demands of motor cyclists. These cylinders are standardised and contain 5 and 10 cub. ft. of acetylene, and, as with the standard cylinders for automobile work, they may be exchanged throughout the country.

For buoy and coast lighting dissolved acetylene has many advantages. The purity of the gas in this form, which is assured before compressing, gives the light increased brilliance and greater penetrating power, hence a shorter light period can be employed than with any other illuminant. Various special apparatus are in use, in which flashes of long or short duration are obtained by automatic means, the light being extinguished between the flashes instead of obscured (a small bypass burner remaining alight), thus economising gas. In some cases a flash of as short duration as one-tenth of a second is obtained, with perfect accuracy and flame. By means of this economical consumption and the great storage capacity of a comparatively small cylinder, a gas supply for periods as long as 12 months can be easily carried on a buoy, no attention being required until the gas is exhausted, when the cylinder is disconnected and replaced by a full one. The cylinders may be arranged in the form of a battery where larger consumption is involved.

Owing to the smallness of space required for acetylene in this form it is particularly adaptable for lighting railway coaches, yachts, and small craft. The effect of atmospheric temperature on the gas in this form is almost negligible, and other advantages are obtained by reason of the purity of gas and the elimination of handling carbide, water, waste and residue, and by the fact that the gas in this form can be stored indefinitely without deterioration.

The applications of this form of acetylene are ever increasing, and it has been recently applied with success to cinematographs, searchlight, signalling and projection work, where a small and compact plant consistent with efficiency is required.

The most recent application for illuminating purposes is that of its adoption for portable hand-lamps for mining and other similar purposes.

Lately acetylene buoys have commenced to displace other forms of coast-lighting to a very considerable extent. Such buoys take a very large charge of calcium carbide, usually one to two tons. They are entirely automatic, require practically no attention, and as they run for six to twelve months before requiring recharging, they form an exceedingly economical means of coast and harbour-lighting. Another great advantage is the fact that the light can be recognised at a much greater distance than that of the oil, or oil-gas, buoys previously used.

PETROL AIR GAS.

(*Contributed by E. Scott-Snell, A.M.S.E.*)

Petrol gas is made by impregnating air with the vapour of petroleum spirit, this mixture being sent through pipes in the usual way for combustion with incandescent mantles. The essential features of an apparatus for producing this gas are an air compressor, a carburettor—in which the air and petrol are brought into contact—and a gas-holder from which the pipe service is fed. In automatic apparatus, *i.e.* machines in which gas is made according to the demand, the holder is also used as a governor for controlling the supply of air and petrol to the carburettor. The relative proportions of air to petrol are very important: in fact, upon the relation of these ingredients, and the constancy of this relation under varying load and climatic conditions, the success or failure of any apparatus must depend. As a mantle is used for light-giving, a flame is required at the burner which shall give the highest possible temperature, *i.e.* the amount of air mixed with the petrol vapour must be such as to completely combust it. With the more volatile spirit, mainly hexane (C_6H_{14}), used for making this gas, complete combustion is assured when approximately two volumes of vapour are mixed with 98 volumes of air. This is calculated from the typical equation $2C_6H_{14} + 19O_2 = 12CO_2 + 14H_2O$ representing the combustion of the chief constituent of light spirit. At present, opinion seems to be divided as to the advisability of sending such a mixture—which from its composition is obviously an explosive one in a confined space

such as a gas-holder—through the system. It is found, however, that when the percentage of vapour in the mixture exceeds 4½ per cent., *e.g.*, a 5 per cent. or 6 per cent. mixture, the gas is incombustible without the addition of further air, and in the better class machines advantage is taken of this fact to make a slightly richer gas than 4½ per cent., diluting it with the extra air required for complete combustion by the aid of Bunsen burners.

The air-compressing apparatus of a petrol gas machine may be actuated by any motive power, but as the work of forcing the gas through the pipes is comparatively light, nothing more powerful than a hot-air motor or wound-up weights is usually required. In a good apparatus, the relative amounts of air and petrol are mechanically measured before passing to the carburettor, so that constancy of mixture is obtained under all conditions of temperature within wide limits. When other means are adopted, such as leading the air across a bath of petrol, selective evaporation takes place, producing erratic results, the make of gas varying with every change of temperature. In fact, attempts to make apparatus of this sort automatic almost inevitably end in failure.

Petrol gas is probably one of the safest lighting mediums known, by reason of the small compass over which mixtures of petrol vapour and air are explosive (1½ per cent. to 4½ per cent., c.f. acetylene 3·3 per cent. to 52·3 per cent., coal-gas 7·9 per cent. to 19·1 per cent.) and the fact that over 90 per cent. of the escaping gas from any leak is air. It is also non-asphyxiating.

The gas varies in calorific value according to the proportions of the mixture and the spirit used, but a 6 per cent. mixture from 0·680 sp. gr. petrol has a gross calorific value of nearly 300 B.Th.U. per cubic foot. Although this is a comparatively low figure, cooking and heating can be efficiently performed with suitably proportioned burners.

One gallon of light spirit gives 5,000 candle-power hours (Lewes). With petrol at 1*s*. a gallon this is equivalent to 2·4*d*. per 1,000 candle-power hours. The Government tax of 3*d*. per gallon is remitted when petrol is used for purposes other than propelling road vehicles.

Petrol gas can be stored almost indefinitely over water, but since a small 'Bijou' mantle requires anything from 4 to 12 cub. ft. per hour (according to whether a 6 per cent. or 2 per cent. mixture is used) the storage of any reasonable quantity of the gas entails the use of very large holders. Hence the advantage of the weight-driven type of automatic generator.

Attempts are being made to effect economy by using the less volatile petrols of 0·720 to 0·760 sp. gr. for lighting purposes, such as are in general use for motor cars. The physical characteristics of these spirits, however, are such that they do not lend themselves readily to cold carburation, for either considerable quantities will pass through the carburettor unvaporised, or if heat is used to evaporate them, heavy condensation is unavoidable in the pipe service, which must necessarily be very carefully laid to provide for it.

An explosive (or 'weak') mixture in the pipes and holder is unavoidable if heavy spirits are used, since they are not sufficiently volatile to allow of the use of a 'rich' mixture. See also Scott-Snell on 'Petrol Air Gas,' Soc. of Eng. Journal and Trans., March 1911; Illuminating Engineers' Soc. Journal and Trans. February 1913.

Heating Value and Candle-Power of Petrol-Air Gas.

According to the Snercold Engineering Co., as the result of tests of a variety of different brands and grades of petrol, the fair average heating value of a gallon of petrol is 143,500 B.Th.U., and the maximum candle-power hours that can be obtained under the best conditions is 6,025. The following table shows different mixtures in percentages of petrol vapour used; for calculation purposes the heating value has been taken as 143,000 B.Th.U., and the candle-power hours as 6,000 per gallon.

TABLE SHOWING VARIOUS MIXTURES OF PETROL-AIR GAS.

Mixture Percentage of Petrol Vapour.	Gallons per 1,000 Cubic Feet of Gas.	Heating Value B.Th.U. per Cubic Foot.	Cubic Feet of Gas to the Gallon.	Candle-power Hours per Cubic Foot.
6	2·184	312	458·3	13·104
5½	2·0	286	500	12·0
5	1·82	260	555	10·92
4½	1·638	234	612	9·828
4	1·4638	208	687·5	8·7828
3½	1·274	182	785·7	7·644
3	1·092	158	916·6	6·552
2½	0·91	130	1100	5·46
2	0·728	104	1375	4·368
1½	0·546	78	1833·2	3·276

The heating value of coal-gas varies from 420 to 550 B.Th.U. per cubic foot according to the source of supply.

HEAT TREATMENT OF GAS CYLINDERS.

(*Second Report.**)

In view of experiments made by the Gas Cylinders Research Committee the Committee agree that the periodical re-heat treatment of carbon steel gas cylinders which have not been obviously damaged, serves no useful purpose, and they recommend that the practice be discontinued.

* 'Periodical Heat Treatment of Gas Cylinders.' Published for the Department of Scientific and Industrial Research by H.M. Stationery Office. Price 2s. 6d. net.

SECTION XXXIV

PART I

HEATING BY HOT WATER, STEAM, GAS, &c.—DRYING

(pp. 659–674)

(Revised by L. Copeland Watts, M.I.Mech E., P.P.I.H.V.E., A.C.G.I.)

PART II

VENTILATION (pp. 675-683)

(Revised by L. Copeland Watts, M.I.Mech.E., P.P.I.H.V.E., A.C.G.I.)

PART III

LIGHTING (pp. 685–708)

(Revised by the Lighting Service Bureau.)

PART IV

FIRE EXTINCTION AND PREVENTION

(pp. 709-716)

(Contributed by Hubert B. Graham, M.I.Mech.E.)

SECTION XXXIV

PART I

HEATING.

(Revised by L. Copeland Watts, M.I.Mech.E., P.P.I.H.V.E., A.C.G.I.)

AMOUNT OF HEAT REQUIRED.

Heat Losses.

When any room or building is maintained at a higher temperature than the outside air there is a loss of heat from the structure through walls, windows, etc., and also a loss due to air change. Modern practice demands that these losses shall be estimated as accurately as possible. A few of the more usual factors for materials in common use are tabulated below, the figures being the loss in B.Th.U.'s per hour per sq. ft. per deg. F. difference between inside and outside temperature.

Material.	Loss in B.Th.U.'s.
Walls.	
Brick wall 4½ ins. thick with plaster inside	0·57
Brick wall 9 ins. thick with plaster inside	0·43
Brick wall 13½ ins. thick with plaster inside	0·35
Brick wall 18 ins. thick with plaster inside	0·29
Brick cavity wall 11 ins. thick, 2 ins. air space	0·34
Brick cavity wall 15½ ins. thick, 2 ins. air space	0·29
Concrete wall 4 ins. thick	0·64
Concrete wall 8 ins. thick	0·47
Brick wall 9 ins. thick with 4 ins. stone facing	0·40
Windows.	
Single window	1·0
Double window	0·5
Floors.	
6 ins. concrete on earth	0·20
Concrete on earth with wood blocks	0·15
1 in. wood floor with air space under	0·40
Ceilings and Roofs.	
Plaster ceiling with air space above (roof boarded and felted with tiles or slates)	0·30
Concrete 6 ins. thick, finished asphalt	0·57

It is usual to allow for the air contents of a room (natural ventilation) to be changed from 1 to 3 times per hour. A usual figure for office blocks, etc., would be 1½ to 2, while for hospitals, etc., at least 3 changes should be allowed.

The heat to be allowed for air change can be ascertained by multiplying the total quantity of air to be heated per hour by 0·019 for each deg. F. of the required rise.

'The Computation of Heat Requirements for Buildings,' published by the Institution of Heating and Ventilating Engineers, gives data for many types of building materials together with recommended allowances for air change and suitable room temperatures for many classes of buildings.

The total heat requirements for heating a room or building are the sum of the heat required to warm the incoming air plus the heat required to replace that lost through walls, windows, floors and ceilings, found by multiplying the various surface areas by the coefficients given in the above table and by the difference between the internal and external air temperatures.

It is usual practice to make allowances on the heat losses estimated by means of the above co-efficients according to the exposure of the building or room. This varies according to the aspect and may increase the total emission through the surfaces by as much as 15 per cent. for brick walls, 30 per cent. for glass, and 20 per cent. for flat roofs. Similarly in sheltered positions the emission is reduced. For buildings in towns and cities and in positions not particularly sheltered or exposed, the co-efficients given will be found suitable. Again, however, allowance in the amount of heating surface and the output of the boiler should be made according to the way the heating installation will be operated. In the case of buildings with very intermittent use such as churches or public halls which are only occupied one or two days a week, the heating not being maintained when the building is unoccupied, it is usual to increase the heating surface by approximately 60 per cent., the boiler power being increased by 100 per cent. For office buildings where the boiler is banked at night, it is desirable to increase the heating surface by approximately 10 per cent. and the boiler power by approximately 50 per cent.

The effect of the wall finishings on the heating-up period of a room, that is, the time taken to reach comfort conditions after a period of non-occupancy, needs more consideration than it usually receives. Experiments carried out by Mr. A. F. Dufton* demonstrated that particular rooms were warmer after half an hour's heating when the walls were panelled with wood than they were after an hour and a quarter's heating when the walls were only plastered. It follows that in cases of intermittent heating, as for example an office block where the period of occupation is 8 or 9 hours a day, it is an advantage if the walls are finished with some material of small thermal capacity, of which wood is a good example.

Heating Surface.

The most common form of heating surface is the ordinary cast iron radiator. Though this name is used, it is, of course, well known that the heat emitted by a radiator is both by radiation and convection; the usual proportion for this being 30 per cent. radiation and 70 per cent. convection. The average figures for total emission from radiators vary between 150–185 B.Th.U.'s per hour per sq. ft. of surface with a difference of temperature between the room and the mean radiator flow and return of 100° F. The following list gives the approximate emission for the different types of cast iron radiators available at the present time, small variations occurring between the different makes:

Two column .	185	B.Th.U.'s per hour per sq. ft. of heating surface for 100° F. diff.
Four column .	170	,, ,, ,, ,, ,, ,, ,, ,, ,,
Six column . .	160	,, ,, ,, ,, ,, ,, ,, ,, ,,
Wall . . .	171	,, ,, ,, ,, ,, ,, ,, ,, ,,
Hospital (Narrow)	185	,, ,, ,, ,, ,, ,, ,, ,, ,,
Hospital (Wide) .	151	,, ,, ,, ,, ,, ,, ,, ,, ,,

If radiators are placed in any kind of enclosure, or fixed behind grilles, an allowance of from 5 to 30 per cent. extra surface should be made to counteract the loss of efficiency of the surface —very often the greater part of the radiant emission is lost and the circulation of air over the hot surface much impeded.

The heat emission from radiators depends upon the difference between the mean temperature of the water or steam in the radiator and the surrounding air temperature. To find the approximate heat emission from radiators for various water and air temperatures, the above emissions may be multiplied by the factors given in the following table.

TABLE OF FACTORS FOR MULTIPLYING STANDARD RADIATOR AND PIPE EMISSIONS FOR VARIOUS AIR AND RADIATOR TEMPERATURES.

Mean Radiator Temperature. °F.	Inside Air Temperature. °F. 45	50	55	60	65	70	75
140	0·93	0·87	0·81	0·75	0·69	0·63	0·57
150	1·07	1·00	0·93	0·87	0·81	0·75	0·69
160	1·21	1·13	1·07	1·00	0·93	0·87	0·81
170	1·35	1·27	1·21	1·13	1·07	1·00	0·93
180	1·48	1·41	1·35	1·27	1·21	1·13	1·07
190	1·63	1·56	1·48	1·41	1·35	1·27	1·21

* *Proc. Inst. Heat. and Vent. Eng.*, vol. xxx, p. 132.

The amount of heat to be provided in each room by the radiators may be found by subtracting the heat emission of the circulating pipes from the total heat losses. The heat emission in B.Th.U.'s per hour per lineal foot of horizontal pipe, freely exposed, with a temperature difference of 100° F. between the pipe and air temperature is given in the following table :—

Nominal Bore of Pipe. Ins.	Emission B.Th.U.'s per Lineal Ft. at 100° F. Temperature Difference.
½	56
¾	68
1	82
1¼	101
1½	113
2	138
2½	163
3	191
4	243
5	291
6	337

To obtain the approximate emission from pipes for various pipe and air temperatures the emissions given above may be multiplied by the factors given in the previous table. The emissions from vertical pipes freely exposed are approximately 95 per cent. of the above figures and if pipes are near walls or ceilings the emission is about 80 per cent. of a freely exposed pipe. For horizontal pipe coils where the pipes are placed one above the other the emission for each pipe will be reduced by 5 per cent. for two pipes, 15 per cent. for four pipes and 25 per cent. for six pipes.

The use of low temperature radiation is becoming much more popular in modern heating installations. The heating surface employed when this type of system is installed consists in the main of two alternatives : (*a*) superimposed panels, and (*b*) embedded panels. Superimposed panels may consist of cast iron sections nippled together, or may consist of steel plates provided with a system of waterways on the back. In the case of embedded panels, pipe coils consisting of ½ in. or ¾ in. steel or copper tube at 6-in. centres are embedded in the face of the structure behind the plaster. These panels may be placed anywhere in the room, but the most convenient position is in the ceiling. The emission in the case of superimposed panels fixed to the ceiling is approximately 80 per cent. by radiation and 20 per cent. by convection, while in the case of the embedded panels the radiation component may be even higher.

With embedded panels these are usually laid on the shuttering during construction of the building, the concrete being poured over the pipes and fixing them in position. To ensure a proper key for the plaster, slip tiles should be provided between the pipe coils or some other provision made. Special precautions must also be taken to ensure that the plaster employed is a first-class specification, the final coat being reinforced by means of a layer of scrim worked in with a trowel over the whole of the surface below the panel and extending 6 in. to 12 in. on each side of the embedded coil. Firms who specialise in this form of heating will provide satisfactory specifications for the plaster work. It is also usual to limit the temperature of the water in the system to a maximum of 130° F. to reduce the risk of cracking due to the expansion of the pipes. When panels are embedded in the ceiling under a flat roof, insulation consisting of not less than 2 in. of cork should be provided over the whole of the roof.

When designing a heating installation for a building in which air change is expected to be relatively large, as for instance open air schools, sanatoria, hospitals, etc., it is desirable to arrange that the heat shall be provided mainly in the form of radiant heat. To meet this condition, particularly for Open Air schools, floor heating has been developed in which the surface temperature of the floor is maintained at not more than 70° F.

It is fairly common to find radiators finished with aluminium paint, or decorated with one of the bronze powder finishes, the result being that the radiation is appreciably diminished. This is shown by the following table which is the result of experiments carried out by the Department of Scientific and Industrial Research at the Watford Research Station, and set out in their Food Investigation Board Special Report, No. 9 (revised edition, 1931):

'Taking emissivity of dead black as unity, and expressing the values for other materials as fractions of this, we have—

'Dead Black	1·0
Black paint	1·0
Black enamel	1·0
Dark Green paint	1·0
White paint or enamel	1·0
Aluminium paint (one coat)	0·7
Aluminium paint (two coats)	0·5 to 0·7
Aluminium paint (rubbed with fine sandpaper)	0·8
Dull copper surface	0·15.'

The above figures are, of course, only applicable to the radiant heat given out by a radiator, the effect of different finishes on the heat given out by convection being practically nil.

The actual amount by which the total emission (by radiation and convection) is affected will in any particular case depend on the type of surface, and be much more serious where a large part of the emission is in the form of radiation.

In the case of radiators the use of aluminium or bronze paint may reduce the total emission by up to approximately 12 per cent., and in the case of superimposed panels on the wall this reduction may be as much as 25 per cent., and in the case of superimposed panels on the ceiling as much as 36 per cent.

While the amount of heating surface required should always be calculated from a careful estimate of the heat losses from the building, the following figures giving the amount of heating surface in the form of hot-water radiators and exposed pipes required for various types of building per 1,000 cub. ft. contents may be taken as a rough guide :—

Type of Building.	Surface required. Sq. ft.
Hospitals	20–35
Offices	15–20
Public buildings	12–15
Churches	12–15
Theatres	5–10
Houses	10–15
Shops	7–10

For horticultural work the following figures may be taken as representative :

Type of House.	Approximate Surface in ft. super per 1,000 cub. ft.
Houses requiring to be kept at a temperature above 35° F. in order to prevent frost entering	16
Peach and similar houses	27
Greenhouses	40
Tomatoes, vines, etc.	60
Forcing pits	130

Effective Temperature and Equivalent Temperature.

During the last few years, the effect of radiant heat on comfort conditions has received much consideration. It is generally accepted that the readings of an ordinary thermometer do not always give a true indication of the actual feeling of comfort in a room. With a proportion of radiant heat, comfortable conditions seem to be obtainable with a lower air temperature than would be necessary if no radiant heat were available.

The American term 'effective temperature' has been brought into use. This is derived from measurements of (1) dry bulb temperature ; (2) wet bulb temperature ; and (3) air velocity. The figures corresponding to different values of the above variables are set out on psychrometric charts from which the effective temperature for any set of conditions can be ascertained. These charts, while no doubt being useful in America for determining the indoor comfort conditions in summer, do not take into account the effects of radiation.

The Royal Naval Personnel Research Committee of the Medical Research Council has recently adopted a scale of measurements called 'Corrected Effective Temperatures.' With this scale a correction is made to the Effective Temperature Chart by using the readings of the Globe thermometer instead of the ordinary thermometer. The globe thermometer which was devised by Dr. H. M. Vernon is a 6-in. diameter copper globe painted matt black and containing an ordinary thermometer with its bulb at the centre of the sphere. The temperature of the instrument is mainly influenced by radiant heat and air temperature and to some degree by air movement. The publication by H.M. Stationery Office entitled 'Environmental Warmth and its Measurement,' by Dr. T. Bedford, explains the use of The Effective Temperature Scale under normal conditions, and conditions of high temperatures and humidities.

Another measurement of comfort called equivalent temperature is commonly used in England. (The term for this was originally the same as the American, but has been altered to avoid confusion.) The equivalent temperature of an environment is defined as 'that temperature of a uniform enclosure in which, in still air, a black body of sufficient size would lose heat at the same rate as in the environment, the surface temperature of the body being one-third of the way between the temperature of the enclosure and 100° F.'

The Building Research Station at Watford has developed an instrument called the Eupatheoscope, for recording equivalent temperature. This instrument consists of a black-painted, hollow copper cylinder, whose surface is maintained at a temperature in accordance with the above definition. The heat input required to maintain the surface of the instrument at this temperature in any room gives a measure of the equivalent temperature of the room.

Whilst the Eupatheoscope admirably serves the purpose for which it was designed, namely, the examination of ordinary conditions of artificial heating, it takes no account of the effects of such factors as humidity at high temperature, etc.

Contract Temperatures.

Many attempts have been made to develop formulæ to predict the internal building temperature when the heating installation is run at its designed temperature but the external temperature is different from the design temperature. These should be used with extreme caution since none take into account the effect of rising or falling external temperature or the thermal capacity of the building. Further, the effect of air change being different from that estimated has a very considerable effect on the final result.

Boiler Power.

It is good practice to allow for a listed boiler power from 25 per cent. to 33⅓ per cent. more than actually indicated by the heat emission from the heating system. This provides a margin for quick heating up from cold when necessary, and also tends to fuel economy, a boiler worked habitually at or above its rated capacity being generally much less efficient than when it has a good margin. In cases where the heating is intermittent the margin should be greater as previously indicated.

The boiler heating surface, as listed by the makers of boilers, is of little use in determining the output unless the proportion of primary and secondary surface is also known. In general it is better to rely on the makers listed output in B.Th.U./hr. Boilers of ample size for their duty give higher efficiencies, due to the lower temperature of the flue gases passing to the chimney.

Efficiency of Heating Boilers.

As regards efficiency, a boiler of moderate size may well give 70 per cent. or even more under test conditions at makers' works, but the overall efficiency under actual working conditions would hardly equal this figure, except for very large plants, a more usual figure being 50 per cent. or 55 per cent., and many boilers, with unskilled attention and long firing intervals, undoubtedly do not maintain 30 per cent. efficiency.

Fuels and their Relative Costs.

The above remarks regarding efficiency refer generally to solid fuel hand fired, the fuel most commonly used being coke. Anthracite is also a most suitable fuel to use in central heating boilers, although it is not often that this can be obtained at a price economical enough to compare favourably with coke. In recent years small graded bituminous coal has been used to a considerable extent for firing boilers by means of automatic stokers. These consist essentially of a hopper from which the fuel is fed by means of a worm into a retort built into the base of the boiler, air provided by a fan being introduced through slots in the sides of the retort. These stokers are thermostatically controlled, the rate of burning being adjustable to suit the requirements so that steady boiler temperatures can be maintained within fairly close limits.

For central heating and hot water supply installations the use of oil as a fuel has many advantages in regard to cleanliness, storage space and labour for operating and many large and small solid fuel plants have been converted to burn fuel oil. Small installations usually comprise semi- or fully-automatic plant, with thermostatic control, burning gas oil or fuel oil (200 seconds Redwood No. 1 at 100° F.). For large plants where maintenance labour is available the cheaper heavy oil having a viscosity of 960 seconds is generally burned with semi- or non-automatic equipment.

Town gas is also a very useful fuel, which lends itself to automatic control, and while largely used for small and medium sized installations of hot water supply, and in a lesser degree for central heating, it is generally found that for large central heating installations its cost is prohibitive, except when it can be obtained at, say, 6*d*. per therm (100,000 B.Th.U.'s), equivalent to 2*s*. 6*d*. per 1,000 cub. ft., when the matter requires careful consideration, as it must be remembered that with gas, as with oil firing, a certain proportion of the labour costs can be saved.

Electricity, owing to its higher prime cost, when compared with most other fuels, is chiefly of use in small or medium installations where running costs are not the first consideration. There are two general methods of utilizing this source of heat: (1) by thermal storage, and (2) by installing direct heaters—either tubular heaters or low temperature panels, or radiants—at the points at which the heat is actually required. When a thermal storage system is employed, current can usually be obtained at special rates during the night time, or other off-peak periods, and used to raise the temperature of water contained in large storage cylinders. During the period the heat is required this water is circulated as in an ordinary heating system, the required flow temperature being maintained by special automatic mixing valves. The price of current must, however, be exceedingly low, say in the neighbourhood of 0·5*d*. per unit, before this method can be considered on a cost basis, even when reduced labour charges, etc., are taken into consideration.

Electric heaters, either as electric fires, tubular heaters, or high or low temperature panels, are particularly useful in cases where the heat is required for short, intermittent periods only, as the heat is available almost as soon as the current is switched on, and can be turned off when no longer required.

In estimating the relative cost of various fuels, other factors besides the price paid have to be borne in mind, the chief of which are the calorific value of the fuel, the efficiency of the boiler plant and the labour costs involved.

A third method of using electricity for space heating has been developed in recent years. This system which uses a refrigerator for heating is known as the Heat Pump and may be installed where there is a source of low grade waste heat such as river water; provision being incorporated in the design to utilise the heat given up by the condenser and to discard the substance from which heat is extracted in the evaporator. When electricity is used to drive the compressor of the refrigerator the heat value of the energy used is only a fraction of that required to raise the temperature of the refrigerant, the rest of the heat being obtained from the substance surrounding the evaporator coils. In practice the heat provided by the condenser is some three or four times the heat value of the electricity used depending upon the relative temperatures and the efficiency of the compressor.

Insufficient information is available to prove that the installation of a Heat Pump is an economical proposition in all cases, but where refrigerating apparatus is installed for other purposes such as air conditioning it may be used economically for heating in winter providing a supply of waste heat is available.

Table I gives a list of the usual fuels, their probable calorific value, and the B.Th.U.'s obtainable from each for 1*d*. in round figures at the prices shown, which it will be appreciated may vary very widely at the present time and are based on convenient round figures so that an estimate of the probable cost of fuel can be made readily by adjusting the figures in accordance with the prices ruling at the time the estimate is made.

TABLE I.

Fuel.	Basic Price.	Calorific Value.	Gross B.Th.U.'s per 1*d*. at 100 Per Cent. Efficiency.	Approx. Cost per Therm. (100,000 B.Th.U.'s.)
Coke . . .	80/– per ton	11,500 B.Th.U.'s per lb.	26,800	3·72
Coal . . .	80/– „ „	11,500 „ „ „	26,800	3·72
Anthracite . .	100/– „ „	14,000 „ „ „	26,160	3·83
Fuel Oil . . .	£10 „ „	19,000 „ „ „	17,750	5·64
Town's gas . .	10*d*. per therm	—	10,000	10·0
Electricity (off-peak)	0·5*d*. per unit	3,415 B.Th.U.'s per unit	6,825	14·66

Table II gives the usual boiler efficiencies for the different classes of fuel and makes allowance for the probable labour costs, the final column gives the cost per 100,000 B.Th.U.'s taking these two factors into consideration. The remarks given above regarding fuel costs ruling at the time the estimates are made, apply also to this table.

TABLE II.

Fuel	Size of Installation.	Cost per Therm from Table I (see note above).	Running Boiler Efficiency. (Per cent.)	Equivalent Cost per Therm.	Cost per Therm including Labour. (Approx.)
Coke	Small	3·72*d.*	50	7·44*d.*	10·50*d.*
Coke	Large	3·72*d.*	60	6·20*d.*	7·50*d.*
Coal	Small	3·72*d.*	40	9·30*d.*	11·80*d.*
Coal	Large	3·72*d.*	60	6·20*d.*	7·50*d.*
Anthracite . .	Small	3·83*d.*	55	6·97*d.*	9·61*d.*
Anthracite . .	Large	3·83*d.*	60	6·38*d.*	7·72*d.*
Fuel Oil . . .	Small	5·61*d.*	50	11·28*d.*	12·00*d.*
Fuel Oil . . .	Large	5·64*d.*	65	8·68*d.*	9·20*d.*
Town Gas . .	—	10·0*d.*	80	12·50*d.*	13·00*d.*
Electricity . .	—	14·66*d.*	95	15·44*d.*	15·54*d.*

The above figures are representative only and each case must be calculated independently, very careful consideration being given to the probable boiler efficiencies, labour involved, etc. It should also be noted that the basic price of fuel in Table I can vary very considerably according to locality and conditions of contract. The figure given in the two last columns of Table II can be taken to vary approximately *pro rata*, although this is not always strictly correct: for instance, in the case of a cheap solid fuel, with a large percentage of ash, the labour costs would be increased.

It must be remembered that the heat produced in the boiler, whatever fuel is used, is not, as a general rule, all introduced into the various rooms to be heated. There is a certain loss from the pipes conveying the heat to the required point, though this may be a very small percentage if the mains are properly insulated to prevent loss of heat.

There is one aspect of the matter that, owing to the possible great variation, is not fully taken into consideration in the above figures, and that is the excess of fuel that might be used, particularly in the case of solid fuel where the boilers have to be kept lighted, but not doing the full amount of useful work compared with the fuel consumed. It will be appreciated that this figure would vary with different classes of buildings and different fuels, but as between solid fuel and electricity, 15 per cent. might be taken as a fair average. It will be appreciated that there is also the question of cleanliness and the space occupied by the fuel to be considered.

FUEL CONSUMPTION.

In estimating the fuel consumption for any installation it should be remembered that in this country during a normal winter there are relatively few days when the full heating effect is required, the full boiler power only being necessary in case of severe weather, or in order to raise the heat quickly, say after a period of non-occupation of the building.

In practice it is found that the fuel consumption during an ordinary heating season works out at about one-half to two-thirds of the theoretical consumption if the installation were worked at full power during the period of occupation of the building.

As a guide, the fuel consumption (generally coke) may be taken, for office blocks and similar buildings, as 150 lb. per week per 100 sq. ft. of surface installed. This is equivalent to 134 tons per season of 30 weeks for every 1,000,000 B.Th.U.'s emission for which the system is designed. This is for heating only, and an allowance has generally to be added for hot water supply.

For large institutions, etc., the following figures may be taken as the average fuel consumption per annum for all purposes, including laundry and kitchen.

Mental and Public Assistance Institutions . .	$2\frac{1}{2}$ to 3 tons per person.
Hospitals	4 to $4\frac{1}{2}$,, ,, ,,
Sanatoria	$3\frac{1}{2}$ to 4 ,, ,, ,,

If a more close estimate of fuel consumption is required, the method usually referred to as 'degree days' is adopted. Taking the required temperature of the building as the datum level, the days on which the external temperature falls below this level are obtained from meteorological records, the mean temperature by which each of these days falls below the required level being summed up and compared with the number of days during which it is anticipated the heat will be required multiplied by the temperature difference for which the installation is designed. This will give a percentage approximating to the figures given above, and will, of course, vary according to the situation and degree of exposure of the particular building under consideration.

Size of Boiler-house required.

The following table indicates the minumum size of boiler-house for the various sized heating installations, and also the minimum fuel space.

Cubic Contents of Building.	Heating Surface at 20 sq. ft. per 1,000 c/ft.	B.Th.U.'s at 150 per sq. ft.	Boiler-house L. × W. × H. Ft.	Sq. ft. of Fuel Space Recommended.
50,000	1,000	150,000	10 × 6 × 7	30
100,000	2,000	300,000	10 × 8 × 7	60
250,000	5,000	750,000	14 × 9 × 8	140
500,000	10,000	1,500,000	14 × 13 × 9	280
1,000,000	20,000	3,000,000	28 × 13 × 9	570
2,500,000	50,000	7,500,000	30 × 20 × 12	1,400

The above sizes are given as a guide only, and the size and shape of the boiler-house will, of course, vary according to the type of boiler chosen. The sizes above are for the ordinary type of cast iron sectional boiler and are the minimum that should be allowed. They are for heating only, and in order to allow for hot-water boilers, etc., in addition, the space should be increased by, say, from 20 per cent. to 50 per cent. according to requirements.

When any method of ventilation is to be considered no rules, even approximate, can be given as the space required would vary according to the type and extent of the ventilating plant.

For institution work, where the plant would include steam boilers, calorifiers, pumps, etc., the total space required, including fuel storage space, may be taken as from 2 to 5 sq. ft. per head, according to the circumstances. The lower figure may be taken as an extreme case and represents an actual example where new boiler plant, etc., had to be accommodated in existing buildings.

As a further guide, the following examples are given of actual systems designed by the author.

Building.	Services provided for.	Cubic Contents of Building.	Space allowed for Engineering Equipment.		Total Space allowed.	Space for Central Station Equipment per 1,000 cu. ft. of Contents.
				sq. ft.	sq. ft.	
House (medium sized)	Heating and hot-water service.	24,600	Boiler-house and fuel store	200	200	8·0
House (large)	Heating and hot-water service.	100,000	Boiler-house Oil fuel store	400 120	520	5·2
Hospital (80 beds)	Heating and hot-water service.	350,000	Boiler-house and fuel store	820	820	2·3
Public building (medium sized)	Heating, hot-water service and some ventilation.	1,250,000	Boiler-house Fuel store Ventilation	750 280 350	1,380	1·1
Block of 50 residential flats.	Partial heating and hot-water service.	750,000	Boiler-house Oil fuel store	340 260	600	0·8
Public building (large)	Comprehensive engineering equipment, heating, ventilation, hot-water service, vacuum cleaning, artesian wells, and water-treatment plant, etc.	2,750,000	Boiler house Ventilation plant Oil storage Other equipment	1,500 3,000 1,500 2,000	8,000	3·0
Mental colony (750 patients)	Heating and hot-water service.	...	Boiler-house Fuel store	2,700 900	3,600	4·8 per person
Institution (380 residents)	Heating and hot-water service.	...	Boiler-house Fuel store	1,150 450	1,600	4·2 per person
Sanatorium (100 patients)	Heating and hot-water service.	...	Boiler-house Fuel store	970 126	1,096	10·96 per person
Municipal offices.	Heating, hot-water service and ventilation.	1,540,000	Boilers, etc. Fuel store Ventilation	1,035 500 750	2,285	1·5

In oil-fired installations it is usual to provide 6-ton tanks (1,500 gallons) for fuel storage as the oil supply companies usually deliver in 5-ton lots. For small plants, such as for private houses, delivery can be made in 2-ton lots, so that in these cases a 3-ton tank would be ample.

Special precautions are needed in storing oil, it being essential to stand the tank in a chamber sufficiently large and so constructed that it would hold the whole of the contents of the tank or tanks in case of leakage. Ventilation of the tank room should also be borne in mind, sufficient means being provided to enable this to be done mechanically if natural ventilation is not possible.

Adequate ventilation of the boiler house must always be provided by natural or mechanical means to ensure satisfactory draught.

CHIMNEY SIZES.

The following curves give some indication of the size of chimney required for the various sized installations:

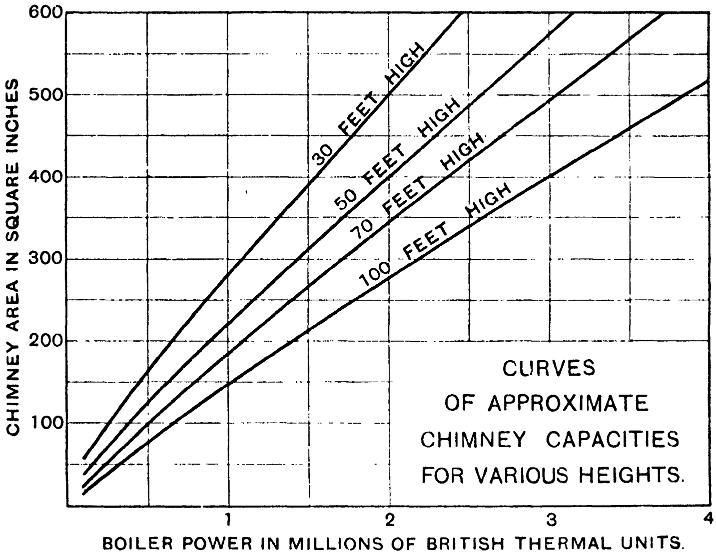


FIG. 1.

For all ordinary conditions, the sizes indicated will be found ample. Long horizontal flues and sharp bends should be avoided. Special attention should be paid to the surroundings at the top of the chimney so as to avoid any possibility of down-draught.

The sizes given by the curves are for the ordinary type of heating boiler, but if special boilers are used involving long travel of the flue gases through the boiler, or the resistance to flue gas travel is high for any reason, the heights given will need to be increased. For large installations the makers of the boilers proposed should be asked to state the draught required. Chimney and flue design is dealt with fully under Section XXVII, Part II.

Heating Systems.

The heating medium may be either water, steam, gas or electricity. In the case of water and steam, these may be distributed in various ways as follows:

Hot water	Low pressure gravity circulation. Low pressure accelerated circulation. High temperature hot water. (This is always accelerated with modern systems.)
Low pressure steam . . .	Condense through traps (live-steam or exhaust steam). Vacuum systems. Automatic vapour systems.

Hot Water.

Hot water heating systems may be designed either for circulation of the water by the gravity head set up by the difference in density between the water at flow temperature and return temperature, or for circulation by pumps, these latter being in nearly all cases of the centrifugal type.

Gravity systems are useful for small installations and where the height of the building is great in relation to the floor area. Owing to the small circulating pressure set up by the difference of temperature, pipe sizes are of necessity somewhat larger than in accelerated systems.

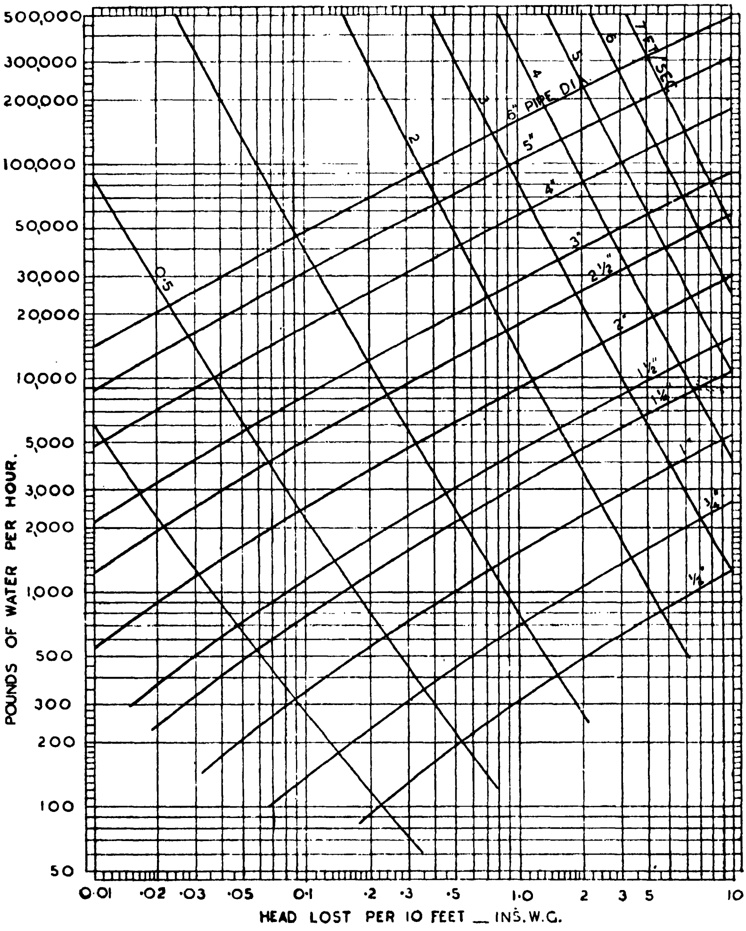


FIG. 2.

A temperature difference between flow and return of 30° F. may be taken as good practice, though to reduce the initial cost of an installation, pipe sizes are often reduced so that a much greater difference is found between flow and return. This, however, is not recommended.

Accelerated systems should be designed for, say, 15° F. to 20° F. difference between flow and return, and the pump head should be kept as low as possible consistent with reasonably sized mains, this making for low running costs. For medium-sized installations, the head might be 5 to 10 ft., whereas a head of 20 or 30 ft. or even more may be used in case of a central plant for, say, an institution with a number of scattered blocks to be served.

In favourable cases it is often possible to design the system so that normally the circulation is accelerated by a pump working at low head, but which is capable of working quite satisfactorily by gravity in mild weather, and also during times when the boiler is banked, as at night.

High temperature hot water systems are now frequently employed for factory installations, as they have certain of the advantages of the steam system without the disadvantages. Heat is provided through steam boilers which work at 150 to 200 lb. per sq. in., but steam is not drawn off from the boiler, the water in the boiler being circulated through the mains at the boiler temperature and pressure. This enables a very much larger temperature drop to be employed, a temperature drop of 80°–100° F. being common, which allows the pipe sizes to be kept smaller without introducing the necessity for a considerable amount of maintenance which results from the use of steam traps.

It is also possible to vary flow temperature between fairly wide limits by mixing some of the return water with the flow, so regulating the heat output from the installation to a larger extent than is possible from a steam plant. Particular care in design has to be taken to avoid trouble from water hammer due to accumulation of air or steam in the pipes and ample provision must be made for taking up movement of the pipes due to expansion.

Unit heaters are usually employed for distributing the heat, but if radiators are necessary, calorifiers have to be used to serve the radiators, as they will not stand the working pressure, and also it is not wise to run the high temperature hot water pipes in positions where persons can get in contact with the pipes, owing to the risk of burns.

Pipe Sizing.

The size of mains depends on the length, the quantity of water to be circulated and the head available to overcome friction. In the case of gravity circulations the available head is calculated by measuring the height of the centre of the least favourably placed radiator (known as the index radiator) above the centre of the boiler, and multiplying this dimension by the factor p, where $p = 0{\cdot}192\,(D_2 - D_1)$.

where p = circulating pressure in inches water gauge per foot of height.
D_1 = Density of water at flow temperature in lb./cub. ft.
D_2 = Density of water at return temperature in lb./cub. ft.

Table III gives the values of p for the temperatures usually obtaining in heating installations.

Table III.

Circulating pressure 'p' per ft. of height in inches W.G.

Diff. of Temp. ° F. between flow and return.	Flow Temperature °F.							
	120	130	140	150	160	170	180	190
10	0·029	0·032	0·034	0·036	0·039	0·041	0·043	0·045
15	0·043	0·047	0·050	0·054	0·058	0·061	0·064	0·067
20	0·054	0·060	0·066	0·071	0·075	0·080	0·084	0·089
25	0·067	0·074	0·081	0·087	0·093	0·099	0·103	0·109
30	0·078	0·086	0·094	0·101	0·109	0·116	0·122	0·129
35	—	0·098	0·108	0·117	0·125	0·113	0·141	0·149
40	—	0·109	0·120	0·130	0·140	0·149	0·159	0·168
45	—	—	0·132	0·145	0·156	0·167	0·177	0·187
50	—	—	0·142	0·157	0·171	0·182	0·193	0·205

In the case of accelerated circulations the size of pipe is limited by the allowable velocity of the water in the pipe, which it is usual to limit to a figure between 60 and 200 ft. per min. If the latter figure is exceeded there is a risk of noise occurring. The quantity of water circulating in both gravity and accelerated circuits is determined by the heat emission of the pipes and radiators on that particular circuit and the temperature drop allowed between the flow and return pipes. The weight of water to be circulated is calculated as follows:

$$\text{Weight of water per hour} = \frac{\text{Emission B.Th.U./hr.}}{\text{Temperature drop.}}$$

Having estimated the head available to overcome pipe resistances the pipe size can be estimated from the following equation by dividing the head available by the total length of the pipework and the equivalent length of fittings.

$$p = 174 \cdot 2 \, f \, \frac{w^2 v L}{d^5} \quad \text{where } f = 0 \cdot 0054 + 0 \cdot 375 \, \frac{\mu v}{Vd} \quad \text{(according to McAdams and Sherwood).}$$

and p = pressure lost in lb. per sq. in. d = internal diameter of pipe in ins.
w = water flowing in lb. per sec. μ = viscosity in centipoises.
L = length of pipe in ft. V = velocity in ft. per sec.
v = volume of 1 lb. of water in cub. ft.

Fig. 2, based on the above, gives the resistance in ins. W.G. per 10 ft. run of pipe.*

All pipe fittings such as bends, tees, valves, etc., and also the boiler and radiators set up a resistance to water flow. In many normal installations this additional resistance may be allowed for by adding 30 per cent. or 50 per cent. to the measured length of pipework. Where a more accurate estimate is required of the equivalent length to be added to the pipework the factors in the following table should be multiplied by the diameter of pipe in inches.

Fittings.	Factor Multiplied by Dia. of Pipe in Ins. equals approx. equivalent length in Ft.
Gate valve	0·65
Easy bend	0·9
Round elbow	1·8
Straight through tee	0·9
Straight through reducing tee	1·8
Close return bend	2·7
Tee through branch	3·5
Tank, radiator or boiler	3·5
Angle valve	15·0
Globe valve	30·0

Steam Heating Systems.

Only in rare cases is high-pressure steam used for heating. Generally the pressure used is under 5 lb. per sq. in., and in so-called vacuum systems the pressure may be below atmospheric.

In steam heating the temperature of the surface depends on the steam pressure and this may be obtained from the table on p. 3, which gives the temperature and other properties of steam at the pressures usually obtaining in the steam heating systems.

It is usually accepted that the emission from heating surface varies approximately as the 1·3 power of the temperature difference between the heating medium and the surrounding air. This is a very useful factor as, within reasonable limits, the heat transmission can be obtained for any temperature difference.

It will, therefore, be seen that with steam heating the surface required is much less than with hot-water heating. For instance, to provide an emission of 100,000 B.Th.U.'s per hour to a room at 60° F. the comparative amounts of surface required would be approximately:

With water at 160° F. : 660 sq. ft. of surface.
With steam at 212° F. : 390 sq. ft. of surface.

Steam heating, while having some advantages over hot water, such as less liability of damage from frost in case of intermittent heating, damage by water in case of a leak, is not, except possibly in industrial work, generally so suitable for our very variable climate as hot-water heating.

Steam heating may be divided into two main classes:

1. Low-pressure systems generally in which the condensate is returned by gravity to the boiler plant. Alternatively the condensate may be collected in a hotwell and pumped into the boiler.

2. Vacuum systems in which the condensate mains are connected to a pump which exhausts the air and draws away the condensate.

Low-Pressure Systems.

Ordinary low-pressure systems may be divided into single-pipe and two-pipe systems.

* Chart prepared by permission from data published by the Institution of Heating and Ventilating Engineers.

In the single-pipe system the steam and condensate are carried in the same pipes, often in opposite directions, so that unless great care is taken in designing the system, trouble may be caused by noise set up by water-hammer.

In two-pipe systems the condensate is carried in separate pipe lines and run down into a wet return, or fitted with an automatic water seal at the radiator outlet.

Size of Pipe.—The size of pipe to supply a given amount of heat at a given point depends on the length, initial pressure and the maximum loss of pressure to be allowed.

The results shown in fig. 3 have been calculated for about 1·3 lb. initial gauge pressure and 3 ins. water gauge drop and are for the two-pipe system. For single-pipe systems multiply B.Th.U.'s by 0·5.

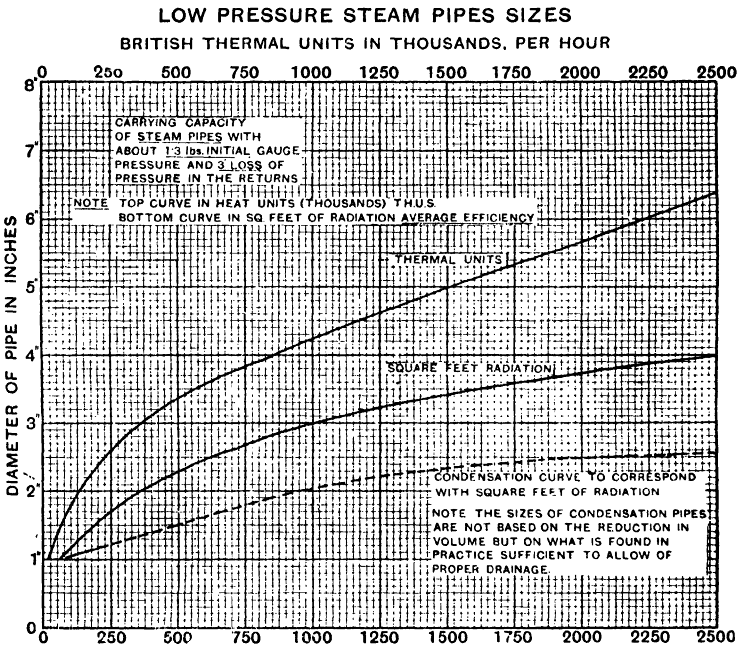


FIG. 3.

VACUUM SYSTEMS.

One of the objections to the ordinary gravity steam installation is the difficulty of displacing the air in the radiators and at the same time controlling the heat. This may be overcome by attaching a system of piping to the air valves connected to a small exhausting pump which draws away the air and forms a partial vacuum. Another way is to attach all the condense mains to a pump which will exhaust the air and draw away the condensate. This latter method is known as the 'Vacuum System.'

The piping is much the same as in an ordinary gravity steam system except that the mains are rather smaller and the condense pipes much smaller.

Owing to the thorough circulation of steam, the radiating surface has a higher efficiency per sq. ft. per degree F. difference. It should be noted, however, that this does not materially affect the fuel efficiency.

The success of a vacuum system depends to a large extent on the proper working of the traps which are fixed on the radiators or wherever condensate is likely to collect.

Hot Water Supply.

The quantity of hot water required for domestic use, or in institutions, asylums, etc., may be taken as from 20 to 25 gallons per head per day. In hospitals and sanatoria the quantity is much more, varying from 30 to 40 gallons per head per day.

For small domestic installations, the range boiler will generally supply the needs, but this type of boiler is very inefficient and in all but the smallest houses might well be replaced by an independent boiler.

With independent boilers care should be taken to see that the boiler has ample surface for its duty, otherwise inefficiency will result. In hard water districts the boiler should be periodically inspected and cleaned out so that the deposit which has collected is removed.

It is recommended, where the water is hard, that the indirect method of heating the water should be employed. In this method the hot water supply, *i.e.* the water actually drawn off from the taps, is heated in an indirect cylinder or calorifier, the primary of which is in turn heated by an ordinary heating boiler. Thus the primary water, *i.e.* the water actually in contact with the hot surface of the boiler, is not changed, and the amount of deposit is limited. In addition, the heating surface in the indirect cylinder or calorifier is at a much lower temperature than that of the boiler plates with which the water is in contact when heated directly and there is much less tendency for scale to form on the surfaces.

Where steam, either live or exhaust, is available, steam calorifiers are the most common means of providing the hot water supply, and lend themselves to easy thermostatic control. (See notes on ' Use of Exhaust Steam ' given on p. 673.)

In small or medium-sized installations, the water may be heated by gas or electricity, the system being entirely automatic. These two methods are generally found to be too expensive for large installations, but they do, however, provide an easy and convenient means of obtaining small quantities of hot water at irregular intervals.

Boilers and Boiler Power Required.

Boilers for direct heating of the water should be as simple as possible and designed so that all points are easily accessible for cleaning, especially in districts where the water is at all hard. In soft water districts special attention should be paid to the material of which the boiler is constructed. In some places, the only material suitable for the boiler, in fact for the whole system, is copper.

In determining the size of the boiler required, it may be assumed in all ordinary cases that the water will need to be raised from 50° F. to 150° F. Thus, if the quantity of water required is Q gallons, the heat required in B.Th.U.'s per hour, is

$$\frac{10Q \times 100}{T} + \begin{array}{c}\text{Radiation losses from storage}\\ \text{cylinder and pipework.}\end{array}$$

where T is the time in hours allowed for obtaining the required temperature.

An ordinary hot water boiler will transmit, say, 2,500 to 7,000 B.Th.U.'s per hour per sq. ft. of surface, the lower figure being for the secondary surface, and the higher figure for the fire-pot surface. An average figure of 5,000 B.Th.U.'s per sq. ft. per hour may be taken for ordinary purposes.

In large systems the load on the boilers on account of the hot water supply may be taken as the equivalent of 3 gallons per head per hour (150° F.) provided ample storage is arranged to even out the demand. This is approximately equivalent to 3 lbs. of steam per head per hour, where steam is the heating medium.

Storage and Distribution System.

It cannot be too well stressed that whatever system is employed for heating the water, ample storage should be allowed. This provides a good reserve in case of heavy demands on the system, and has the advantage that the load on the boilers is levelled out, tending towards higher efficiency.

For a private house with, say, two baths and six or seven other draw-offs, the storage should be, say, 75 gallons. In institution work a good figure for hot water storage may be taken as 5 gallons per head (minimum).

All storage vessels and circulations should be adequately lagged to prevent heat losses. For the same reason, the temperature at which the water should be circulated should not exceed 140° F. to 150° F., which is ample for all normal purposes, and in large systems where really hot water may be required at certain points—say in the kitchen for washing up greasy utensils—it is often advantageous to boost up at these points with gas, electricity or steam, maintaining the general supply at the above temperature. To guard against undue wastage of water, the circulations should be run as near the actual taps as practicable, thus avoiding long draw-off pipes with ' dead ' water which must be run to waste before hot water is obtained.

The following draw-off rates may be taken as average figures in determining pipe sizes:

Bath (private)	5 gal. per min.
Bath (public)	8 ,, ,, ,,
Sink	4 ,, ,, ,,
Lavatory basin	1½ ,, ,, ,,
Shower nozzle (spray type)	2 ,, ,, ,,

In designing the circulation for hot water supply it should be remembered that the pipes must be sized to supply the quantity of water necessary to meet the maximum demand at the taps. This maximum demand in the case of a small installation will be when all taps are open together, but in larger installations the simultaneous demand at any one time will be less than the possible maximum and it is usual to make suitable allowance for this according to the size of installation. Fig. 4 gives an approximation for this purpose.

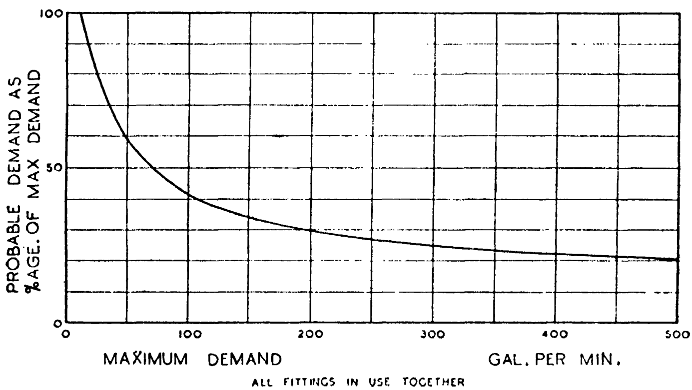


FIG. 4.

The flow of water to the taps will be along both the flow and return pipes, but since it is desirable to limit the admixture of the cooler return water as much as possible it is usual to size the flow on the assumption that this pipe will supply 75 per cent. of the demand and to size the return to take care of the flow required to make good the heat losses from the system when no draw-off is taking place. The flow from any tap depends on the difference in level between the point at which water is drawn off and the level of the water in the cold water storage tank feeding the system, and also on the loss in pressure due to friction of the water flowing in the pipes including the cold feed from the storage tank. This basis is used for sizing the flow. The return is sized on the heat loss from the system in a similar manner to that employed for a heating system. The circulation may be by gravity or accelerated by a pump according to conditions and the size of installation. In some cases, where the feed tank cannot be placed high enough booster pumps may be required to give the necessary delivery at the taps. Certain economies in the size of pipes can sometimes be made by the employment of head tanks arranged to take care of momentary peaks.

Use of Exhaust Steam.

The use of exhaust steam from prime movers for such purposes as heating or hot water supply, etc., is to be recommended wherever possible, as this allows of the latent heat in the steam being utilised, which cannot be used for power production.

The reason for the relatively low thermal efficiency of the electrical stations throughout the country, supplying the grid, is largely the non-utilisation of the exhaust steam, and it is possible for a small generating plant, such as that in a public institution or industrial concern where exhaust steam can be used, to have a very much higher thermal efficiency than the largest condensing super-power station.

The exhaust steam can either be taken direct from the engines for direct steam heating, or it can be used in calorifiers to heat water which is pumped round the heating systems.

In many institutions, hotels, etc., it is found that there is practically no difference in their annual coal consumption whether electric current is purchased from outside or generated on the

premises, as there is a demand for heat all the year round which can be met by the exhaust steam from the power plant, so that the extension of this system to general domestic heating from the existing small power plants would enable these small power plants to run at a greater efficiency than any large condensing plant.

These combined systems are of great use and lead to economical working in such cases as laundries, works where process steam is required, hotels, institutions, etc., where it is often an economical proposition to generate the electric supply, the exhaust being utilised for such purposes as heating, hot water supply, etc. In these cases, the power or electric current may be often looked on as a bye-product of the steam-raising plant, the actual cost of the energy used in producing power or current being extremely small. The chief factor in many cases is the interest and sinking fund and capital outlay involved. Even so, taking all costs into account, including the extra labour involved, electricity can often be generated at, say, 1½*d*. per unit. Each case, of course, must be gone into on its merits, but this may be taken as a figure quite often obtaining in actual practice.

Taking 1 sq. ft. of steam radiation as requiring 250 B.Th.U.'s per hour, 1 lb. of steam per hour will supply about 4 sq. ft. of radiation. A rough and ready rule for finding the radiation which may be supplied by any engine is to multiply the horse power by 130.

When exhaust steam is employed for heating, it is necessary to extract the grease by means of a separator, so placed as to intercept the grease before the steam enters the heating mains unless the engine is designed to run without cylinder lubrication or other means are taken to ensure clean steam. Usually it is necessary to have an outlet to atmosphere which is closed by a back pressure valve so that any steam which cannot be absorbed in the heating or other system will be discharged to atmosphere without building up back pressure. The back pressure valve may be set for, say, 5 lb. per sq. in., to ¼ lb. per sq. in., the latter pressure only being permissible where a vacuum return line is maintained as in the vacuum system.

See also Descriptive Section XXXIV, Part I.

Liptak Furnace Arches, Ltd.

SECTION XXXIV

PART II

VENTILATION.

(Revised by L. Copeland Watts, M.I.Mech.E., P P.I.H.V.E., A.C.G.I.)

Amount of Air necessary for Ventilation.

The air required for ventilation can be obtained in one of two ways: (*a*) natural ventilation, and (*b*) mechanical ventilation. While natural ventilation is quite satisfactory for small rooms not very thickly populated, or in cases where there is a large window area which can be kept open, in buildings such as theatres and cinemas, offices and similar buildings where the density of population is great, mechanical ventilation is usually employed. For theatres and cinemas there are usually local regulations to be complied with stating the quantity of air which must be provided per person per hour. The L.C.C. Regulations lay down that the minimum quantity of fresh air to be provided per person is 1,000 cu. ft. per hour. In certain classes of buildings, such as public swimming baths which are used as public halls or dance halls during the winter, this quantity may be reduced in certain cases to 750 cu. ft. per hour. It can be taken that these figures are the absolute minimum which should be provided, a more satisfactory quantity being from 1,800 to 2,000 cu. ft. of fresh air per person per hour. In cases where complete air conditioning is installed and cooling is effected in the summer, while this quantity of fresh air should still be introduced, the total quantity including recirculated air should preferably be in the neighbourhood of 3,000 cu. ft. per person per hour.

In the case of offices where the number of occupants in the room is frequently unknown at the time the installation is designed, it is usual to base the air quantities on a rate of change in which case not less than three changes per hour should be provided.

Distribution of Air.

It is not sufficient merely to introduce the requisite quantity of air into a room, it is important that it should also be well distributed so that every part of the room obtains its proper share without draught. This needs considerable care when designing installations dealing with large quantities.

When mechanical ventilation is employed the air is moved by fans, these usually being situated at some point remote from the room or rooms being ventilated, the air to and from the rooms passing through ducts connecting the gratings with the fans.

There are three methods of mechanical ventilation employed, the particular method depending on circumstances:

(*a*) *Mechanical extract only.*—Inlets are provided at a low level, preferably behind radiators, the air being extracted by means of a fan at high level. This method should not be employed in halls seating over 500 people.

(*b*) *Mechanical inlet only.*—Inlets are provided at a low level or about 6 ft. above floor level. A common distribution is one-third of the total area of gratings at low level and two-thirds at 6 ft. up. Outlets are provided through the walls at high level or through the ceiling. This method is employed when fresh air from outside is not otherwise obtainable.

(*c*) *Both mechanical inlet and extract.*—In this case air is drawn in and extracted by fans. It is usual to extract approximately 75 per cent. of the amount of air introduced by the inlet fan. As to the situation of the inlet and extract gratings, this depends very much on the design

of the building and the most convenient position for them. This system, which is frequently referred to as a 'balanced system,' is employed in underground rooms, theatres and other rooms where open windows are undesirable as, for instance, in noisy or dirty situations.

There are three main systems of air distribution employed :

(1) *Upward Ventilation.*—In this case the fresh air is introduced through gratings at low level, as described in (*b*) above, the air being extracted through gratings at high level, or in the ceiling over.

(2) *Cross Ventilation.*—In this case the air is introduced through gratings along one side or end of the hall, the extract gratings being provided in the opposite walls. In this case it is usual to arrange for the gratings to be approximately 6 ft. above floor level.

(3) *Downward Ventilation.*—The downward system of ventilation is often adopted in theatres and cinemas ; the air is introduced at high level, commonly in the ceiling, and extracted through a large number of well distributed extract outlets at low level. These are often of the mushroom type fixed under seats. Downward ventilation is particularly advantageous when air is introduced at a temperature lower than room temperature, in order to give a cooling effect in the summer, the incoming air thus falling gradually and being well diffused before reaching the occupied zone, which tends to avoid a feeling of discomfort and complaints of draught.

Where circumstances allow, it is often found advantageous to arrange the distributing ducts in such a way that the direction of the air currents in the room can be reversed, the particular direction of air flow depending upon outside atmospheric temperatures. The rate of air movement through the room should, of course, be kept very low in order to avoid a sensation of draught. It is, however, advantageous to be able to direct a perceptible movement of air through a room when the external temperature is high in cases where cooling is not provided. A stream of air moving at about 600 ft. per minute gives a decided feeling of comfort when the temperature of the air is above 75° F.

In certain types of building where it is essential to keep the floor space clear, as, for instance, dance halls and skating rinks, high velocity jets are sometimes employed, arranged to throw streams of air across the room at a little above head level, so inducing an air movement below, the air velocity being so arranged that the whole of the air throughout the room is disturbed and changed.

Should it be necessary to trace the air movement in a room, this can conveniently be done by blowing air through HCl and NH_4OH, the vapour so formed causing, when mixed, a dense white smoke which is at atmospheric temperature. Alternatively, titanium tetrachloride can be used. If a glass rod is dipped into the liquid and held in the air it gives off a dense white smoke also at atmospheric temperature. Since the white smoke so formed is approximately at air density, the drift indicates the direction and intensity of the air currents.

Air-Filtering, Washing and Conditioning.

It is always desirable and frequently essential that the air should be as free from dust and impurities as possible, This dust can be removed by filters and air washers. There are several types of air filters obtainable, but they can be divided into three main groups :

(i) *Dry Filters.*—These consist of a series of perforated plates arranged so that the perforations are staggered, the plates being covered with a sheet of wadding on the up-stream side for the better collection of the dust. The filtering effect is dependent on the number of plates. To clean, the filter unit is detached from its frame and shaken.

(ii) *Fabric Filters*—in which air passes at low velocity through muslin, canvas or cotton wool specially made for the purpose and stretched on frames or over supports, designed so as to give a large area of fabric compared with the space occupied by the complete filter unit. The resistance of this type of filter increases as dust is deposited and the filter has either to be cleaned or the fabric renewed periodically. With low velocities through large areas of suitable fabric, high efficiencies are obtained.

(iii) *Viscous Fluid Filters.*—In these the air passes over a series of plates or troughs coated with a viscous fluid, usually a special oil, to which the dust adheres. Certain designs employ steel wool or other fillings, arranged so as to ensure that there is no direct passage for the air through the filter. These also are coated with similar viscous fluids. To clean, the filter unit is washed in hot, strong soda water and then dipped in the viscous fluid and allowed to drain before replacing.

A development of this type of filter is the continuous viscous filter in which the units are mounted on a continuous band arranged so that each unit in turn passes through an oil bath which removes the dirty oil and replaces it with clean oil, usually hand operation being employed to rotate the filter at intervals, say, every day or two.

Air Washers.—These commonly consist of a series of fine water sprays through which the air passes mixing thoroughly with this finely divided spray, afterwards passing over a series of eliminator plates which collect the surplus water together with the dust.

The use of dry or viscous filters removes the dust without altering the temperature on humidity of the air, while the air washer varies both the temperature and humidity thus

making it unsuitable for certain purposes although it is frequently advantageous, especially when applied to the ventilation of halls and theatres.

The efficiency of dust removal by air washers is not usually very high, particularly if the dust is of an oily or greasy nature. When a greater cleaning efficiency is required than can be obtained with an air washer alone, it is common to place a series of dry or viscous filters after the air washer to remove a proportion of the dust remaining in the air after leaving the washer.

Air Cooling.

The endeavour to produce a pleasant and healthy inside atmosphere at all times of the year, whatever the outside conditions may be, has received a good deal of attention of late years. If, however, it were 90° F. outside, it would be very unwise and actually uncomfortable for people to go into a temperature of, say, 60° F., or even 70° F. In this case, the reduction of the dry bulb temperature by about 10° F., together with a perceptible air movement, would probably give rise to a marked degree of comfort. The most desirable relative humidity for public halls and offices is between 50 per cent. and 60 per cent.

Air can be cooled by an air washer, but the relative humidity will always rise. In a washer using recirculated water, the heat content will not be substantially changed although a certain cooling effect can be obtained with a consequent rise in relative humidity. The wet bulb temperature is unaffected by this method, so that the cooling is limited by the initial wet bulb temperature and will in practice always be a few degrees above this. Actually it will be found that the dry bulb temperature can be reduced by about 75 per cent. of the difference between the wet and dry bulb temperatures.

When artesian well water or some other water having a temperature of 52° F. is obtainable, the air can be cooled to about 57° F. with a consequent reduction in the vapour content of the air should the initial dew-point be above 57° F., which is most probable in hot weather. The air will then be saturated.

When air is required with a lower temperature than can be obtained by the air washer, refrigeration will have to be employed. Cooling air is far more expensive than heating. For general ventilation, the air may be cooled from 10° F. to 15° F., but for industrial purposes, when it is required to have complete control over the relative humidity and temperature, it is usual to cool the air to 35° F. The amount of brine to be circulated should be calculated on a drop of 3° F. The specific heat of brine may be taken as 0·82 and in calculating the heat required for the reduction in temperature the condensation of the vapour must not be forgotten. The heat in British Thermal units per hour, divided by 12,000, will give the size of plant required in tons of refrigeration per day of 24 hours.

The heat exchange in cooling or heating air over a given range of temperature is not materially different, provided the dew-point is not reached. Should, however, the initial humidity be such that the dew-point is reached in the cooling, another very important factor is introduced, a portion of the aqueous vapour being condensed and giving up its latent heat in the process. The British Thermal Units involved in the heat exchange may be ascertained by the following, which is accurate enough for all practical purposes and shows the various steps involved :

$$\begin{array}{ccccc} \text{SECTION A.} & & \text{SECTION B.} & & \text{SECTION C.} \\ U = L_e(W_1 - W_2) & + & S_v(T_1 - T_2)W_1 & + & S_a(T_1 - T_2) \end{array}$$

where

T_1 = initial dry bulb temperature ;

T_2 = final „ „ „

W_1 = weight of aqueous vapour per lb. of dry air at initial temperature T_1 ;

W_2 = „ „ „ „ „ „ „ „ „ „ final temperature T_2 ;

L_e = mean latent heat of vaporisation or condensation at initial dew-point, and T_2

S_v = specific heat of vapour ;

S_a = specific heat of air ;

U = B.Th.U.s to cool 1 lb. of air from T_1 to T_2 ;

Basic figures to be taken for this and other equations :

S_v = specific heat of aqueous vapour = 0·444 ;

S_a = specific heat of air = 0·2400.*

L_e = latent heat of vaporisation varies from 1073·4 B.Th.U.s per lb. at 32° F. to 1052·3 B.Th.U.s per lb. at 70° F.

Section A is the heat required to be extracted to condense the water vapour.

Section B is the heat required to be extracted to reduce the temperature of the water-vapour content of the air and is so small that it can usually be neglected.

Section C is the heat required to be extracted to cool the air from T_1 to T_2.

* 0·238 was at one time the accepted figure for specific heat of air and will be found in many textbooks, but 0·2400 is now accepted as the more correct value for the temperatures usually obtaining in air conditioning work.

Humidity.

Saturated Air.

Air is said to be saturated when it has mixed with it the maximum possible amount of vapour. The maximum amount of vapour in a given volume is dependent on the temperature.

Dew-point.

This is the temperature corresponding to saturation (*i.e.* 100 per cent. humidity) for a given weight of vapour per cu. ft. Cooling of any vapour beyond dew-point results in condensation of part of the vapour as moisture. If a mixture of air and water vapour is cooled to dew-point the air becomes saturated—relative humidity 100 per cent.—and any further cooling causes part of the vapour to separate out as moisture, giving up its latent heat.

Wet and Dry Bulb Hygrometer.

This instrument provides a ready method of determining the relative humidity, dew-point, etc., of the air. It consists of two ordinary thermometers, the bulb of one of which is kept moist by means of a loose cotton wick which is either moistened before use or which dips into a vessel of water. The most reliable readings are obtained when the air is moving quickly relative to the instrument and the sling psychrometer is a useful means of obtaining the necessary air movement. This instrument consists of a frame with the two thermometers mounted thereon which can be whirled round rapidly. The thermometers should be whirled for about half a minute and the readings taken. This should be repeated several times until two successive readings agree.

The hygrometric chart, fig. 1, gives the relative humidity, dew-point, and grains of water vapour per cub. ft. corresponding to different thermometer readings. For instance, if dry bulb

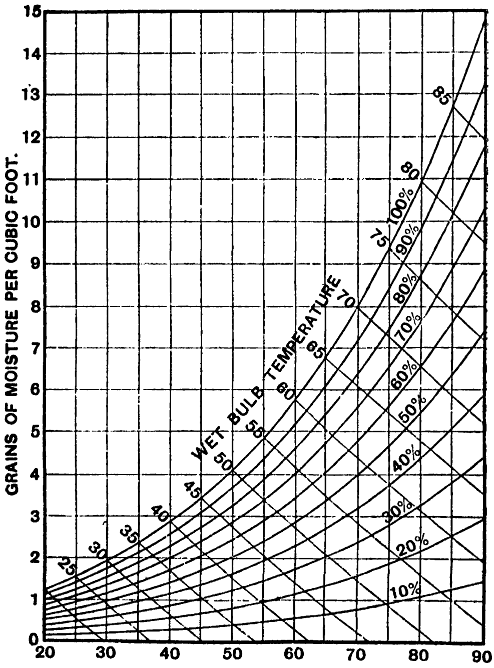


FIG. 1.

temperature is 70° F. and sling wet bulb temperature is 60° F., the relative humidity is 55 per cent., the dew-point is 53° F., and the weight of vapour per cub. ft. is 4·4 grains. Vapour in a cub. ft. at saturation point is 7·98 grains at above temperatures.

Velocities of Air Flow for Mechanical Ventilation and Flow of Air Through Ducts.

Where silence is essential, the velocities of flow of air in ducts must be kept low, the ideal velocity being from 10 to 15 feet per second, and on no account exceeding 20 feet per second. For factory work speeds up to 30 feet per second are allowable, but this velocity should not be exceeded. In general, the outlets to the rooms (if at low level) should be of such an area that the air issues at a velocity of from 1½ to 3 feet per second, as such velocities do not give rise to a sensation of draught. In certain circumstances, however, such as during hot weather, it may be found desirable to direct a flow of air across a room at a velocity of 10 to 15 feet per second, this movement giving a feeling of coolness. (See 'Effective Temperature,' p. 662.) The following table gives the permissible velocities for different parts of the ventilation system. The lower figure refers to ventilation where silence is desired, and the higher gives the maximum permissible for factories, etc.

Fresh air intake .	500 to 800 ft. per min.	Main ducts . .	800 to 1,200 ft. per min.
Heating battery .	800 ,, 1,500 ,, ,,	Branch ducts .	600 ,, 900 ,, ,,
Filter . . .	150 ,, 350 ,, ,,	Registers . .	90 ,, 180 ,, ,,
Washer . . .	300 ,, 500 ,, ,,		

The flow of air through a building resembles an electrical circuit, the inlets, outlets, and ducts acting as the resistance, and absorbing some of the aeromotive force. *To ensure an even flow in various ducts requires considerable care*; the resistance must be low, as the initial force in ventilation is small, usually equal to about 0·25 in. of water gauge. In natural ventilation it is much less than this.

In order to obtain an even distribution of air throughout the system, the design of the ducts is very important. Changes in direction and section should be as gradual as possible, particular care being taken to avoid sudden changes. The finish of the ducts should always be of the best, as roughness and irregularities increase immensely the resistance to the flow of air. The ducts are usually made of galvanised iron or in builders' work. When made by the builders the ducts should be lined with glazed bricks, or, if this is too expensive, finished with Keene's cement or some similar hard, smooth finish. It is always advisable to whitewash the latter, as the presence of dust is then easily noticed and more likely to be removed. The importance of the foregoing will be appreciated when it is realised that in mechanical ventilation where silence is essential the fan should not work against a resistance exceeding 1 in. W.G. and in other general work it is advisable not to exceed 2 in. W.G., except for dust extraction and similar industrial processes.

There is still a need for further research work in connection with the flow of air in ducts, but the following will be found a useful formula for the flow of air through galvanised iron pipes:

$$C = 39d^{2\cdot5} \Big/ \left(\frac{L}{H}\right)^{\frac{1}{2}},$$

where,

C = cubic feet of air per minute; L = length of pipe in feet;
d = diameter of pipe in inches; H = drop of pressure in inches of water.

The values derived from this formula are shown more conveniently in fig. 2.

For resistance in ducts of other materials than galvanised iron, multiply the result obtained from the diagram or by the above formula by the following factors:

Smooth concrete . . . . 1·5 Brickwork . . . . . . 2·0

When arriving at the total resistance in a duct, allowance must be made for bends and loss due to entry and egress at registers. These losses are often assessed by calculating the length of ducting which would set up frictional resistance equivalent to the local resistances, in a similar manner to pipework on a heating system. The loss of a sharp elbow varies from 30 to 60 diameters, a smooth bend 5 diameters and a usual register 80 diameters.

To obtain the total head against which the fan has to work, the velocity head must be added to the resistance in the duct; this is obtained from the following formula:

$$h = \frac{6D \cdot v^2}{gk},$$

where,

v = velocity in ft. per sec.; h = velocity head in. W.G.;
g = 32·2 ft. per sec./sec. D = density of air at temperature required
k = density of water = 62·4 lb. per cub. ft. = 0·0749 for dry air at 70° F.

All the above is based on circular pipes. In the case of rectangular pipes, find the equivalent circular pipes to give equal discharge for equal resistance from the following formula, and then use fig. 2 to find the resistance to flow:

$$D = \sqrt[5]{3\cdot25\,\frac{A^3B^3}{A + B}},$$

RESISTANCE OF FLOW OF AIR PER 100 LINEAL FEET OF CIRCULAR GALVANIZED IRON TRUNKING.

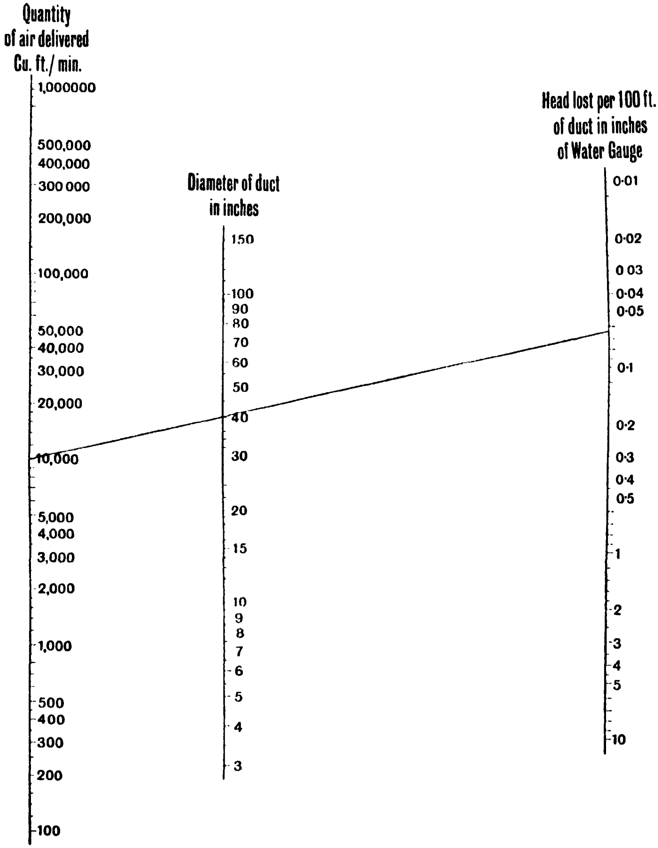


FIG. 2.

EXAMPLE: To find the loss of head in a 40-in. diam. duct carrying 10,000 cu. ft. of air per minute, lay a straight edge as shown connecting 10,000 cu. ft./min. and 40-in. diam., reading the loss of head 0·064 in. W.G. per 100 ft., on the right.

where,
A = long side of rectangular pipe ; B = short side of rectangular pipe ; D = dia. of equivalent round pipe.

This equivalent diameter can conveniently be found by means of fig. 3.

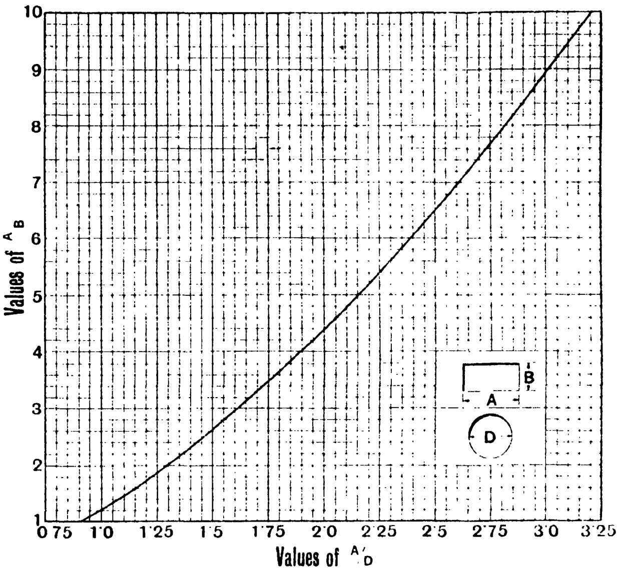


FIG. 3.

Diagram for obtaining equivalent Diameter of Round Duct carrying the same amount of air for the same loss as a Rectangular Duct.

Early consideration should be given to the position and size of ducts when mechanical ventilation is to be employed in a building.

Assuming a building in which there are, say, 3,000 people, and taking the L.C.C. requirements of 1,000 cu. ft. per hour, this means about 830 cu. ft. per second. The velocity permissible in the ducts will depend to some extent on their length, but 10 to 15 ft. per second may be taken as a good maximum average. This means that the inlet and distributing duct should have an area of not less than 55 sq. ft. As a streamline section must be maintained, it follows that where a change of direction in a duct of this size takes place, considerable space is taken up, and unless provision has been made for same, difficulties may be encountered. Steelwork is often a very important factor, especially in connection with the vertical subsidiary ducts, and is very difficult to modify ; this trouble can only be avoided by close collaboration between the various parties concerned in the early stages of the building.

It must be remembered that high air velocities and high speed fans are a source of trouble and expense however attractive it may be at the time to install small trunks that are easily adapted to the building.

Movement of Air by Fans.

The choosing of a suitable fan is a most important factor in any scheme of mechanical ventilation.

There are two distinct types of fans used for this work : the propeller and centrifugal, one similar to a ship's propeller, and the other more like a paddle wheel.

A propeller is a very efficient and useful fan if used in the proper way, that is, either with free inlet and outlet, as, for instance, when fixed in a window or wall without any trunking attached, or with short trunking of ample size, causing very little resistance head.

In the average building a fan must run at a low speed to avoid noise. At this speed the output and efficiency of a propeller fan fall off very considerably against resistance, as will be

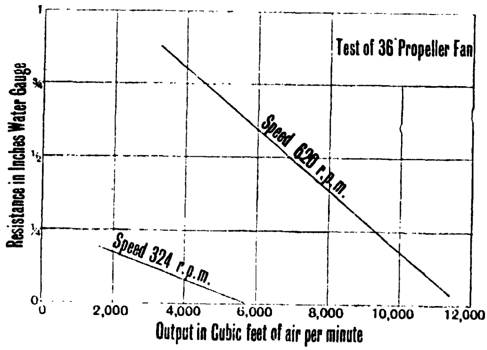


FIG. 4.

Diagram showing how the volume of air delivered by a propeller fan falls off with increased resistance.

seen from fig. 4. For factory and similar work where noise and extra power do not matter, the fan may be run at a much higher speed, in which case it can be used at a higher water gauge, see fig. 4, but even then the economical limit is soon reached. It must be remembered also that as the quantity of air decreases, due to resistance, the power increases.

During recent years, propeller fans with blades shaped to an airfoil section, have been marketed. This design of fan is known as the axial-flow type. They are more suitable for use with higher pressures than the normal propeller fan, they are more efficient, except under free air conditions, they have a non-overloading power characteristic and are quieter for equivalent size and speed.

The centrifugal fan can be used up to any reasonable pressure, but for general ventilation, where absence of noise and economy of power is desired, it is good practice not to exceed about 1 in. total head (resistance head + velocity head), the filters, heaters, ducts, etc., being designed accordingly.

In the usual type of centrifugal fan (generally called a multi-vane) the vanes or blades are curved forward (in the same direction as rotation) the resultant fan characteristic being that when the static pressure drops the volume of air increases and the power also increases, so that there is danger of overloading the driving motor unless this has sufficient margin of power and proper care is taken in calculating the resistance of the system to which the fan is to be coupled.

If the vanes are curved backwards the fan is given a non-overloading characteristic, *i.e.* a reduction in resistance does not produce a substantial increase in volume and the driving motor is not overloaded. With any given system, when varying the speed of a centrifugal fan, the quantity of air delivered varies directly as the fan speed, the resistance varies according to the square of the speed and the power taken according to the cube of the speed.

Movement of Air other than by Fans.

As the air in a flue such as a chimney or outlet duct from a warmed room is at a higher temperature than the outside atmosphere, its density is less, and the tendency is for it to rise. The resulting velocity of flow varies within wide limits, but is theoretically given by

$$V = 480 \sqrt{\frac{h(T - t)}{t + 460}}$$

where,

V = velocity of air in feet per minute;	T = temperature of heated air °F;
h = height of flue in feet;	t = outside temperature °F.

The actual velocity is considerably lower in practice, and the result from the above formula should be multiplied by 0·25 to 0·6, according to size, shape, and materials.

In addition to the movement of air due to difference of temperature, the wind has a considerable aspirating effect on a properly designed ventilator.

Heating units such as gas jets, electric heating, steam or hot water coils, are sometimes used for the purpose of causing an up-draught. These should never be used if any other method is available, as they are very inefficient and costly in up-keep, costing, in the case of steam coils, for example, on an average six times that of moving air by fans.

Silencing Equipment.

In cases where noise will be objectionable, such as offices or public halls, special precautions should be taken to avoid transmission of noise through ducts or from the moving machinery to the structure.

The aim should always be to make a system of ventilation as noiseless as possible, though this may not be necessary in industrial work. A good deal can be done by attention to details, but the fundamental factor is to keep low velocity, enabling the fan to be run at slow speed which also saves power. The space required and the initial cost will probably be more than for a system having high velocities, but it cannot be too strongly emphasised that in good class work low air velocities and slow running fans are essential to successful ventilation. Guide vanes should be employed at all changes of section and direction and any obstructions crossing the ducts should be provided with stream-lined casings.

The fan and motor should be insulated from the structure by means of cork or similar material placed between the foundations and the floor upon which this stands. It is also advisable to provide the motor with anti-vibration foundations and to employ a belt drive between the motor and the fan, using endless rubber belts for this purpose. Also the sheet metal connections to the inlet and outlet of the fan should not be bolted direct to the fan casing but should terminate a few inches from the casing, the connection being completed by a sailcloth joint. In special cases where absolute silence is essential it may be found necessary to line parts of the trunks with sound absorbing material. This may also be necessary in cases where a common trunk serves a number of rooms, the trunk connecting the various rooms being similarly lined with acoustic sheets so as to avoid transmission of noise from one room to another. An increase in the section of the trunk to allow for this material must be made so that the air velocity is not increased.

See also Descriptive Section XXXIV, Part II:—

James Howden and Co. Ltd.
F. W. Potter & Soar Ltd.
Visco Engineering Co. Ltd

SECTION XXXIV

PART III

LIGHTING.

(Revised by the Lighting Service Bureau.)

Terms in Illumination.

International Candle.—This is the unit of light intensity standardised in Great Britain, the United States of America, and France (*Bougie decimale*), and is one-tenth the intensity of the standard 10 candle Harcourt Pentane lamp burning under specific conditions of barometric pressure and humidity.

The relationship between the International Candle and other units is shown in the following table:

Unit.	Is Equivalent to: International Candle.	Hefner Candle.	Carcel Unit.
One International Candle	1·0	1·11	0·104
One Hefner Candle	0·9	1·0	0·09375
One Carcel Unit	9·6	10·66	1·0

Mean Horizontal Candle Power.—As the intensity of a light source generally varies in all directions, this term is employed to denote the mean candle power obtained by averaging the candle-power readings measured in all directions on a plane passing through the centre of the light source and perpendicular to its axis.

Mean Spherical Candle Power (M.S.C.P.).—This is the mean of the candle powers measured in all directions.

Lumen.—The unit of light flux. It is the amount of light flux contained in a unit solid angle (57·3°) of a sphere with a non-reflecting internal surface which has a uniform light source of one candle situated at the centre. A light source of one mean spherical candle emits 4π (12·57 lumens (see fig. 1).

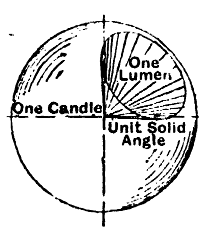


FIG. 1.

Foot Candle or Lumen per Square Foot.—The unit of illumination. If a uniform point source of light of 1 candle intensity in all directions is situated at the centre of a sphere of 1 foot radius, every square foot of the spherical surface will be illuminated to a density of 1 foot candle. Alternatively, it may be said that if one lumen illuminates an area of 1 sq. ft., the average illumination will be 1 foot candle.

Lux or Metre Candle.—The Continental unit of light density is analogous to our foot candle unit. When an area of 1 square metre is illuminated by one lumen, the resultant illumination is 1 lux or metre candle. It follows that 1 foot candle equals 10·76 lux.

Brightness.—The brightness of a surface is the quotient of the luminous intensity in a given direction divided by the area of the surface projected on a plane perpendicular to that direction.

For electric lamps and fittings this quantity is expressed in candles per square inch, while for large surfaces it is usually stated in candles per square foot.

Illumination.

Illumination, or the density of light received by an object, is measured in terms of foot candles, and can be shown to be directly proportional to the candle power and inversely proportional to the square of the distance from the light source, *i.e.* $I = \frac{C.P.}{D^2}$.

If a surface is oblique to the rays, $I = \frac{C.P.}{D^2} \cos\theta$, where θ is the angle subtended between the incident light and normal to the surface.

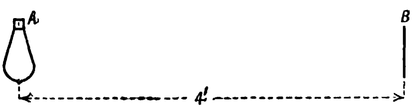


FIG. 2.

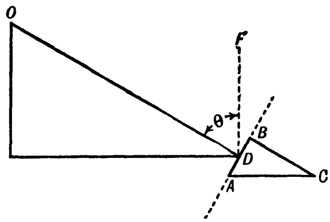


FIG. 3.

Example.—In fig. 2, if A is a lamp (electric or otherwise) giving 16 C.P. in a horizontal direction, the illumination at the point B, 4 ft. away, would be $\frac{16}{4^2} = 1$ foot candle, since the intensity of light varies inversely as the square of the distance. To get the normal illumination at any given point, the candle power in that particular direction must be divided by the square of the distance to the point illuminated. As previously explained, if the surface illuminated is not at right angles to the direction of the light, the value of the illumination obtained, as above, must be multiplied by a reduction factor, taking into account the angle at which the rays strike. A beam of light coming in the direction of OD (fig. 3) falls upon a plane AB, illuminating it with a density of 1 foot candle, then the illumination on the plane AC, which intercepts the same amount of light as AB, would be less as the light is spread over a larger surface, the reduction being in the ratio of AB to AC, which is the cosine of the angle ODF. Thus the effective illumination on any plane at a given point will be:

$$\frac{C.P}{\text{distance}^2} \times \cos\theta$$

where θ is the angle between the direction of the ray and a perpendicular to the plane considered.

NOTE:—The above is only true for a point source of light, *e.g.* a street lamp in an arterial road. In interiors, the physical size of fittings, and the reflection from walls and ceiling modify the law, so that the fall in illumination with distance is less than the amount indicated above.

BRIGHTNESS.

The brightness of an object under a given illumination depends on the amount of light reflected back by its surface, and this depends on the colour and the nature of its finish. The following table gives the approximate percentage of incident gas-filled lamp-light reflected by various coloured paints. Figures for fluorescent light are generally similar.

APPROXIMATE REFLECTING POWER.

	Per Cent.		Per Cent.
Matt White	84	Golden Brown	31
Deep Cream	70	Sea Green	38
Light Buff	61	Sage Green	19
Primrose	70	Sky Blue	47
Orange	36	Turquoise Blue	27
Salmon Pink	42	Light Battleship Grey	44
Post Office Red	21	Dark Battleship Grey	26

Photometry.

As its name implies, photometry deals with the measurement of light. By comparing the unknown light source with a known standard and using the fundamental stated above—that is, that the intensity is inversely proportional to the square of the distance—photometric readings are reduced to the measurement of lengths. In a simple photometer the standard lamp and the light source to be measured are mounted on a horizontal bench with a movable screen placed between them (see fig. 4). The screen is then moved between the lamps until both its sides appear to have the same brightness. The distance AO and BO are then measured and the candle power calculated from the fact that:

$$\frac{\text{OP.A}}{\text{AO}^2} = \frac{\text{OP.B}}{\text{BO}^2} \quad \therefore \text{OP.A} = \text{OP.B} \times \frac{\text{AO}^2}{\text{BO}^2}$$

While making such readings care must be taken that the screen is shielded from any light from other sources.

FIG. 4.

A great difficulty in making candle-power measurements is the difference in the colour of various light sources, and for accurate work special means have to be adopted. One method is to employ a flicker photometer such as the Simmance-Abady Flicker Photometer. This instrument is based on the fact that if a surface is illuminated alternately by two light sources it appears to flicker, due both to variations in light intensity and colour. By varying the speed of the alternations it is found that the eye is more sensitive to variations of intensities than to changes of colour; so by using a critical speed of exposure the surface assumes a neutral tint only affected by flicker due to variations in intensity, and so, under these conditions, the illuminations on the screen can be balanced in the usual way.

In the actual instrument these alterations are obtained by rotating a specially shaped disc between the light sources. The rotor is provided with surfaces that slope alternately to either side, so that at any instant during the rotation the portion in front of the eyepiece is only illuminated by one lamp.

INTEGRATING PHOTOMETER.

The sphere photometer is another form of instrument specially designed for measuring the mean spherical candle power of a light source. The apparatus consists of a large sphere with a whitened mat interior surface; the light source being placed in the centre. It can be shown that every part of the interior surface will receive the same amount of light by repeated reflections—in other words, the light is automatically averaged out; by placing a small window of diffusing glass in the side of the sphere this average intensity can be measured by the usual photometer method.

As mentioned, the effect depends on reflections in the interior, and so the window has to be carefully shielded from direct light from the source of light, also the readings must be multiplied by a constant depending on the nature of the internal surface.

PORTABLE PHOTOMETERS OR 'LIGHTMETERS.'

Although the measurement of candle power is important for certain purposes, we are usually more concerned with the measurement of the actual illumination reaching the working plane. Small portable 'lightmeters' have recently become available for this purpose, working on the photo-electric principle; the light falling on a light-sensitive cell generates current which is passed to a micro-ammeter calibrated in foot candles, no auxiliary apparatus being required.

These lightmeters are generally calibrated to measure illumination from gas-filled lamps, and are reasonably accurate under natural daylight or fluorescent lamp light. Where strongly coloured light is to be measured, however, a correction factor, obtainable from the makers of the instrument, should be used.

The usual range is 0–50 foot candles, while in most instruments a shunt or masking plate may be used to enable readings up to 250 or 500 foot candles to be obtained.

Polar Curves.

Before suitable reflectors can be designed for a given source of light, or in fact before any real idea can be formed as to the characteristics of such a source, it is necessary to know how the light intensity varies in all directions. The polar curve is a very convenient way of showing

these variations, and an example is given in fig. 5. The distance scaled along any line from the centre is proportional to the candle power in that particular direction. When handling light sources, the illuminating engineer needs information as to the nature of the light distribution complete with reflector. This data is recorded in a similar type of graph; and fig. 5 also shows how the light distribution has been completely changed by fitting a reflector; curve A shows the light distribution of a particular lamp alone, while curve B gives its distribution after a reflector has been installed. It will be seen that by means of the data from such polar curves, light can be properly controlled both as regards the design of fittings and installation for specific purposes.

Often it is necessary to compute the total lumens output from a polar curve, or more exactly the mean spherical candle power from which the lumens can be easily calculated. If the light source is symmetrical in its horizontal distribution and not too irregular in its vertical variations, this can be readily performed by taking readings at what are known as the Russell Angles; the mean spherical candle power being the average of such readings. Taking the lower vertical ordinate as our zero, readings should be taken at the following angles:

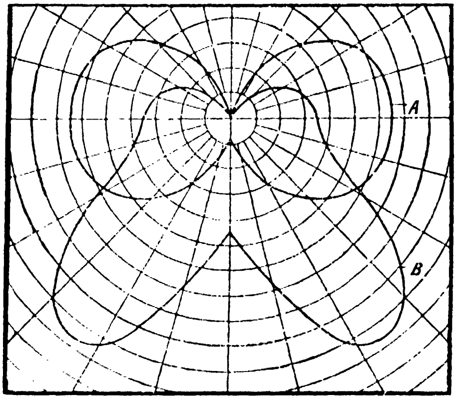


FIG. 5.

Degrees.

25·8	60·0	84·3	107·5	134·4
55·6	72·5	95·7	120·0	154·2

As explained, the M.S.C.P. is the arithmetic mean of the ten readings; while if the lumen output is required this result has simply to be multiplied by 12·57.

Requirements of Good Illumination.

Illumination must be acceptable to the eye. We should be able to see objects clearly and with a minimum of fatigue. To accomplish this result certain conditions must be fulfilled.

(1) There must be sufficient illumination. Since objects are seen by means of the light which they reflect, more light must be thrown on dark objects than on light ones.

(2) Intensely bright lights in the field of vision should be avoided. The iris closes somewhat in order to afford a protection from such lights, and the amount of light received from illuminated objects is thereby so reduced that they cannot be seen clearly. This accounts for the well-known dazzling effects of searchlights and of many headlights. The intrinsic brilliancy or candle-power per square inch of luminous area should therefore be kept as low as possible; it should preferably not be higher than 3, if the source of light is in the field of vision. Clear bulbs should certainly never be used where the filaments are continuously in view. When reflectors are used it is generally desirable to employ some form of obscured lamp.

(3) Flickering lights should be avoided. Poorly regulated circuits, such as those having varying power loads, cause disagreeable flickering. Metal filament lamps have an inherently better regulation than carbon filament lamps, and electric discharge lamps have better regulation than metal filament lamps.

(4) Where colour matching is vitally important special colour matching fittings may have to be employed but for most industrial purposes absolutely correct daylight is not required, and

fluorescent tubular lamps referred to on p. 705, which give a very close approximation to natural daylight, may be employed.

The size of the fittings is a question of design. In deciding upon the proper sizes of lamps to use, one must consider the spacing of present points, if building is already wired, and the position of pillars, beams, etc. In general the fewest fittings consistent with good distribution will be the most economical, as the total cost of wiring, reflectors, and lamps, as well as the maintenance charges, will be reduced, but a greater number of smaller fittings will give less shadow.

In many cases it is preferable entirely to rewire the premises in order to obtain correct illumination.

Illumination Calculations.

During recent years, the 'point-to-point' system referred to in the early part of this section has been generally abandoned in favour of the 'lumen' method of pre-determining the average illumination of an installation. This is essentially due to the fact that, owing to the accumulated experience of illuminating engineers, it has been found possible to take into consideration the contributory effects due to reflection from the ceiling and the walls. The following gives the method of procedure that is adopted generally at the present time :—

Steps in the Design of a Lighting System.

1. *Intensity of Illumination.*

The foot-candle illumination according to good modern practice for different classes of operations are given in Table I., pp. 691–695. In many circumstances, particularly when dealing with dark materials, higher values are desirable.

2. *Type of Lighting Fitting.*

The selection of the lighting fitting depends, not only on the requirements of the work, but in some measure on the construction of the room and the colour of the ceiling and walls. For instance, it would be undesirable to install totally indirect lighting equipment where the ceilings are very dark. The choice of a particular fitting would depend on which of the factors of glare, efficiency, shadow, appearance, etc., are considered of primary importance. A general guide to the characteristics and applications of various types of fittings is given below :—

Characteristics of Lighting Fittings.

Direct (A)—Reflector such as Standard Dispersive and equivalent reflectors for electric discharge lamps.

For industrial interiors where materials worked are not highly polished. Usually mounted at medium heights, it may occasionally be used for high bays where a sideways spread of light is desirable. Direct light from the lamp is cut off at an angle at least 20° below the horizontal. It has a high efficiency and is easy to clean. Maximum illumination is provided on the horizontal plane, fair illumination on the vertical. The appearance of the room is essentially practical, as no light goes directly upwards. Depth of shadow is moderate to hard, depending on the spacing. The reflector may be fitted with a clear or frosted dustproof cover glass, the latter reducing direct and reflected glare.

Direct (B)—Reflector with enclosed diffusing globe such as Industrial Diffusing fitting.

For industrial and other interiors where there is risk of direct or reflected glare. It gives softer shadows and requires slightly more wattage than the Direct (A) type, but (in most examples) allows some light to go upwards to illuminate the ceiling, giving a better appearance to the room.

Direct (C)—Concentrating Reflector.

For high mounting above travelling crane, etc. Most of the light is concentrated into the 0–25° zone from the downward vertical. The cut-off angle is about 30° below the horizontal. A cover glass may be fitted as for the Direct (A) type.

Direct—Laylights and Ceiling Panels.

Suitable for a wide variety of commercial interiors. If the luminous area is of sufficiently low brightness, high quality lighting can be obtained, with very soft shadows and no glare. Where the top is open to admit daylight the efficiency will be lower than when a reflecting top is used. The ceiling tends to be dark unless the laylight or panel projects slightly below the ceiling and translucent side panels are employed, or unless supplementary upward lighting is provided by other means.

Semi-Direct (A)—Enclosed Fittings with major light flux downwards.

This type is generally of good appearance, suitable for 'utility' lighting of unpretentious offices and shops. It has a good efficiency and allows 10 to 40 per cent. of the light to go upwards towards the ceiling. Some examples may be too bright for glossy materials to be viewed comfortably.

Semi-Direct (B)—Suspended luminous beam with diffusing casing.

This type which is generally designed and built to suit the particular interior, can give excellent lighting in all respects. The line of light emphasises the length or width of the room, as desired,

The appearance and efficiency of the fitting very widely according to the constructional details and the materials used. The lighting effect may be ruined by a few dead lamps, therefore maintenance should be frequent and thorough.

General Diffusing—Enclosed diffusing fitting giving approximately equal light output in all directions.

This is generally regarded as an alternative to Semi-Direct (A), with comparable efficiency but greater diffusion, and allowing more light to reach the ceiling. It is suitable also for school classrooms with ceiling height above the average. Shadows are soft and the fitting is easily kept clean. For equal average brightness, the light output in a particular direction is proportional to the projected area; therefore flat or vertical cylindrical shapes tend to give most light in the up-and-down or sideways direction respectively.

Semi-Indirect.—Fitting giving the major part of the light in an upward direction.

This usually has a very good appearance, and is suitable for offices, reception rooms, and shops where prestige is important: also for schools. It needs a greater wattage than the more direct types, but gives very little shadow or likelihood of glare. If of the open type, it requires frequent cleaning to maintain efficiency. A light-coloured ceiling is essential.

Indirect (A).—Indirect Fitting and Indirect Trough.

Suitable for providing moderately high intensities in commercial interiors without glare and with practically no shadow, but tending to give dull and monotonous lighting at low and medium illumination levels unless supplementary lighting provides bright areas to relieve the effect. This type requires relatively high wattage and frequent cleaning. The fitting or trough appears as a dark patch against the lighted ceiling unless it is designed to allow leakage of light to exterior surfaces. A very light colour of ceiling and upper walls is essential.

Indirect (B).—Indirect Cornice.

This type is particularly useful when the ceiling is required to be kept clear, *e.g.* to show ornamentation. It should not be used alone in rooms of great width in comparison to height, as sufficient light will not reach the centre part of the ceiling. The character of the lighting will be similar to the Indirect (A) type, but the efficiency varies widely with the structural details. A very light colour of ceiling and upper walls is essential.

3. *Location of Fitting.*

Tables II (*a*) and II (*b*) show the maximum desirable spacing of fittings, according to the permissible mounting height. When these or less spacings are used, reasonably uniform illumination will result.

4. *Size of Lamp.*

After the position of the fittings has been settled on the plan according to the space available the size of the lamp can be determined by consideration of the following factors:—

(*a*) *The Room Index.*—In a large room the lighting system has a higher efficiency than in a small room, owing to the fact that a number of fittings contribute directly to the amount of light that reaches the working plane, without loss due to absorption of light by walls, etc. The room index should be obtained from Table III, p. 698, taking into account the proposed mounting height of the fitting. Note that the fitting hangs down at least 1 ft. from the ceiling, and that the working plane is some distance above the floor.

(*b*) *Maintenance Factor.*—Allowance has to be made for the depreciation of lighting efficiency due to a normal deposit of dust and dirt on the lamps, fittings, walls and ceiling. Special circumstances may require a higher or lower allowance, but generally the illumination is assumed to fall by 20 per cent. due to these causes, when the installation is in service. Thus the initial illumination should be 25 per cent. higher than that required in service.

Note.—The decrease in lamp efficiency during life is catered for by using the 'average throughout life' lumen output figures shown in Tables V to VIII, p. 704.

(*c*) *Coefficient of Utilisation.*—On p. 701, Table IV, coefficients of utilisation are given for a particular fitting. Having chosen the fitting and obtained the value of the Room Index from Table III, the corresponding coefficient of utilisation, according to the nature of the walls and ceiling, can readily be obtained direct or by interpolation. This figure represents the percentage of the lumens generated by the lamp that reach the plane of work. Note that all windows, skylights, etc., should be treated as dark surfaces, unless covered by light-coloured blinds, as they permit light from the fittings to pass through and thus become lost.

$$\left.\begin{array}{c}\textbf{Lamp Lumens required}\\ \textbf{per sq. ft.}\end{array}\right\} = \frac{\text{Foot-candles}}{\text{Coefficient of utilisation} \times \text{maintenance factor}}$$

Area in sq. ft. per fitting = Transverse × longitudinal spacing (sq. ft.)

$$\left.\begin{array}{c}\textbf{Lamp Lumens required}\\ \textbf{per unit.}\end{array}\right\} = \textbf{(Area in sq. ft. per fitting)} \times \textbf{(Lamp lumens required per sq. ft.)}.$$

Having thus determined the lamp lumens required per point, the wattage of the lamp to be used can be obtained by reference to Tables V to VIII, pp. 704–706.

ILLUMINATION DESIGN FOR A MULTI-STORY MACHINE SHOP FOR ROUGH WORK.

Room is 65 ft. long and 39 ft. wide, with 13 ft. ceiling. Beams divide the ceiling naturally into 13 ft. by 13 ft. sections. Ceilings and walls painted white. Height of plane of reference 3 ft. above floor. Supply voltage 230 volts.

Summary of Steps in the Design.	Reference.	Required Data
1. Foot-candles recommended	Table I	10
2. Type of fitting used	Page 2233.	Direct (A).
3. Spacing and mounting height		
Possible mounting height above work	Depends upon ceiling height	9 ft.
Maximum spacing permissible	Table II (a), page 2241.	13½ ft.
Actual spacing to suit interior	Symmetrical with respect to bays	13 ft.
Actual mounting height above work	Table II (a), page 2241.	9 ft.
Height above floor	—	12 ft.
4. Size of Lamp		
Room index	Table III, page 2243.	F
Coefficient of utilisation	Table IV, page 2244.	0·53
Maintenance factor	—	0·8
Lamp lumens required per point	Formulæ	4000
Lamp size	Table V, page 2245.	300 watt.

Illumination Values.

The values given below are service values of average illumination. Designs planned in accordance with this book will give initially 25 per cent. more light due to the maintenance factor of 0·8.

The recommendations include most of the headings in the Table of Recommended Values of Illumination, published by the Illuminating Engineering Society and are in line with good modern lighting practice.

TABLE I.—RECOMMENDED AVERAGE SERVICE VALUE OF ILLUMINATION.

COMMERCIAL AND PUBLIC BUILDINGS.

Description.	Foot-candles	
ART STUDIO		Special lighting
AUTOMOBILE SHOW ROOM		Special lighting
CHURCH:		
Church	7	
Church Hall	7	
CINEMA		Special lighting
CORRIDORS AND STAIRWAYS	3	
DANCE HALL		Special lighting
DENTIST:		
Waiting Room	7	
Surgery (Operating Area)	70	
DRAWING OFFICE (on boards)	30	
(general)	10	
GARAGE:		
Garage	7	
Garage Repair Department	20	
GYMNASIUM	10	Special lighting for games

TABLE I (*cont.*)—RECOMMENDED AVERAGE SERVICE VALUE OF ILLUMINATION.

COMMERCIAL AND PUBLIC BUILDINGS.

Description.	Foot-candles	
HOSPITAL:		
Wards and Private Rooms	3	Supplementary local lighting
Waiting and Receiving Room	7	
Operating Table	300	
Operating Room	30	
Laboratories	20	
HOTEL:		
Lounge	7	Often special lighting
Dining Room	7	
Writing Room	7	
Kitchen	7	Bed-head lights, etc., also required
Bedrooms	5	
INDOOR RECREATIONS:		
Bowling (on Alley, Runway, and Seats)	15	
(on Pins)	20	Special lighting
Billiards (General)	3	
(on Table)	20	
Racquets, Badminton, Squash and Indoor Tennis		Special lighting
Skating Rinks	7	
LIBRARY:		
Reading Rooms (general lighting)	7	
Reading Tables	15	
Book Room	7	
MUSEUM	7	Extra lighting for showcases
OFFICES:		
General Office	20	
Typing and Book-keeping Rooms	20	
Filing Room	20	
Private Office	15	
PUBLIC HALL	7	Sometimes special lighting
REFRESHMENT ROOM	7	
RESTAURANT	7	
SCHOOL: Class Rooms	15	
Drawing and Art	20	
Gymnasiums	10	
Laboratories	15	
Lecture Theatre	10	
Sewing Rooms	20	
SHOPS:		
Interiors	10 and upwards	
Display Windows	100 and upwards	
THEATRE		Special lighting
TOILET AND WASHROOM	7	
WAITING ROOM	7	

INDUSTRIAL INTERIORS.

Description.	Foot-candles
ASSEMBLING AND ERECTION SHOPS:	
Rough Work	7
Ordinary Work	10
Medium Work	20
Fine Work, Small Mechanisms	50
Very Small Work	100

TABLE I (*cont.*)—RECOMMENDED AVERAGE SERVICE VALUE OF ILLUMINATION.

INDUSTRIAL INTERIORS.

Description.	Foot-candles	
BAKERY	10	
BOOKBINDING:		
Assembling, Cutting and Embossing	30	
Folding, Pasting, Punching and Stitching	10	
CANNING	10	
CHEMICAL WORKS:		
Hand Furnaces, Boiling Tanks, Stationary Driers, Stationary or Gravity Crystallizing, Mechanical Furnaces, Generators and Stills, Mechanical Driers, Evaporators, Filtration, Mechanical Crystallizing, Bleaching	7	
Tanks for Cooking, Extractors, Percolators, Nitrators, Electrolytic Cells	10	
CLAY PRODUCTS AND CEMENTS:		
Grinding, Filter Pressing, Kiln Rooms	7	
Moulding, Pressing, Cleaning and Trimming	10	
Enamelling	10	
Colouring and Glazing	10	
CLOTH PRODUCTS:		
Cutting, Inspecting, Sewing, Cloth Treating (Oil Cloth, etc.), Medium Colours	20	
Pressing	10	
COAL BREAKING AND WASHING, SCREENING:		
Control Points	7	Band lighting extra
Picking Belt	20	
DAIRY PRODUCTS	10	
DIE SINKING	100	
ELECTRICAL MANUFACTURING:		
Battery Manufacture, Coil and Armature Winding, Mica Working, Insulating Processes	20	
ENGRAVING:		
Stone and Machine	50	
FLOUR MILLING:		
Cleaning, Grinding, or Rolling	10	
Baking or Roasting	10	
Flour Grading	50	
FOUNDRY:		
Charging Floor, Tumbling, Cleaning, Pouring, and Shaking Out	7	
Rough Moulding and Core Making	10	
Fine Moulding and Core Making	20	
GLASS WORKS:		
Mix and Furnace Rooms	5	
Grinding, Cutting Glass to Size, Silvering, Pressing, Glass-blowing Machines	10	
Fine Grinding, Bevelling, Inspection, Etching, and Decorating	20	
Glass Cutting (cut glass), Inspecting fine	50	
GLOVE MANUFACTURING:		
Cutting, Pressing, Knitting, Sorting, Stitching, Trimming and Inspecting	20	
HAT MANUFACTURING:		
Dyeing, Stiffening, Braiding, Cleaning, Refining, Forming, Sizing, Pouncing, Flanging, Finishing, Ironing	10	
Sewing	20	

Table I (*cont.*)—Recommended Average Service Value of Illumination.

Industrial Interiors.

Description.	Foot-candles	
Inspecting:		
Ordinary	10	
Medium	20	
Small	50	
Very Fine	100	
Minute	200	
Jewellery and Watch Manufacturing	100	
Laundries and Dry Cleaning:		
Receiving, Sorting and Checking	15	
Washing	7	
Calendering	10	
Ironing and Pressing	15	
Despatch	7	
Leather Manufacturing:		
Cleaning, Tanning and Stretching	7	
Cutting, Fleshing and Stuffing	5	
Finishing and Scarfing	15	
Leather Working:		
Cutting, Scarfing, Grading, Matching, Sewing (medium colours)	30	
Pressing and Winding (medium colours)	15	
Machine Shops and Fitting Shops:		
Rough Bench and Machine Work	10	
Medium Bench and Machine Work, Ordinary Automatic Machines, Rough Grinding, Medium Buffing and Polishing	20	
Fine Bench and Machine Work, Fine Automatic Machines, Medium Grinding, Fine Buffing and Polishing	50	
Very Fine Bench and Machine Work Grinding (fine work)	100	
Packing:		
Crating	7	
Boxing	7	
Paint Manufacturing	15	
Paint Shop:		
Dipping, Spraying, Firing	7	
Rubbing, Ordinary Hand Painting and Finishing	15	
Fine Hand Painting and Finishing	30	
Extra Fine Hand Painting and Finishing (Automobile Bodies, Piano Cases, etc.)	70	
Paper Box Manufacturing:		
Fancy boxes	15	
Cartons	7	
Paper Manufacturing:		
Beaters, Grinding	7	Fittings grouped round work
Calendering	10	Fittings grouped round work
Finishing, Cutting and Trimming	20	
Power House:		
Boilers, Coal and Ash Handling, Storage Battery Rooms, Auxiliary Equipment, Oil Switches and Transformers, Engines, Generators, Blowers, Compressors	7	
Printing Industry:		
Matrixing and Casting, Miscellaneous Machines, Presses	10	
Proof Reading, Lithographing, Electrotyping, Linotype, Monotype, Type-setting, Imposing	20	
Type-setting by hand (up to 6 point)	30	
Rubber Manufacturing and Products:		
Calendars, Compounding Mills, Tyre and Tube making, Fabric Preparation, Creels	7	
Sheet Metal Works:		
Miscellaneous Machines, Bench Work, Punches, Presses, Shears, Stamps, Welders, Spinning	15	Very well diffused
Tin Plate Inspection		Special lighting

TABLE I (*cont.*)—RECOMMENDED AVERAGE SERVICE VALUE OF ILLUMINATION.

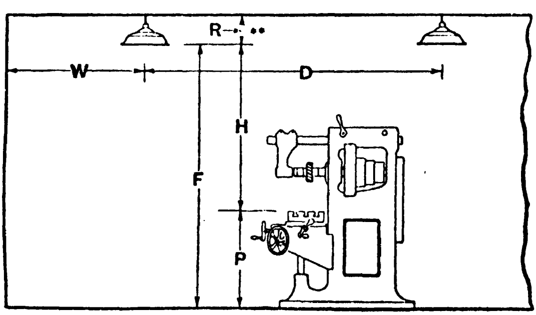
INDUSTRIAL INTERIORS.

Description.	Foot-candles
SHOE MANUFACTURING:	
Hand Turning, Miscellaneous bench and Machine Work, Cutting, Lasting and Welting	15
Stitching, Inspecting and Sorting	20
SMITH SHOP:	
Forging and Welding	7
SOAP MANUFACTURING:	
Kettle Houses, Cutting, Soap Chip and Powder	7
Stamping, Wrapping and Packing, Filling and Packing Soap Powder	7
STEEL AND IRON MILLS, BAR, PLATE, AND WIRE PRODUCTS:	
Charging and Casting Floors	7
Muck and Heavy Rolling, Shearing rough by gauge, Pickling and Cleaning	7
Automatic Machines, Rod, Light and Cold Rolling, Wire Drawing, Shearing, fine by line	10
SUGAR GRADING	50
SWEET MAKING	10
TESTING:	
Rough	7
Fine	50
Very Fine Instruments, Scales, etc.	100
TEXTILE MILLS:	
Cotton:	
Bale Breaking, Scutching, Carding, Combing and Roving.	7
Twisting and Winding	10
Spinning	10
Warping	15
Looms:	
Dark Colours (Fine Counts)	30
Light Colours (Fine Counts)	20
Grey Cloth	10
Silk:	
Winding and Throwing	15
Quilling, Warping, Weaving and Finishing	20
Woollen:	
Scouring, Washing, etc.	7
Carding and Combing (white), Blending, Drawing, and Roving	7
Spinning, Winding, Sorting, Combing (coloured), Twisting	15
Warping	20
Weaving:	
Looms:	
Fine Worsteds	50
Medium Worsteds and Fine Woollen	30
Heavy Woollen	10
Burling and Mending	50
Perching	70
TOBACCO PRODUCTS:	
Grading and Sorting	20
Stripping	7
UPHOLSTERING:	
Automobile, Coach and Furniture	15
WELDING:	
Fairly Small Soldering and Contact Welding	20
Ordinary	15
Flame Welding and Brazing	7
WOOD WORKING:	
Rough Sawing and Bench Work	7
Sizing, Planing, Rough Sanding, Medium Machine and Bench Work, Glueing, Veneering, Cooperage	15
Fine Bench and Machine Working, Fine Sanding and Finishing	20

TABLE II (a)—SPACING—MOUNTING HEIGHT.

Direct Lighting Units, including Semi-Direct and General Units.

Mounting Height of Unit. Above Plane of Work. (H)	Maximum Distance between Points. (D)	Maximum Distance between Points and Sidewalls.	
		In Usual Locations where Aisles and Storage are Next to Wall. (W)	In Offices or where Work Benches are Next to Wall. (W)
4	6	3	2
5	7½	3½	2½
6	9	4½	3
7	10½	5	3½
8	12	6	4
9	13½	6½	4½
10	15	7½	5
11	16½	8	5½
12	18	9	6
13	19½	9½	6½
14	21	10½	7
15	22½	11	7½
16	24	12	8
18	27	13½	9
20	30	15	10
22	33	16½	11
24	36	18	12
27	40½	20	13½
30	45	22½	15
35	52½	26	17½
40	60	30	20

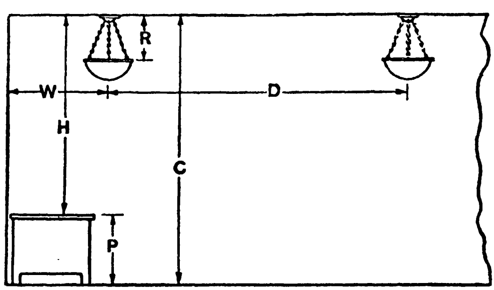


** Minimum allowance for (R) usually 1 ft.

TABLE II (*b*) SPACING—MOUNTING HEIGHT.

Semi and Totally Indirect Lighting Units.

Ceiling Height. Above Plane of Work. (H)	Maximum Spacing Distance Between Points. (D)	Maximum Distance Between Points and Sidewalls. In Usual Locations where Aisles and Storage are Next to Wall (W)	Maximum Distance Between Points and Sidewalls. In Offices or where Work Benches are Next to Wall (W)	Suspension Distance Ceiling to Top of Fitting* (R)
5	7½	3½	2½	1¼
6	9	4½	3	1½
7	10½	5	3½	1¾
8	12	6	4	2
9	13½	6½	4½	2¼
10	15	7½	5	2½
11	16½	8	5½	2¾
12	18	9	6	3
13	19½	9½	6½	3¼
14	21	10½	7	3½
15	22½	11	7½	3¾
16	24	12	8	4
18	27	13½	9	4½
21	31½	15½	10½	5¼
24	36	18	12	6
27	40½	20	13½	6¾
30	45	22½	15	7½
35	52½	26	17½	8¾
40	60	30	20	10



* Suspension distances (R) in Table are based on best distribution of light and efficiency of utilisation for standard units. In some installations other considerations may require a different suspension distance.

TABLE III.—ROOM INDEX.

Ceiling Height above working plane.—Feet.												
For Semi-Indirect and Indirect Fittings.		6 and 6½	7 to 8½	9 to 10½	11 to 13½	14 to 17	18 to 21	22 to 27	28 to 33	34 to 47	—	—
Mounting Height above Working plane—Feet.												
For Direct, Semi-Direct and General Diffusing Fittings.		4 and 4½	5 and 5½	6 and 6½	7 to 8½	9 to 10½	11 to 13½	14 to 17	18 to 21	22 to 27	28 to 33	34 to 47
Room Width (feet).	Room Length (feet).	ROOM INDEX.										
8½–9½	8½–10	*c*	*b*	*a*								
	10–14	C	B	*b*	*a*							
	14–20	D	C	B	A	*a*						
	20–30	D	D	C	B	A						
	30–42	E	D	C	B	A						
	42 up	F	E	D	C	B						
9½–11	11–14	D	C	*b*	*a*							
	14–20	D	C	B	*a*	*a*						
	20–30	E	D	C	B	*a*						
	30–42	E	D	D	C	B						
	42–60	F	E	D	C	B						
	60 up	F	E	E	D/C	C						
11–12½	12½–14	D	C	B	B	*a*	*a*					
	14–20	E	D	C	B	A	*a*					
	20–30	E	D	D	C	B	A					
	30–42	F	E	D	C	B	A					
	42–60	F	E	E	D	C	B					
	60 up	F	F	E	D	C	B					
12½–15	15–20	E	D	C	C	B	A	*a*				
	20–30	F	E	D	C	B	A	A				
	30–42	F	E	E	D	C	B	A				
	42–60	F	F	E	E	D/C	B	A				
	60–90	G	F	F	E	D	C	B/A				
	90 up	G	F	F	E	E	D/C	B				

Room Indices printed in Italics.—In certain instances where normal spacing indicates the use of one, or at the most two fittings, the average illumination (obtained by the lumen method of design) may not be a good indication of the usefulness of the lighting owing to the comparatively large variation of intensity at different points on the working plane. In such instances it is preferable to use the point-by-point method of design when direct or semi-direct fittings are to be employed.

This also applies to general diffusing fittings in rooms with dark decorations, but not to fittings giving a greater proportion of upward light.

TABLE III.—ROOM INDEX—(*cont*).

		Ceiling Height above Working Plane—Feet.										
For Semi-Indirect and Indirect Fittings.		6 and 6½	7 to 8½	9 to 10½	11 to 13½	14 to 17	18 to 21	22 to 27	28 to 33	34 to 47	—	—
		Mounting Height above Working Plane—Feet.										
For Direct, Semi-Direct and General Diffusing Fittings.		4 and 4½	5 and 5½	6 and 6½	7 to 8½	9 to 10½	11 to 13½	14 to 17	18 to 21	22 to 27	28 to 33	34 to 47
Room Width (feet).	Room Length (feet).	ROOM INDEX.										
15–18	18–20	F	E	D	C	B	A	A				
	20–30	F	E	E	D	C	B	A				
	30–42	G	F	E	D	C	C	B/A	A			
	42–60	G	F	F	E	D	D/C	B	A			
	60–110	G	F	F	E	D	D	C/B	A			
	110 up	H	G	F	F	E	D	C	B			
18–22	22–30	G	F	E	D	C	B	A	A			
	30–42	G	F	F	E	D	C	B	A	A		
	42–60	G	G	F	F	E	D/C	B	A	A		
	60–90	H	G	F	F	E	D	C	B/A	A		
	90–140	H	G	G	F	E	E/D	C	B	B		
	140 up	H	G	G	F	E	E	D/C	C	B		
22–26	26–30	G	F	F	E	D	C	B	A	A		
	30–42	H	G	F	E	D	D/C	B	A	A		
	42–60	H	G	G	F	E	D	C	B	A	A	
	60–90	H	G	G	F	E	E/D	C	B	A	A	A
	90–140	H	H	G	F	F	E	D	C	B	A	A
	140 up	H	H	G	F	F	E	D	C	B	B	A
26–30	30–42	H	G	G	F	E	D	C	B	A	A	
	42–60	H	H	G	G	F/E	E	D/C	C	B	A	
	60–90	I	H	H	G	F	E	D	C	B	A	A
	90–140	I	H	H	G	F	F	E	E/D	C	B	A
	140–180	I	H	H	G	F	F	E	D	C	B	B/A
	180 up	I	H	H	G	F	F	E	D	C	C/B	A
30–36	36–42	I	H	G	F	E	E/D	C	B	B	A	
	42–60	I	H	H	G	F	E	D	C	B	A	A
	60–90	J	I/H	H	H/G	F	F	E	D/C	C/B	A	A
	90–140	J	I	H	H	G	F	E	D	C	B	A
	140–200	J	I	H	H	G	F	E	E	D	C	B
	200 up	J	I	H	H	G	F	E	F	D	C	C/B

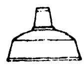
TABLE III.—ROOM INDEX—(*cont*).

Ceiling Height above Working Plane—Feet.												
For Semi-Indirect and Indirect Fittings.		6 and 6½	7 to 8½	9 to 10½	11 to 13½	14 to 17	18 to 21	22 to 27	28 to 33	34 to 47	—	—
Mounting Height above Working Plane—Feet.												
For Direct, Semi-Direct and General Diffus- Fittings.		4 and 4½	5 and 5½	6 and 6½	7 to 8½	9 to 10½	11 to 13½	14 to 17	18 to 21	22 to 27	28 to 33	34 to 47
Room Width (feet)	Room Length (feet).	ROOM INDEX.										
36–44	44–60	J	I	H	H/G	F	E	D	C	B	B	A
	60–90	J	I	I	H	G	F	E	D	C	B	A
	90–140	J	I	I	H	G	G	F	E	D	C/B	A
	140–200	J	J	I	H	G	G	F	E	D	C	B
	200 up	J	J	I	H	G	G	F	E	E	D/C	B
44–55	55–60	J	J	I	H	G	F	E	D	C	B	A
	60–90	J	J	I	H	H	G	F/E	E	D	C/B	A
	90–140	J	J	J/I	H	H	G	F	E	E	D/C	B
	140–200	J	J	J	I/H	H	G	F	F	E	D/C	B
	200 up	J	J	J	I/H	H	G	F	F	E	D	C
55–70	70–90	J	J	J	I	H	G	F	E	D	C	B
	90–140	J	J	J	I	H	H	G	F	E	D	C
	140–200	J	J	J	I	H	H	G	F	F	E/D	C
	200 up	J	J	J	I	H	H	G	F	F	E	D/C
70–90	90–140	J	J	J	J	I	H	G	F	E	E	D/C
	140–200	J	J	J	J	I	I	H	G	F	E	D
	200 up	J	J	J	J	I	I	H	G	F	E	E/D
90–115	115–140	J	J	J	J	J	I	H	G	G/F	E	D
	140–200	J	J	J	J	J	I	H	G	G/F	E	D
	200 up	J	J	J	J	J	I	H	H	G	F	E

In rooms more than 115 ft. wide, the Coefficient of Utilisation will continue to increase gradually, and in the case of a Direct lighting installation will eventually equal the Luminous Efficiency of the fitting (Column 2 of Table IV) in rooms of infinite size.

TABLE IV.—COEFFICIENT OF UTILISATION.

"MAINLY DIRECT" FITTINGS

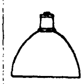
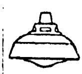
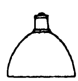
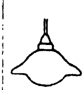
Lighting Equipment.	Typical Outline and Efficiency	Ceiling	70%			50%			30%	
		Walls	50%	30%	10%	50%	30%	10%	30%	10%
		Room Index.	Coefficient of Utilisation.							
DIRECT (A) Standard Dispersive Reflector for Gasfilled Lamps and for 125W and 80W Mercury Lamps (Type MB/V) and Sodium Lamps (Type SO).	 	A	·33	·28	·24	·32	·28	·24	·27	·24
		B	·41	36	·33	·40	·36	·33	36	·33
		C	·44	·40	·38	·43	·40	·37	·40	·37
		D	·47	·44	·42	·46	·43	·41	·43	·41
		E	·50	·47	·45	·49	·46	·44	·46	·44
		F	·53	·50	·48	·52	·49	·47	·49	·47
		G	·56	·53	·51	·55	·53	·50	·53	·50
		H	·59	·56	·54	·58	·56	·53	·56	·53
Also for Open-top and Closed-top Enamelled Troughs for Fluorescent Lamps* (Type MCF/U).		I	·62	59	·57	·61	·59	·56	·59	·56
	Down 70%	J	·64	·62	·60	·63	·61	·59	·61	·59
DIRECT (A) Dispersive Reflector for 400W and 250W Mercury Lamps (Type MA/V). Concentrating (Enamelled) Reflector for 80W. and 125W. (MB/V) Mercury, Sodium (SO/H) and Gasfilled Lamps.		A	·30	·25	·22	·29	·25	·22	·25	·22
		B	·36	·33	·31	·36	·32	·30	·32	·30
		C	·39	·36	·34	·39	·35	·33	·35	·33
		D	·42	·39	·37	·42	·38	·36	·38	·36
		E	·45	·42	·40	·45	·41	·39	·41	·39
		F	·48	·45	·43	·48	·44	·42	·44	·42
		G	·51	·48	·46	·51	·47	·45	·47	·45
		H	·54	·51	·49	·54	·50	·48	·50	·48
		I	·56	·54	·52	·56	·53	·51	·53	·51
	Down 60%	J	·58	·56	·54	·57	·55	·53	·54	·53

COVER GLASSES

When a front cover of glass is fitted to reduce the brightness of any of the above reflectors, or to exclude dirt, an additional allowance must be made for light absorption, depending on the shape, type and quality of glass used, the last being especially critical with reflectors for discharge lamps. As a general guide, the above figures should be multiplied by 0·7 for flashed opal, 0·75 for frosted, and 0·9 for clear glasses, but it is strongly recommended that definite figures be obtained from the manufacturers of the equipment in question.

* *Multi-lamp fittings.*—In general, fittings housing 2 and 3 lamps will have a Coefficient of Utilisation approximately 0·9 and 0·85 of the above figures respectively.

TABLE IV.—COEFFICIENT OF UTILISATION—(*cont*).

Lighting Equipment.	Typical Outline and Efficiency	Reflection Factor.								
		Ceiling	70%			50%			30%	
		Walls	50%	30%	10%	50%	30%	10%	30%	10%
		Room Index.	Coefficient of Utilisation.							
DIRECT (A) Dispersive Reflectors for 400W. (Isothermal) 125W. and 80W. Fluorescent Mercury Lamps (Types MAF/V and MBF/V). Also Concentrating Enamelled Reflector for 400W. and 250W. (MA/V) Mercury Lamps.	 Down 55%	A	·26	·22	·19	·25	·22	·19	·22	·19
		B	·33	·29	·26	·32	·29	·26	·29	·26
		C	·36	·32	·29	·35	·32	·29	·32	·29
		D	·39	·35	·32	38	·35	·32	·35	·32
		E	·42	·38	·35	·40	·38	·35	·38	·35
		F	·45	·41	·38	·42	·41	·38	·41	·38
		G	·47	·44	·41	·44	43	·41	·43	·41
		H	·49	·46	·44	·46	·45	·43	·45	·43
		I	·51	·48	·46	·48	·47	·45	·47	·45
		J	·53	·50	·48	·50	·49	·47	·49	·47
DIRECT (B) Reflectors with Enclosed Globe such as Industrial Diffusing Fitting for Gasfilled Lamps or for Blended Light from both Mercury and Gasfilled Lamps. Also Open Reflectors for Mercury/Tungsten Lamps (Type MAT).	 Down 50%  Down 60%	A	·28	·25	·22	·26	·23	·20	·22	·20
		B	·34	·30	·27	·31	·29	·27	·28	·26
		C	·39	·34	·32	·36	·34	·32	·32	·31
		D	·43	·38	·36	·40	·38	·36	·36	·35
		E	·47	·42	·40	·44	·42	·40	·40	·39
		F	·50	·45	·43	·47	·45	·43	·43	·42
		G	·53	·48	·46	·50	·48	·46	·46	·45
		H	·55	·51	·49	·53	·51	·49	·49	·48
		I	·57	·53	·51	·54	·52	·50	·50	·49
		J	·58	·54	·52	·55	·53	·51	·51	·50
SEMI-DIRECT (A). Enclosed Fittings with major light flux downwards. (Figures based on 55 per cent. of light output from lamp downward and 15 per cent. upward.)	 Up 15% Down 55%	A	·27	·24	·22	·26	·23	·21	·22	·20
		B	·31	·28	·26	·30	·27	·25	·26	·24
		C	·35	·32	·30	·34	·31	·28	·30	·28
		D	·39	·36	·34	·38	·34	·31	·33	·31
		E	·42	·39	·37	·41	·37	·34	·36	·34
		F	·45	·42	·40	·44	·40	·37	·39	·37
		G	·48	·45	·43	·47	·43	·40	·41	·39
		H	·51	·48	·46	·49	·45	·42	·43	·41
		I	·54	·51	·48	·51	·47	·44	·45	·43
		J	·56	·53	·50	·53	·49	·46	·47	·45

SEMI-DIRECT (B). Luminous beams, etc. The Coefficient of Utilisation of lighting features in the form of luminous beams varies greatly with the details of construction and the method of mounting. In the case of a luminous beam glazed with untinted flashed opal glass on three sides and fixed to the ceiling the coefficients will average approximately 95 per cent. of those for Semi-Direct (A) above.

TABLE IV.—COEFFICIENT OF UTILISATION—(*cont*).

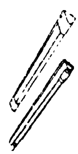
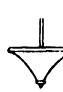
Lighting Equipment.	Typical Outline and Efficiency		Reflection Factor.							
		Ceiling	70%			50%			30%	
		Walls	50%	30%	10%	50%	30%	10%	30%	10%
		Room Index.	Coefficient of Utilisation.							
SEMI-DIRECT (C). Batten type and V-Channel type fittings for fluorescent lamps (Type MCF/U)		A	·30	·25	·21	·27	·22	Not generally to be recommended with dark surroundings on account of the likelihood of discomfort glare.		
		B	·38	·32	·28	·34	·29			
		C	·42	·36	·33	·39	·34			
		D	·46	·41	·37	·43	·38			
		E	·50	·45	·40	·47	·42			
		F	·55	·50	·45	·51	·45			
		G	·60	·54	·49	·54	·49			
		H	·63	·58	·52	·57	·52			
	Up 25%	I	·67	·62	·57	·59	·54			
	Down 60%	J	·70	·65	·60	·61	·56			
"GENERAL DIFFUSING" FITTINGS										
GENERAL DIFFUSING. Enclosed Diffusing Fittings of Flashed Opal Glass.		A	·26	·21	·18	·22	·18	·16	·17	·15
		B	·31	·26	·22	·27	·23	·20	·20	·18
		C	·35	·30	·26	·31	·27	·24	·23	·21
		D	·38	·33	·30	·34	·30	·27	·26	·24
		E	·41	·37	·33	·37	·33	·30	·29	·27
		F	·44	·40	·36	·40	·36	·33	·32	·30
		G	·47	·43	·39	·43	·39	·36	·35	·33
	Up 35%	H	·50	·47	·42	·45	·41	·38	·37	·36
	Down	I	·53	·50	·45	·47	·43	·40	·39	·38
	45%	J	·56	·53	·48	·49	·45	·42	·41	·40
"MAINLY INDIRECT" FITTINGS										
SEMI-INDIRECT. Pendant Translucent Bowls and enclosed fittings giving some 75 per cent. of their light output upwards.		A	·18	·14	·12	·14	·12	These coefficients are based on a 75 per cent. luminous efficiency of fitting, and will need adjustment in proportion when using fittings of higher or lower efficiency. These types of fittings are only suitable for use in rooms with light coloured decorations.		
		B	·21	·18	·16	·17	·15			
		C	·24	·21	·19	·20	·18			
		D	·27	·24	·22	·22	·20			
		E	·30	·27	·25	·24	·22			
		F	·33	·30	·28	·26	·24			
		G	·36	·33	·31	·28	·26			
		H	·39	·36	·34	·30	·28			
	Up 55%	I	·42	·39	·37	·32	·30			
	Down 20%	J	·45	·42	·40	·34	·32			
INDIRECT (A). Pendant Fittings and Troughs giving all their light output in an upward direction but without sharp directional control of light.		A	·15	·12	·10	·10	·08			
		B	·18	·16	·13	·12	·10			
		C	·21	·19	·16	·14	·12			
		D	·24	·22	·19	·16	·14			
		E	·27	·25	·22	·18	·16			
		F	·30	·28	·25	20	·18			
		G	·33	·30	·27	·22	·20			
		H	·36	·32	·29	·23	·21			
		I	·38	·34	·31	·24	·22			
	Up 75%	J	·40	·36	·33	·25	·23			

TABLE IV.—COEFFICIENT OF UTILISATION—(*cont*).

Lighting Equipment.	Typical Outline and Efficiency	Ceiling	70%			50%			30%	
		Walls	50%	30%	10%	50%	30%	10%	30%	10%
		Room Index.	Coefficient of Utilisation.							
INDIRECT (B). Cornices, Recessed Coves and Coffers, giving all their light above the horizontal.		A	·08	·06	·05	·05	·04	These coefficients are based on a 40 per cent. luminous efficiency of the feature, but details of construction may vary this figure very considerably.		
		B	·10	·08	·07	·07	·06			
		C	·12	·10	·09	·08	·07			
		D	·13	·11	·10	·09	·08			
		E	·15	·13	·11	·10	·09			
		F	·17	·15	·13	·11	·10			
		G	·18	·16	·15	·12	·11			
		H	·19	·18	·16	·13	·12			
		I	·21	·20	·18	·14	·13			
		J	·22	·21	·20	·15	·14			

TABLE V.—NOMINAL LUMEN RATINGS, ETC., OF GASFILLED LAMPS.

Watts.	Type.	Dimensions. Length (mm.)	Dimensions. Diameter (mm.)	Light Centre Length (mm.)	Standard Cap.	Minimum average Lumens throughout Life. At 110v. Single Coil.	At 230v. Single Coil.	At 230v. Coiled Coil.
15	Pearl	92·5	55	65	B.C.	133	113	—
25	"	100	60	70	"	228	206	—
40	"	110	60	80	"	449	330	389
60	"	117·5	65	85	"	759	584	665
75	"	125	70	90	"	1,000	785	883
100	"	137·5	75	100	"	1,400	1,160	1,270
150	"	160	80	120	"	2,230	1,970	—
200	Clear	178	90	133	E.S.	3,090	2,725	—
300	"	233	110	178	G.E.S.	4,950	4,430	—
500	"	267	130	202	"	8,950	7,930	—
750	"	300	150	225	"	14,270	12,740	—
1,000	"	300	150	225	"	19,640	17,800	—
1,500	"	335	170	250	"	30,220	28,380	—

Electric Discharge Lamps.

One of the latest developments in the manufacture of electric lamps is the introduction of the high pressure mercury vapour electric discharge lamp, which has no filament, as in the ordinary gas-filled lamp, but gives light by means of an electric discharge in an inner tube surrounded by an outer jacket. These lamps give about 2½ times as much light as the same size of filament lamp, and are therefore highly economical in those situations where their particular colour characteristics are desirable or not a disadvantage.

The spectrum of these lamps is not continuous, being a line spectrum composed of green, yellow, some blue, and a little red. The light appears to be greeny-blue, and is particularly suited for all inspection purposes (except those involving discrimination of red colours) since scratches, cracks and other flaws show up much better than under the light of filament lamps. Stroboscopic effects are usually negligible except in the case of very high speed machinery, and they can then be

mitigated by connecting adjacent lamps to different phases of supply. The penetrating and resolving power of the light from these lamps, combined with their low running cost, has resulted in rendering any local lighting unnecessary in many situations, such as foundries, machine shops, stores, etc.

One type of mercury lamp is fitted with a fluorescent outer bulb, which adds some orange-red light and thus slightly improves the colour-rendering properties of the lamp. This modification is effected with little loss of efficiency, and allows some industrial operations to be carried out which could not be done under ordinary mercury lamps.

In another type a good colour of light is obtained by combining a mercury discharge and a tungsten filament in the same outer bulb. The filament exerts the necessary control over the current, and auxiliary apparatus is therefore not required.

TABLE VI.—NOMINAL LUMEN RATINGS, ETC., OF MERCURY VAPOUR DISCHARGE LAMPS.

Type and Wattage.	Bulb Shape.	Dimensions, Length.	Dia. (mm.)	Power Loss in Control Gear (Watts).	Average Lumens throughout Life.	Cap.
Mercury (MB/V) 80	Pear	160	80	10	2,480	3-pin B.C.
" " 125	"	178	90	12	4,250	" "
" (MA/V) 250	Tubular	290	48	15	8,000	G.E.S.
" " 400	"	330	48	20	14,400	"
Flu. Mercury (MBF/V) 80	Pear	178	110	10	2,240	3-pin B.C.
" " " 125	"	233	130	12	3,750	G.E.S.
" " (MAF/V)400	Isothermal	335	165	20	12,800	"
Tungsten Mercury (MBT/U)160	Pear	178	90	—	2,400	E.S.
" " (MBT/V)200	"	178	90	—	3,400	"
" " (MBT/U)250	"	233	110	—	4,250	G.E.S.
" " (MAT) 300	Tubular	285	85	—	5,400	"
" " (MAT) 500	"	380	100	—	10,500	"

Sodium Lamps.

Sodium vapour electric discharge lamps are also used to a considerable extent where their monochromatic yellow light is suitable, particularly in factories, streets and for flood-lighting. Nominal lumen ratings of these lamps are shown below.

TABLE VII.—NOMINAL LUMEN RATINGS, ETC., OF SODIUM VAPOUR ELECTRIC DISCHARGE LAMPS.

Watts.	Average Lumens throughout Life.	Cap.
45	2,200	B.C.
60	3,420	B.C.
85	5,350	B.C.
140	8,960	B.C.

Tubular Fluorescent Lamps.

In these lamps almost the entire light output is obtained from the fluorescence of a powder deposited on the inner surface of the tube, in which a discharge takes place through a low pressure of mercury vapour.

Three colours of light are available. (1) *Daylight*, giving a light apparently similar to natural daylight in general, though weak in deep red light which makes colour rendering of red tones rather unsatisfactory. (2) *Warm White*, giving a pinker light than the above, sometimes preferred for applications where the lighting is required to be restful and decorative. This type has the same efficiency as the Daylight. (3) *Natural*, in which a better balance of colours than in the other two types is obtained, though with a slight reduction of efficiency. This type renders colours in general more truly and, though only recently introduced, is likely to become the most widely used.

The total heat generated by fluorescent lamps is about one-third of that generated by filament lamps giving equal light output, and the radiated heat about one-fifth as much. The surface brightness of the lamps is low, and glare is thus reduced to a minimum, though it is usually still desirable to screen the lamps and/or ensure that the background against which they are seen is not dark, particularly at high illumination levels.

In common with all other discharge lamps the fluorescent lamp must be connected in series with a current-limiting device (choke) and in addition special arrangements are necessary for starting, since the electrodes require to be pre-heated before the discharge takes place. This is usually effected by a starter switch which allows current to pass through the electrodes for a few moments, and then automatically breaks the circuit, inducing in the choke a voltage surge which starts the lamp. Starters are of two types, thermal and glow, and in modern installations either may be used without alteration to the circuit.

An alternative device, available at the time of writing (December 1948), for the 80-watt lamp only, is an instant (or quick) -start unit. This is a high reactance auto-transformer connected across the lamp, with a low-voltage winding connected to each electrode. When switched on, the discharge will strike automatically as soon as the electrodes reach operating temperature (*i.e.* in a fraction of a second). This arrangement requires the usual series choke to be retained, and an earthed metal strip must make contact with the lamp exterior along its whole length.

Lamps more than 2 ft. in length require one set of control gear each. Lamps of 2 ft. length and less may be run two in series with a single choke on 200/250 volt mains ; in this case lamps must be of the same wattage rating.

TABLE VIII.—NOMINAL LUMEN RATINGS, ETC., OF TUBULAR FLUORESCENT LAMPS.

Lamp Size Watts.	Voltage Range.	Approximate Choke Consumption (Watts).	Brightness Average through life (Candles/in.²).	Overall Length (Ft.)	Diameter (Ins.)	Caps Each End.
15	100–130	12*	$2\frac{3}{4}$	$1\frac{1}{2}$‡	1	Bi-pin
20	100–130	12*	2	2‡	$1\frac{1}{2}$	,,
30	200–250	12	4	3‡	1	,,
40	100–130	20*	$3\frac{3}{4}$	2‡	$1\frac{1}{2}$	,,
40	200–250	12	$2\frac{3}{4}$	4‡	$1\frac{1}{2}$	,,
80	200–250	20†	$3\frac{3}{4}$	5	$1\frac{1}{2}$	B.C.

Lamp Size Watts.	Light Output—Lumens.								
	Daylight.			Warm White.			Natural.		
	At 100 Hrs.	Average through Life.	Final.	At 100 Hrs.	Average through Life.	Final.	At 100 Hrs.	Average through Life.	Final.
15	—	—	—	510	420	375	465	375	345
20	—	—	—	760	620	540	680	560	480
30	—	—	—	1,380	1,200	1,050	1,260	1,080	960
40 (2 ft.)	—	—	—	1,320	1,160	1,000	1,200	1,040	920
40 (4 ft.)	2,000	1,720	1,440	2,000	1,720	1,440	1,800	1,520	1 280
80	3,600	3,040	2,560	3,600	3,040	2,560	3,280	2,720	2,240

Installation Details.—When used for general lighting at heights greater than 4 ft. 6 ins. above the working plane, enamelled trough reflectors for fluorescent lamps perform in a similar manner to Direct (A) reflectors for gas-filled lamps, so far as distribution of light and efficiency is concerned, and the coefficient of utilisation for Direct (A) fittings may therefore be used.

Since there is an appreciable drop in light output from fluorescent lamps during the first 150/200 hours of life, the initial efficiency of the lamp should be ignored for design purposes.

* With two lamps in series on 200/250 volts.
† Elongated type choke. For block type the loss is approximately 10 watts.
‡ Including standard bi-pin holders.

Instead, the 'average lumens throughout life' should be taken as the output, and a maintenance factor (usually 0·8) should be applied, as for any other class of lamp.

Lamp life will in general be about three times that of gas-filled lamps, but is dependent to some extent on the frequency of switching. With the lamps kept burning for prolonged periods the life may be longer, or shorter when they are only used for brief periods.

Tubular fluorescent lamps lend themselves particularly well to localised lighting of particular areas such as lines of machines or benches. The average illumination to be expected in service from an 80-watt lamp may be gauged from the tables below. (Deduct 10 per cent. for Natura colour lamps):—

SINGLE FITTING—AREA OVER WHICH DIVERSITY OF ILLUMINATION DOES NOT EXCEED 2 : 1.

Height above Working Plane.	Approximate Area.	Foot-Candles. Max.	Min.	Mean.
2 ft.	Ellipse 5 ft. × 3 ft.	90	45	70
3 ft.	Ellipse 6 ft. × 5 ft.	45	25	35
4 ft.	Circle 6 ft. × 6 ft.	30	15	22
5 ft.	Circle 7 ft. × 7 ft.	20	10	15

SINGLE ROW OF FITTINGS—ILLUMINATION VALUES ON A BENCH 4 FT. WIDE

Height above Working Plane.	Spacing (Centres).	Foot-Candles. Max.	Min.	Mean.	Distance between Rows.*
3 ft.	6 ft.	54	24	40	12 ft.
4 ft.	6 ft.	36	24	30	14 ft.
4 ft.	8 ft.	32	18	25	13 ft.
5 ft.	6 ft.	26	22	24	15 ft.
5 ft.	8 ft.	22	16	20	14 ft.
5 ft.	10 ft.	20	12	16	13 ft.

A continuous line of single-lamp 80-watt fittings mounted centrally 2 ft. above a bench 2 ft. wide will give an average illumination of the order of 80 foot-candles.

Architectural Lighting.

Except where artificial light is used purely for utilitarian purposes, the modern tendency is to spread the light over large surfaces instead of using small, brilliant light sources. This effect is commonly achieved by fitting the lamps behind sheets of opal or otherwise obscured glass, the whole fitting being built-in to the existing structure. In order to obtain an even lighting effect over the whole surface of the glass, the lamp spacing—distance behind glass ratio for pearl lamps and various types of glass should be as shown in the table below. With tubular fluorescent lamps the spacing may be approximately 50 per cent. greater.

LAMP SPACING IN INCHES.

Glass.	D = 4	6	8	10	12
Frosted stippolyte .	2½	3	3½	4½	5
Acid etched .	2	3	4	5	6½
Flashed opal . .	5	7	9	11½	13½

D = Distance of lamp centres behind glass.

*** *Note.*—This column gives the maximum distance apart at which it is safe to mount rows of fittings, in order to comply with the statutory minimum requirements of 6 foot-candles in working areas. If this distance is exceeded it may be necessary to install additional general lighting.**

Street Lighting.

Modern street lighting practice is based on the recommendations contained in the Report issued by the Ministry of Transport Departmental Committee on Street Lighting in 1937. This aims, in general, at providing an evenly bright road surface against which obstructions are seen as dark objects in silhouette. Provision of even road brightness does not necessarily require even illumination, in fact, the variation of illumination from point to point may be very wide.

The siting of street lighting posts is perhaps more important than any other consideration. A lantern will provide a T-shaped patch of brightness on a part of the road surface situated between the lantern post and the observer, and the aim is generally to locate the posts in such a position relative to each other that the individual bright patches merge to form a uniformly bright field. Special care is required to ensure that entry to and exit from roundabouts, crossings and junctions are clearly visible whatever the direction of movement; on bends lanterns are generally located on the outside of the bend and closely spaced.

The fuel shortage has emphasised the advantages of discharge lamps for non-residential roads, and most new or improved installations for such locations use either mercury or sodium discharge lamps. Recently a considerable amount of tubular fluorescent lighting has also been installed in the civic centres or main shopping streets of towns, and it is already clear that this type of lighting will be used to an increasing extent in future. Partly on account of the limited light output obtained from the largest fluorescent lamp (80 watts) multi-lamp lanterns are generally used on a fairly close spacing, resulting in comparatively high capital costs; but the pleasant and comfortable nature of the light obtained is often considered to justify its use in the situations mentioned. In addition, the use of multi-lamp lanterns permits some lamps in each lantern to be extinguished at, say, midnight for purposes of economy without upsetting the distribution of light on the road surface.

British Standard Specifications Dealing with Electric Lamps or Lighting.

No. 33—Carbon Filament Electric Lamps.
„ 97—Watertight Fittings for Incandescent Electric Lamps.
„ 98—Edison-type screw lamp caps and lampholders, dimensions of.
„ 161—Tungsten Filament General Service Electric Lamps.
„ 229—Flameproof enclosure of electrical apparatus for power and lighting plant.
„ 230—Portable Photometers (Visual Type).
„ 232—Vitreous Enamelled Steel Reflectors for Electric Lighting. Open Dispersive Type.
„ 233—Terms used in Illumination and Photometry, Glossary of.
„ 307—Street Lighting.
„ 324—Translucent (Diffusing) Glassware Fittings for Interior Lighting.
„ 364 Neck and Flange Dimensions of Illumination Glassware and Carriers.
„ 495—Fittings for Double-capped Tubular Lamps.
„ 535—Miners' Lamp Bulbs.
„ 555—Tungsten Filament Electric Lamps (other than General Service Lamps).
„ 667—Photoelectric Type Portable Photometers.
„ 710—An Electric Study and Reading Table Lamp.
„ 793—Tungsten Filament Electric Lamps and Fittings, with Partial Daylight Colour Corrections.
„ 867—Traction Lamps (Series burning).
„ 889—Flameproof Electric Light Fittings for use in coal mines and other places where inflammable gas or vapour may be present in the surrounding atmosphere.
„ 950—Artificial Daylight Fittings for Colour Matching.
„ 1270—Electric Discharge Lamps for General Purposes.

SECTION XXXIV

PART IV

FIRE EXTINCTION AND PREVENTION.

(Contributed by Hubert B. Graham, M.I.Mech.E.)

First-Aid Precautions.

First-aid fire precautions include the provision of portable extincteurs, stirrup pumps, water in buckets, sand in bins and bicarbonate of soda and dry sand in sealed containers. The three last named are suitable for use only in cases of very small fires on bench or floor.

A standard Stirrup Pump (BS/ARP 33) is capable of dealing with fires at a higher level and of a more serious character. It requires a team of three persons, however, to maintain it in operation for the discharge of water at the rate of 1½ gallons per minute, to a distance of 25 ft.

The comparative performance of a portable extincteur expelling water, or an alkaline solution dilute, operated by one person only, is 2 gallons per minute, thus, with a team of three persons operating three extincteurs, the rate of discharge would be 4 times that of a stirrup pump, which render portable extincteurs the most effective first-aid means of extinguishing carbonaceous fires.

For *special* fire risks, different types of portable extincteurs are available and mention of these is made hereunder.

PORTABLE FIRE EXTINCTEURS.

The British Standards Institution (B.S.I.) has defined a portable fire extinguisher (alt. extincteur) as a first-aid fire appliance that can be carried by hand.

B.S.I. Specifications (ref. BS) have been issued for different types of extincteurs designed to deal with different fire risks, namely :—

(i) BS 138/1922	Water and carbon dioxide (CO_2).
(ii) BS 138/1935	Acid alkali (alt. soda-acid).
(iii) BS 740/1937, Part 1 . . .	Foaming solution.
(iv) BS 740/1937, Part 2 . . .	Carbon tetrachloride ($C.Cl_4$).

Other types for which B.S.I. Specifications have not yet been issued are :—

(v) Methyl bromide (CH_3Br) and nitrogen.
(vi) Carbon dioxide (CO_2) direct gas jet.*
(vii) Carbon dioxide (CO_2) and bicarbonate of soda.*

The specifications for extincteurs (i), (ii), (iii) and (iv) detail the essential requirements according to British practice. They were drafted in consultation with the Admiralty, War Office, Air Ministry, Home Office and Ministry of Works and Buildings, and subsequently adopted by these Government Departments in place of departmental specifications formerly issued.

The Fire Offices Committee (F.O.C.) also assisted in the drafting of these specifications and subsequently adopted them in place of earlier F.O.C. Specifications. Strict compliance with F.O.C. Rules and Regulations is essential in the equipment of industrial buildings in order to qualify them for rebates of premiums on (Tariff) Fire Insurance Policies.

Other bodies represented on the B.S. Committee included the National Fire Brigades Association, Professional Fire Brigades Association, Institution of Fire Engineers, Institution of Chemical Engineers, Institution of Mechanical Engineers, Institution of Gas Engineers, Iron and Steel Institute, Association of British Chemical Manufacturers and the Fire Extinguisher Trades Association.

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* For the composition, strength and test-pressure of CO_2 steel cylinders used for (vi) and (vii), see BS 401/1931-37-38.

The Board of Trade reserves the right to approve extincteurs proposed for installation on marine passenger carrying vessels.

The essential requirements to be noted in the aforesaid specifications are :—regulation markings, standard tests, capacity and expansion space, safe-load indicator, nature of dynamic or chemical charge, method of actuation, velocity and throw of jet, materials, dimensions and construction.

The regulation markings to be indelibly printed or etched on the exterior of a liquid type extincteur operated by gas or air pressure in which the working pressure *exceeds* 50 lbs. per sq. in. are : (*a*) A declaration that it has been 'tested to 350 lbs. per sq in.' implying that it has been tested by internal hydraulic pressure to 350 lbs. per sq. in. for a continuous period of 2½ minutes without leakage or visible distortion. (*b*) Its liquid discharge capacity in gallons. (*c*) The year of its manufacture ; and (*d*) the name and address of its manufacturer or responsible vendor. The markings of a liquid type extincteur operated by gas or air in which the working pressure does *not* exceed 50 lbs. per sq. in. shall be as (*b*), (*c*) and (*d*).

The capacity of types (i) and (ii) shall not be less than 1 gallon or more than 3 gallons, type (iii) shall not be less than 1 gallon or more than 2 gallons, type (iv) shall not be less than 1½ pints or more than 2 gallons.

The expansion space in a gas pressure type liquid extincteur shall be such that when fully charged with liquid at a temperature of 100° F., the outlets of the liquid container temporarily closed and the relative maximum gas charge exerting pressure therein, such pressure shall not exceed 200 lbs. per sq. in. in extincteurs in which the working pressure *exceeds* 50 lbs. per sq. in. or shall *not* exceed 100 lbs. per sq. in. in extincteurs in which the working pressure does *not* exceed 50 lbs. per sq. in. On all such extincteurs refillable by an operator after use, the safe-load (*i.e.* liquid filling level) shall be indicated on the exterior of the liquid container.

In the seven types named there are fundamental differences in their effect on different classes of fires resulting from differences in the extinguishing media and differences in the dynamic or chemical charges, but different methods adopted for actuating a given type, as, for instance, type (ii), do not affect the working efficiency of that type but only the manner of operating it.

Type (i) water and CO_2 extincteur discharges chilled water at high pressure exerted by the expansion of CO_2 from liquefied formation to atmospheric formation on release from a small disc-sealed cylinder (alt. cartridge or capsule). In an extincteur of this type of 2 gallons capacity, the dynamic charge is 65 grammes of liquefied CO_2, which, having been compressed to 60 atmospheres in a cartridge, 9 ins. × 1 in. o/d, 73 c.c. content, will, on release, expand to 33·850 c.c. at atmsopheric formation at 15° C., *i.e.* 450 times its original volume when liquefied and equal to 2¾ times the capacity of the water container in which the expansion occurs. Hence the evaporative chilling effect on the water and its high velocity on discharge.

This type is actuated by striking a knob on the headcap, the knob being connected to a piercing pin which punctures the copper sealing disc of the CO_2 cartridge.

Type (ii) acid alkali extincteurs discharge a dilute alkaline solution resulting from the chemical action of an acid in conjunction with a carbonate and/or bi-carbonate solution. The proportion of acid to alkali in the chemical charge varies to some extent in the charges for different makes of extincteurs, but a standard recognised for a 2 gallon capacity extincteur is 2½ liquid ounces of sulphuric acid (sp. gr. 1·84) to 1 lb. 2 oz. of bi-carbonate of soda.

This type may be operated (*a*) by breaking an hermetically sealed bottle containing an acid and thereby allowing the acid to come into contact with the carbonate or bi-carbonate solution, or (*b*) by turning over the extincteur or the acid bottle therein and thereby spilling acid into the carbonate or bi-carbonate solution. The operation (*a*) may be performed by striking a knob on the extinguisher or by a blow on its side.

This type, while generally known as a chemical extincteur, is so only in the sense of producing by chemical reaction the pressure required to expel its liquid content—not that the chemicals employed for that purpose have any virtue as fire extinguishing media, for in a properly proportioned charge the reaction of the alkali to the acid neutralises the latter and renders the former a weak dilute.

The effect of types (i) and (ii) discharged on materials in combustion is (*a*) to cool them so that inflammable gases given off by disintegration may be arrested, (*b*) to damp them so that vapour may arise to assist their extinguishment, and (*c*) by a forcible jet of water or dilute alkaline solution to beat off flames repeatedly until they no longer re-light.

Types (iii) to (vii) extincteurs act on a different principle. They have no appreciable cooling or damping effect nor do they rely on a forcible jet but on fires on inflammable liquids and of electrical origin, they smother combustion by the exclusion of oxygen therefrom.

Type (iii) foam extincteurs discharge a closely knit stream of adhesive bubbles of gas formed by chemical action of an acid salt, stored in an inner receptacle, in conjunction with a carbonate or bi-carbonate solution in combination with a stabiliser, stored in the outer container of the extincteur. Until the year 1942 it had been common practice to place a solution of sulphate of aluminium in the inner receptacle and in the outer container a solution of sodium bicarbonate, saponin and an extract of vegetable syrup. The two latter act as a stabiliser, the function of which is to affect the surface tension of the bubbles of gas resulting from the conjunction of the two solutions so as to provide a tough adhesive wall structure for each bubble thereby permitting

expansion of the gas therein to its normal limit at any temperature without causing the bubbles to burst as they otherwise would do.

The expansion of these bubbles creates a pressure in the extincteur whereby a stream of them is propelled therefrom a distance of upwards of 20 feet in a specified period of time, which, in the case of a 1-gallon extincteur, is from 30 to 90 seconds, according to the climatic condition then prevailing.

The average rate of expansion of the solution with foam is 8 to 1 at a temperature of 50° F. These extincteurs should not be exposed to frost and preferably placed where the temperature does not fall below 50° F.

In 1942, as a measure to limit imports during a state of war, the B.O.T. directed the F.E.T.A. and Central Research Laboratory to combine in research for a substitute for the stabiliser hitherto used as aforementioned. A successful issue resulted in the summer of 1942, but the constitution of this new product is not yet to be divulged.

This type extincteur may be actuated by conjoining the inner and outer solutions in one of the following ways: (*a*) by turning over (upside down) the extincteur; (*b*) by unlocking or piercing a sealing device and turning over the extincteur; or (*c*) by turning over the extincteur or otherwise causing an hermetically sealed bottle containing the inner solution to be broken.

Type (iv) carbon tetrachloride extincteurs. The $C.Cl_4$ used in these should conform to BS. 575/34. It shall be clear, colourless and free from undissolved water and other visible impurities, Sp. Gr. 1·600–1·608 at 15·50° C. and 1·596–1·604 at 20° C.

The $C.Cl_4$ is propelled from extincteurs by one of four methods: (*a*) by means of a direct-acting self-contained hand force pump operated by outward and inward strokes of the plunger by manual effort not exceeding 25 lbs. pull or thrust; (*b*) by expansion of liquefied CO_2 or compressed nitrogen from a small sealed cylinder attached to the extincteur and operated by suitable mechanical means thereby causing the seal of the cylinder to be pierced and liquefied CO_2 or compressed nitrogen to be released into the $C.Cl_4$ container; (*c*) by pressure of compressed CO_2 or nitrogen stored in the $C.Cl_4$ container and operated by suitable mechanical means thereby puncturing the seal of the $C.Cl_4$ container and releasing its contents; (*d*) by pressure of air compressed into the $C.Cl_4$ container at the time the extincteur is operated by means of a hand air pump forming an integral part of the extincteur.

The minimum length of jet of $C.Cl_4$ propelled by any one of these extincteurs shall be 25 feet, and the minimum and maximum rates of discharge for pumps (*a*) and (*d*) shall be 1½ pints in 40–60 seconds for the smallest size; 1 gallon in 60–90 seconds for the largest size, and *pro rata* for intermediate sizes, while for pressure extincteurs (*b*) and (*c*) it shall be 1 quart in 60–80 seconds for the smallest size; 1 gallon in 40–90 seconds for the largest size, and *pro rata* for intermediate sizes.

Type (v) methyl bromide is a stable, colourless and volatile liquid with a boiling point of 4·5° C. which, being stored in a sealed container with added nitrogen to increase the pressure to 50–100 lbs. per sq. in. will, upon release, evaporate rapidly to a vapour of 3·27 times heavier than air at N.T.P. While no B.S.I. Specification has yet been issued for appliances charged with this product, an Air Ministry specification applies to such extincteurs for use on service aircraft and petrol-driven sea-craft.

Type (vi) carbon dioxide extincteurs consist of a standard CO_2 cylinder (BS. 401/31–38) with a CO_2 capacity of 2, 4, 7 or 15 pounds, fitted with a syphon tube, large valve opening, quick opening valve, non-freezing nozzle and conical entrainment shield to limit the amount of air entrained by the CO_2 at the point of discharge. They are charged with liquefied CO_2, which on rapid release emerges a mixture of CO_2 gas and CO_2 snow, particularly effective in dealing with electrical hazards and inflammable liquids.

Type (vii) CO_2 and carbonates appliances consist of a conical powder-container with a standard CO_2 cylinder attached thereto. They are made in two sizes, the smaller one having a carbonates capacity of 288 cub. in. and CO_2 capacity of 288 c.c., the larger ones having double those capacities. To operate the extincteurs they are inverted and the CO_2 released by a trigger valve into the carbonates-container. The CO_2 then emerges at high pressure from a conical nozzle with spreading baffle drawing with it the carbonates on the ejector principle and propelling them to the seat of combustion on which they settle and are converted into added CO_2. The combined discharge is di-electric at 120,000 volts.

Mobile Secondary-Aid Appliances.

In this group are larger models of some of the appliances before described, mounted on wheels, including water corridor pumps, maximum capacity 20 gallons, delivery 10 gallons per minute, effective range 45 ft.; and water and CO_2 pressure engines, maximum capacity 40 gallons, delivery 20 gallons per minute, effective range 60 ft.; also acid-alkali type and foam type chemical engines, ranging usually from 5 to 34 gallons, with range and rate of delivery according to size of nozzle specified.

Major Appliances.

Trailer Pumps have been extensively installed in industrial works during the past five years. The sizes usually employed for this purpose range from a 120/180 g.p.m. light type with pump

detachable from chassis, to a 350/500 g.p.m. heavier chassis type. Normal output of the former 150 g.p.m. at 100 lbs. sq. in.; single jet 130/90 ft. high, and of the latter 430 g.p.m. at 100 lbs. sq. in. single jet 140 × 110 ft. high, suction lift of former 10 ft. and of latter up to 25 ft. Multiple jets reduce length and height of throw, for instance, a single jet of 130 × 110 ft. as above to approximately 110 × 80 ft. double jet.

Colliery fire and rescue stations throughout Great Britain and many engineering, chemical and other industrial works are provided with complete modern fire brigade equipment as specified by the Home Office for the National Fire Service.

Modern petrol motor fire engines range in pumping capacity from 700–900 gallons per minute, 120 b.h.p. weight about 5½ tons to 300–500 gallons per minute, 60 b.h.p. weight about 3½ tons, both working up to pressures of 200 lb. per sq. in. The pumps employed are of two types, multi-stage turbine and reciprocating, the former being provided with a priming device to 'lift' the water when the pump is not coupled direct to a pressure main. Many of these engines (usually described as dual-purpose or D.-P. appliances) carry wheeled escapes which are removable for use separately in case of necessity.

The brigades of large cities are rapidly acquiring mobile turntable-ladders (alt. T.T. appliances), viz. telescopic revolving ladders extending to heights up to 100 ft., the operating mechanism being driven by the petrol motor that propels the machine. These appliances are primarily designed to enable fires in high buildings to be attacked from above but they are also used for life-saving.

The most modern T.T. are in four sections, made entirely of steel. Some are fitted with pumps.

FIXED INSTALLATIONS.

In this group are included hydrant installations. Sluice valve type, or screw-down type or ball type and unions according to local practice. Automatic sprinkler and drencher systems (wet or dry) usually with dual water supply, viz. town water main (with booster pump) and overhead emergency water tank; continuous foam generating plants; pipe installations for reception of water or foam from fire brigade motor pump; static water and CO_2 pressure tanks with pipe line and hose to special points of danger; batteries of CO_2 cylinders or CH_2Br. cylinders (automatically operated by a thermostat or electricon smoke detector) and emulsion-forming water-jet apparatus. To determine the most efficient of these systems for installation in any given circumstances is a matter for reference to the F.O.C. and/or study of reports issued by technical institutions as, for instance, E.R.A. Report V/T.8–1940.

Fire Hose.

The hose generally used by fire brigades is of 2½-in. and 2¾-in. internal diameter. It is woven from flax line and the average grades are suitable for working pressures not exceeding 130 lbs. per sq. in.

The London Fire Brigade uses 2¾-in. diameter rubber-lined flax hose. This has a much lower frictional loss as compared with unlined hose mentioned above, and also the waste of water by porosity leakage is prevented by the rubber lining. The maximum working pressure is 150 lbs. per sq. in.

The loss of head due to friction in delivering 100 gallons per minute through each 100-ft. length of hose is as follows:—

Size of hose.	*Head lost in friction.*
2½-in. unlined	20 ft.
2¾-in. unlined	13 ft.
2¾-in. rubber-lined	5 ft.

Thus, with ten (100-ft.) lengths of hose or 1,000 ft., the friction loss amounts to 200, 130 and 50 ft., respectively. Assuming the head available at the pump to be 300 ft., or 130 lbs. per sq. in., there would be 100 ft., 170 ft. and 250 ft., respectively, available at the jet, which is equivalent to pressures of 31 lbs., 73 lbs. and 107 lbs. per sq. in.

The maximum lengths of hose at which an efficient ¾-in. jet can be obtained with a nozzle pressure of 50 lbs. per sq. in. are as follows:—

Size of hose.	*No. of 100-ft. lengths.*
2½-in. unlined	9
2¾-in. unlined	14
2¾-in. rubber-lined	37

The above figures indicate the advantage of using hose of a larger diameter and the great advantage of rubber lining.

If it be necessary to pump larger quantities of water over longer distances twin lines of hose should be laid out and several pumps arranged in relay or series working.

BRITISH STANDARD FIRE HOSE COUPLINGS.
(No. 336—1936.)

The specification gives the dimensional particulars for three types of couplings—the V-thread, the round thread, and the Morris instantaneous type. The sizes of the two former are 2¼ ins., 2½ ins., and 2¾ ins., and for the latter 2½ ins. and 2¾ ins. diameter with interchangeability.

The following table will be found useful :—

Fire Streams.

Table showing Pressure required at Nozzle and at Pump, with Quantity and Pressure necessary to Throw Good Effective Streams various Distances through Different Size Nozzles, using 100 Feet of Ordinary $2\frac{1}{2}$-inch Rubber-lined Hose and Smooth Nozzles.

Size of Nozzle, $\frac{3}{4}$ inch.							
Pressure at nozzle, in lbs. per sq. in.	40	50	60	70	80	90	100
Pressure at pump, „ „ „	46	57	68	80	91	102	114
Imperial gallons per minute . .	86	96	105	114	122	129	136
Distance thrown horizontal, in ft.	44	50	54	58	62	65	68
Distance thrown vertical, in ft. .	60	67	72	76	79	81	83
Size of Nozzle, $\frac{7}{8}$ inch.							
Pressure at nozzle, in lbs. per sq. in.	40	50	60	70	80	90	100
Pressure at pump, „ „ „	50	63	75	88	101	113	126
Imperial gallons per minute . .	118	132	144	156	167	177	186
Distance thrown horizontal, in ft.	49	55	61	66	70	74	76
Distance thrown vertical, in ft. .	62	71	77	81	85	88	90
Size of Nozzle, 1 inch.							
Pressure at nozzle, in lbs. per sq. in.	40	50	60	70	80	90	100
Pressure at pump, „ „ „	58	72	87	101	115	130	144
Imperial gallons per minute . .	154	173	189	204	218	232	245
Distance thrown horizontal, in ft.	55	61	67	72	76	80	83
Distance thrown vertical, in ft. .	64	73	79	85	89	92	96
Size of Nozzle, $1\frac{1}{8}$ inches.							
Pressure at nozzle, in lbs. per sq. in.	40	50	60	70	80	90	100
Pressure at pump, „ „ „	69	86	103	120	138	155	172
Imperial gallons per minute . .	197	221	241	260	279	295	312
Distance thrown horizontal, in ft.	59	66	72	77	81	85	89
Distance thrown vertical, in ft. .	65	75	83	88	92	96	99
Size of Nozzle, $1\frac{1}{4}$ inches.							
Pressure at nozzle, in lbs. per sq. in.	40	50	60	70	80	90	100
Pressure at pump, „ „ „	84	106	127	148	169	190	211
Imperial gallons per minute . .	246	275	301	325	348	368	388
Distance thrown horizontal, in ft.	63	70	76	81	85	90	93
Distance thrown vertical, in ft. .	67	77	85	91	95	99	101
Size of Nozzle, $1\frac{3}{8}$ inches.							
Pressure at nozzle, in lbs. per sq. in.	40	50	60	70	80	90	100
Pressure at pump, „ „ „	107	134	160	187	214	240	268
Imperial gallons per minute . .	301	337	369	398	426	452	476
Distance thrown horizontal, in ft.	66	73	79	84	88	92	96
Distance thrown vertical, in ft. .	69	79	87	92	97	100	103

N.B.—The above pressures are based on the supposition that the hose is coupled direct to the delivery of the pump and while the stream is flowing; if, however, the hose is coupled to a hydrant which is supplied direct from the pump, then the corresponding fire pump pressure must be greater than the hydrant pressure by an amount equal to friction loss and difference of head between hydrant and pump.

The pressures given in the table are *indicated* pressures, not effective pressures; effective pressures would be slightly greater.

The distances given are for *effective* fire *streams* adapted for fire purposes, and are not for mere isolated drops. (*Freeman.*)

FIRE STREAMS AND ELECTRIC CIRCUITS.

It has always been believed to be dangerous to those handling a hose to allow the stream to come in contact with an electric circuit. Prof. F. C. Caldwell has found by experiment that there is no danger of shocks when the distance between the nozzle and circuit is more than 25 ft., irrespective of the voltage. ('*Power.*')

Fire-Resisting Buildings.

The following is a reasonable standard for these :—

(1) Walls should be of hard burnt brick or cement concrete (composed of cement and an aggregate of sand, gravel, broken brick, burnt ballast or the like; a coke breeze aggregate should be avoided). External or party walls of brickwork should be at least 8½ ins. in thickness. External walls constructed of reinforced concrete should be not less than 4 ins. in thickness in the panels, and party walls of reinforced concrete should not be less than 8 ins. in thickness.

(2) Walls built as described above and containing no openings are of great value in checking the spread of fire.

(3) Stone walls have not been found to resist action of fire and water so satisfactorily as brick or concrete.

(4) (*a*) In all cases roofs should be covered externally with fire-resisting material.

(*b*) Roofs should be constructed as strongly as possible of incombustible material and, if glazing is necessary, glass should be not less than ¼ in. in thickness in direct combination with metal, the melting point of which is not lower than 1800° F., or 1500° F. for glass reinforced with wire mesh. The panels of glass should not exceed 2 ft. across and should be secured in frame of hardwood not less than 1¾ in. finished thickness or of iron. Broken panes should be replaced immediately.

(5) Floors should be of incombustible material. If wooden flooring is desired the wooden flooring should be laid solidly on the concrete without spaces between the wooden flooring and the concrete. The wooden flooring should not extend through any doorway openings forming connections with adjoining buildings.

(6) Any metal column or beam (except underside of the beams and edges of the flanges thereof) in an external wall or wholly or partly within a recess in a party wall should be protected with not less than 4 ins. of solid incombustible material executed with Portland cement and fixed close to the metalwork without cavities. All other constructional metalwork, including the underside and edges of flanges of beams in external and party walls should be protected with 2 ins. of solid incombustible material fixed close to the metalwork without cavities.

The internal protection to metalwork need not be provided in a one-storey building not exceeding 25 ft. in height.

(7) Staircases, lifts and any other floor perforations should be enclosed by brick walls not less than 8½ ins. in thickness or with reinforced concrete not less than 5 ins. in thickness and all openings in these walls should be fitted with fire-resisting doors arranged with automatic control.

(8) Windows should be glazed with glass conforming with that described above in respect of glazing in roofs and should be in panels not exceeding 2 ft. across either way.

(9) Doorways forming connections with adjoining buildings, where the buildings united by the openings will exceed a combined cubical extent of 250,000 cub. ft., should have the floor jambs and head formed of brick, stone, iron or other incombustible materials and be closed by two wrought iron or mild steel doors, sliding doors or shutters, each not less than ¼ in. thick in the panel at a distance from each other of the full thickness of the wall fitted to grooved or rebated iron frames without woodwork of any kind and all such doors, sliding doors and shutters should be fitted with sufficient and proper bolts or other fastenings and be capable of being opened from either side and shall have on each face thereof styles and rails at least 4 ins. wide and ½ in. thick and should be constructed, fitted and maintained in an efficient condition.

(10) The use of matchboarding for lining or similar purposes should be avoided.

(11) Apart from the above suggestions, building owners and/or their representatives should satisfy themselves that effective compliance with local Acts, Bylaws and/or Regulations is secured.

Fire-Resisting Doors.

The woods mentioned in the following schedule are suitable for use in the construction of fire-resisting doors. The frames of such doors should also be constructed in accordance with the schedule.

It should, however, be noted that wooden doors are effective against fire for a comparatively short period only.

SCHEDULE.

1. *Hard woods.* Oak, Teak, Jarrah, Karri, Indian Silver Grey Wood, Tasmanian Myrtle Wood, Andaman Paduak, Nigerian Iroko, Gold Coast Odum, African Walnut, Nora, Crabwood, Rhodesian Teak, Rhodesian Bean Mahogany, New Zealand Matai, Pyinkado, Andaman Pyinma, Queensland Maple, Secondee Mahogany, Red Moranti, English Ash, Douglas Fir, Keruing, Malayan Kapur, Borneo White Seraya, not less than 1¾ ins. finished thickness.

2. *Soft woods* which have been impregnated throughout with ammonium phosphate, subject to the following conditions:

(*a*) That the whole of the wood shall be thoroughly impregnated;

(*b*) That a door made of soft wood similarly impregnated and measuring 6 ft. by 3 ft. by 2 ins. thick shall resist the passage of smoke or flame when exposed to a temperature of 1,500° F., for a period of one hour, and that, when so exposed, the increase of temperature registered by an unshielded thermometer 12 ins. away from the outer surface of the door shall not exceed 10° F.

(*c*) That the thickness of the wood shall be at least 1¾ ins. and that, where this thickness is built up, the finished thickness shall be not less than 1¾ ins.;

(*d*) That all panels and boarding shall be properly tongued and grooved.

Fire-Resisting Substances.

Exhaustive tests seem to show that sulphate of aluminium is one of the best fire-resisting substances for wood, as it checks combustion by forming an infusible and non-conducting coating, while such substances as ammonium sulphate or ammonium phosphate when heated check combustion by emitting an incombustible gas.

The following is the formula for what is known as the United States 'Government whitewash mixture,' which also acts as a fire-retarding coating over interior wooden surfaces:—Slake a half bushel of quicklime with boiling water, keeping it covered during process; strain and add one peck of salt dissolved in warm water; put 3 lbs. ground rice in water and boil to a thin paste; ½ lb. of powdered Spanish whiting; 1 lb. of clean glue dissolved in hot water. Mix well and let it stand for several days. Keep the mixture in a kettle or receptacle, and apply as hot as possible with a whitewash or paint brush.

(See also p. 306, Vol. I.)

Fire Prevention at Sea.

The International Convention for safety of life at sea, 1929, drafted provisions for passenger-carrying ships which are now compulsory throughout the world. The regulations are obtainable from H.M. Stationery Office. Ship fires both afloat and in harbour have directed attention to the problem of ship protection and in consequence many vessels are now equipped with sprinkler systems, similar to those fixed in buildings or with an installation of CO_2 cylinders.

Air, when diluted with between 25/30 per cent. CO_2 at atmospheric pressure, will not support combustion and under the aforesaid regulations such concentration is required in connection with the fire protection of cargo and machinery spaces of passenger-carrying ships. For fire protection purposes it is stored in cylinders, from which it is released by the perforation or destruction of a metal disc closing the valve orifice, either by a hand or mechanically operated lever or by a supply of gas from a pilot cylinder, which is allowed to press on a piston in contact with the disc perforator.

Oil Fires.

The 'foam' principle is now generally adopted for the protection of storage tanks.

A suitable 'foam' may be made by mixing saturated solutions of bicarbonate of soda and aluminium sulphate with glue, liquorice powder, quassi bark, etc. The solutions when mixed produce foam in a bulk of approximately eight times their volume, which flow readily over the burning surface. Foam to an even depth of 6 ins. over the burning material will extinquish an outbreak in an oil tank.

The solutions can be stored in tanks, or they can be made as required in small tanks having a continuous and controlled water supply into which the dry ingredients are introduced by mechanical means at a desired rate. The two liquids, prepared separately, are pumped to the point where foam is required, or may be delivered by gravity or other means.

Another method for the continuous generation of foam, where a high-pressure water supply exists, is to insert the ingredients in the form of a single dry powder through an open-top hopper

into a chamber in which a partial vacuum is formed by the passage of water through a nozzle on the injector principle. The water supply enters at one end of the chamber and carries the powder from the opposite end into a pipe-line in which the foam is formed. Two machines of this character can be used to convey two separate powders to a distant meeting point where foam is formed.

The ideal to be aimed at is the formation of the foam as close as possible to its point of application to the burning liquid, and to deliver it with as little disturbance as possible of the burning surface.

Apparatus for the continuous generation of foam for City Fire Brigade use can be readily accommodated on an automobile vehicle, the water supply being obtained from street hydrants.

Another method for the production of foam consists in aerating water by mechanical means and introducing a small percentage of saponin, or like material, into the froth produced.

SECTION XXXV

REFRIGERATION AND COLD STORAGE.

(pp. 719–749)

(Contributed by M E Anderson, A.M.IMech.E.)

SECTION XXXV

REFRIGERATION AND COLD STORAGE.

(Contributed by M. E. Anderson, A M.I.Mech.E.)

Principles.

Refrigeration is the removal of heat from air, water, brine or any other substance which it is desired to cool, and its transference to a suitable cooling medium—usually water, sometimes the atmosphere.

In the usual engineering sense of the term, refrigeration involves reducing the temperature of the air or whatever it may be below that of the cooling medium, and, as the Second Law of Thermodynamics states, this cannot be done without the expenditure of energy.

There are three principal systems by which this transfer of heat from a cold substance to a warmer one can be accomplished: the cold air machine, the absorption plant, and the vapour compression system. The first of these is now almost obsolete, though it has recently been revived for cooling aeroplane cabins, the second has a limited field of application, and the third is used in the great majority of cases.

The vapour compression cycle is a modification of the reversed Carnot cycle, which comprises the following four stages: adiabatic compression; isothermal compression with rejection of heat to the cooling medium; adiabatic expansion; isothermal expansion with absorption of heat from the substance to be cooled.

The coefficient of performance (ratio of heat removed to energy expended in heat units) of a plant working on the reversed Carnot cycle would be

$$\frac{T_1}{T_2 - T_1}$$

where T_1 is the temperature at which heat is removed from the substance to be cooled and T_2 the temperature at which heat is rejected to the cooling medium.

In the vapour compression system, the working substance is a fluid capable of being condensed at a temperature not much above that of the cooling medium and at a pressure not excessively high. The isothermal compression stage of the reversed Carnot cycle is replaced by a process of removal of heat at constant pressure, involving the condensation of the fluid; the adiabatic expansion is replaced by expansion through a regulating valve and the isothermal expansion by absorption of heat at constant pressure, involving evaporation of the fluid. As a result of these modifications, and particularly of the use of a regulating valve instead of an expansion cylinder, the coefficient of performance of the plant is always less than would be obtained on the Carnot cycle. The difference, however, is not very great in the case of all the generally used refrigerants except carbon dioxide, although it does become more marked the lower the evaporation temperature and the higher the condensation temperature.

A simple refrigerating apparatus would consist of a coil of piping immersed in brine, the piping containing a suitable liquid such as ammonia, and having the ends open to the atmosphere. The pressure in the coil being that of the atmosphere, the temperature of the ammonia would fall to the saturation temperature corresponding to 14·7 lb./sq. in. abs., *i.e.* -28° F., and at this temperature it would boil, absorbing heat from the brine. The limitations of such an apparatus would be (1) that a temperature lower than -28° F. could not be obtained; (2) that the refrigerant fluid would have to be constantly replenished in order to ensure a continuous supply of refrigeration.

The coil just described is a simple form of an apparatus which forms part of every refrigerating plant, the *evaporator*. The object of the other principal components is to overcome the limitations mentioned above. In order that the vapour leaving the evaporator may be recovered and used again, it must be liquefied. For this purpose its pressure must be raised to such a level that the corresponding saturation temperature is higher than the temperature of the cooling medium. This is the function of the *compressor* (fig. 1). From the compressor the hot vapour passes to the *condenser*, where it first loses its superheat and is then liquefied, giving up heat to the cooling medium—usually water, except in the case of small plants, which frequently have air-cooled condensers. The liquid leaving the condenser is now ready for re-use in the evaporator, to which it is admitted through the *regulator* or expansion valve, which reduces its pressure to that existing in the evaporator, and also controls its rate of flow in such a way that just so much is admitted as can be evaporated with the temperature difference prevailing at the time.

This regulation of the flow was effected by hand in older plants, but is now generally automatic, the regulator in such cases being usually one of three principal types: (1) low pressure float, maintaining a constant level of liquid in the evaporator or in a vessel which feeds it; (2) high pressure

float, keeping the condenser drained of liquid (this necessitates the charge of refrigerant in the plant being correct); (3) thermostatic (controlling the superheat of the vapour leaving the evaporator). Regulation should be such that no unevaporated liquid returns to the compressor; any such liquid causes a serious reduction in the pumping capacity of the compressor, because it is only partially vaporised on the delivery stroke of the piston, some liquid remaining in the clearance space at the end of this stroke and vaporising during the suction stroke. Ideally, the refrigerant leaving the evaporator would be in the dry saturated state; but in practice it is necessary that it

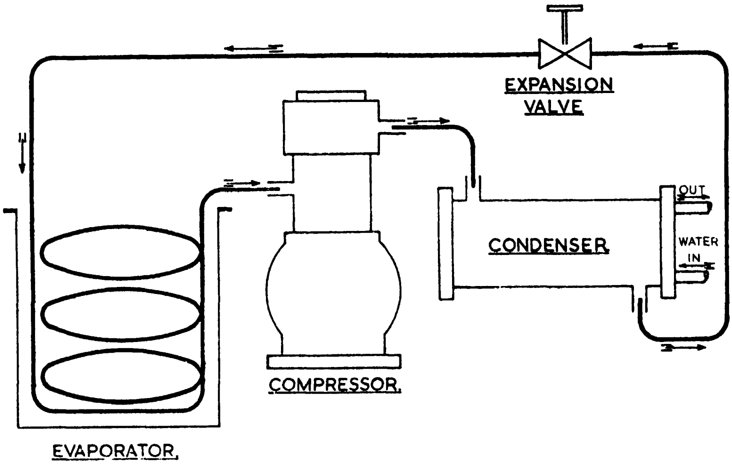

FIG. 1.—Vapour Compression System.

should be superheated to some extent, usually by at least 10° F., to ensure the complete absence of liquid.

In its passage through the regulator, the pressure of the refrigerant falls to that existing in the evaporator, and its temperature falls correspondingly; the heat thus removed from the liquid is used in evaporating a part of it, and the vapour so produced has to be pumped by the compressor, without providing any refrigerating effect, the coefficient of performance being thereby reduced. The proportion of refrigerant wasted in this way varies directly as the ratio of specific to latent heat, which should, therefore, be as low as possible. It does not in fact differ greatly for the various common refrigerants with the exception of carbon dioxide, for which it is exceptionally high.

Effect of Variation in Working Conditions.

The expression for the Coefficient of Performance on the reversed Carnot cycle shows that the ratio of refrigerating duty to power absorbed falls with decreasing evaporation temperature, and also with increasing condensation temperature. In the vapour compression cycle these effects are still more marked, because the loss through the expansion valve increases with increase in difference between the two temperatures.

Moreover, the refrigeration obtainable from a given compressor falls with decreasing evaporation temperature, and this would be so even with a perfect machine, because the refrigerating effect depends largely on the weight of vapour pumped in unit time, and this in turn depends on the density of the vapour, which decreases with the evaporation temperature and pressure. Again, the various losses which reduce the volumetric efficiency of the compressor (clearance effect, heating of gas during suction stroke, leakage past piston and throttling through valves), all increase with increasing difference between evaporation and condensation temperatures, so that the variations in performance with operating conditions to be expected from an ideal machine are still more marked in fact.

Tables I to IV, XIV and XIVA show the fundamental properties of various refrigerants, from which the evaporator duty obtainable from an ideal machine can be calculated as follows: Heat absorbed per pound = enthalpy of saturated vapour − enthalpy of liquid. Heat absorbed per cub. ft. pumped = heat absorbed per pound ÷ specific volume.

In order to determine the refrigeration obtainable from an actual machine, the ideal duty so calculated must be multiplied by the volumetric efficiency of the compressor. This is difficult to determine from first principles, and in the case of a new design experiment is the only safe guide.

Another use of the tables is, when the duty of a machine is known, to determine the volume and weight of refrigerant in circulation, for the purpose of fixing the pipe sizes.

It will be noted that in the case of carbon dioxide an additional table is included, showing the enthalpy of the liquid. This is necessary because the enthalpy of a liquid near its critical point (which for CO_2 is as low as 87·8° F.) depends not only on its temperature, but also to a large extent on its pressure.

Tables V to VII show the evaporator duty to be obtained from an actual machine, working with the principal refrigerants. The values shown are based on tests, but it will be understood that they must be regarded as a guide only, since variations in size, design and manufacture will considerably influence the volumetric efficiency. The machines on which the figures are based have a volumetric clearance of 3 to 4 per cent.

Horse-Power.

The indicated horse-power, *i.e.* the energy consumed in compressing and discharging the refrigerant, is given by the formula

$$\text{I.h.p. for 1,000 cu. in./min.} = 0 \cdot 00252 \frac{n}{n-1} P_1 \left\{ \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right\}$$

Tables X to XII have been calculated on this basis.

To find the i.h.p. absorbed by an actual compressor, the figures obtained from Tables X to XII must be multiplied by a factor corresponding to the work done by the re-expansion of the vapour left in the clearance space at the end of compression. This is always a matter of some doubt, since the precise extent to which the re-expansion departs from an adiabatic process cannot be known. Approximate values, however, are given in Table VIII.

When the i.h.p. has been obtained as described, the actual power required to drive the compressor (b.h.p.) is found by adding the friction h.p. The latter, naturally, depends to a considerable extent on design and manufacture; but approximate values for an average two-cylinder compressor are shown in Table IX.

Indicator Diagrams.

An indicator diagram, besides giving the only precise information as to the indicated horsepower absorbed by a compressor (area of diagram), also reveals any departure from good working conditions.

Fig. 2 shows a diagram from a good compressor.

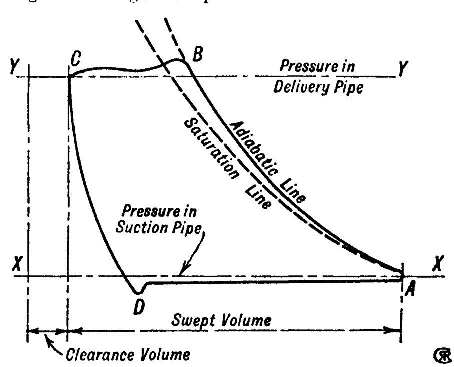

FIG. 2.

Defects are revealed as follows:

Liquid refrigerant drawn into cylinder:	Point D will lie further to the right.
Throttling through delivery valve:	Line BC lies too far (more than 1–2 lb./sq. in. above YY.
Throttling through suction valve:	Line AD lies too far below XX.
Delivery valve spring too heavy:	Peak at B is large.
Leakage past piston:	Line AB approaches saturation line.
Leaking suction valve:	” ” ” ” ”
Leaking delivery valve:	Point D will lie further to the right, and curve AB will be steeper than the adiabatic line.

TABLE I.—PROPERTIES OF SATURATED AMMONIA, NH_3.

Temperature. °F.	Pressure (Absolute). Lb. per sq. in.	Specific Volume. Cu. ft. per lb.		Enthalpy.* B.Th.U.'s per lb.	
		Liquid.	Vapour.	Liquid.	Vapour.
− 80	2·74	0·02236	86·54	− 42·2	580·1
− 70	3·94	0·02256	61·65	− 31·7	584·4
− 60	5·55	0·02278	44·73	− 21·2	589·6
− 50	7·67	0·02299	33·08	− 10·6	593·7
− 40	10·41	0·02322	24·86	0·0	597·6
− 30	13·90	0·02345	18·97	10·7	601·4
− 20	18·30	0·02369	14·68	21·4	605·0
− 10	23·74	0·02393	11·50	32·1	608·5
0	30·42	0·02419	9·116	42·9	611·8
10	38·51	0·02446	7·304	53·8	614·9
20	48·21	0·02474	5·910	64·7	617·8
30	59·74	0·02503	4·825	75·7	620·5
40	73·32	0·02533	3·971	86·8	623·0
50	89·19	0·02564	3·294	97·9	625·2
60	107·6	0·02597	2·751	109·2	627·3
70	128·8	0·02632	2·312	120·5	629·1
80	153·0	0·02668	1·955	132·0	630·7
90	180·6	0·02707	1·661	143·5	632·0
100	211·9	0·02747	1·419	155·2	633·0
110	247·0	0·02790	1·217	167·0	633·7
120	286·4	0·02836	1·047	179·0	634·0

TABLE II.—PROPERTIES OF SATURATED DICHLORODIFLUOROMETHANE (CCl_2F_2; Freon–12).

Temperature. ° F.	Pressure (Absolute). Lb. per sq. in.	Specific Volume. Cu. ft. per lb.		Enthalpy.* B.Th.U.'s per lb.	
		Liquid.	Vapour.	Liquid.	Vapour.
− 120	0·6417	0·00981	46·84	− 16·94	64·04
− 110	0·9709	0·00989	31·84	− 14·78	65·22
− 100	1·430	0·00998	22·20	− 12·64	66·40
− 90	2·054	0·01007	15·86	− 10·51	67·59
− 80	2·885	0·01016	11·57	− 8·40	68·77
− 70	3·971	0·01026	8·608	− 6·30	69·95
− 60	5·365	0·01036	6·516	− 4·20	71·13
− 50	7·125	0·01047	5·012	− 2·11	72·31
− 40	9·317	0·01057	3·911	0·00	73·50
− 30	12·02	0·0107	3·088	2·03	74·70
− 20	15·28	0·0108	2·474	4·07	75·87
− 10	19·20	0·0109	2·003	6·14	77·05
0	23·87	0·0110	1·637	8·25	78·21
10	29·35	0·0112	1·351	10·39	79·36
20	35·75	0·0113	1·121	12·55	80·49
30	43·16	0·0115	0·939	14·76	81·61
40	51·68	0·0116	0·792	17·00	82·71
50	61·39	0·0118	0·673	19·27	83·78
60	72·41	0·0119	0·575	21·57	84·82
70	84·82	0·0121	0·493	23·90	85·82
80	98·76	0·0123	0·425	26·28	86·80
90	114·3	0·0125	0·368	28·70	87·74
100	131·6	0·0127	0·319	31·16	88·62
110	150·7	0·0129	0·277	33·65	89·43
120	171·8	0·0132	0·240	36·16	90·15
130	194·9	0·0134	0·208	38·69	90·76
140	220·2	0·0138	0·180	41·24	91·24

* Saturated liquid at − 40° F. = 0.

Table III.—Properties of Saturated Methyl Chloride (CH_3Cl).

Temperature. °F.	Pressure (Absolute). Lb. per sq. in.	Specific Volume. Cu. ft. per lb.		Enthalpy.* B.Th.U.'s per lb.	
		Liquid.	Vapour.	Liquid.	Vapour.
− 80	1·953	0·01493	41·08	− 13·888	184·75
− 70	2·751	0·01508	29·84	− 10·521	186·25
− 60	3·799	0·01523	22·09	− 7·039	187·74
− 50	5·155	0·01538	16·64	− 3·532	189·19
− 40	6·878	0·01553	12·72	0·000	190·66
− 30	9·036	0·01568	9·873	3·562	192·08
− 20	11·71	0·01583	7·761	7·146	193·49
− 10	14·96	0·01598	6·176	10·75	194·87
0	18·90	0·01613	4·969	14·39	196·23
10	23·60	0·01631	4·038	18·04	197·58
20	29·16	0·01647	3·312	21·73	198·84
30	35·68	0·01665	2·739	25·44	200·03
40	43·25	0·01684	2·286	29·17	201·17
50	51·99	0·01704	1·920	32·93	202·28
60	62·00	0·01724	1·624	36·71	203·33
70	73·41	0·01744	1·382	40·52	204·34
80	86·26	0·01764	1·183	44·36	205·27
90	100·6	0·01786	1·018	48·21	206·13
100	116·7	0·01808	0·8811	52·09	206·94
110	134·5	0·01833	0·7672	56·00	207·70
120	154·2	0·01859	0·6710	59·93	208·39
130	175·9	0·01887	0·5889	63·89	209·02
140	199·6	0·01915	0·5189	67·87	209·58

Table IV.—Properties of Saturated Monochlorodifluoromethane (CH_3ClF_2; Freon-22).

Temperature. °F.	Pressure (Absolute). Lb. per sq. in.	Specific Volume. Cu. ft. per lb.		Enthalpy.* B.Th.U.'s per lb.	
		Liquid.	Vapour.	Liquid.	Vapour.
− 150	0·260	0·0103	146·06	− 27·79	87·36
− 140	0·433	0·0103	90·61	− 25·25	88·53
− 130	0·695	0·0104	58·21	− 22·73	89·70
− 120	1·079	0·0105	38·60	− 20·22	90·88
− 110	1·626	0·0106	26·33	− 17·73	92·07
− 100	2·386	0·0107	18·43	− 15·23	93·27
− 90	3·417	0·0108	13·20	− 12·73	94·47
− 80	4·787	0·0109	9·650	− 10·22	95·68
− 70	6·571	0·0110	7·192	− 7·69	96·88
− 60	8·856	0·0111	5·452	− 5·16	98·08
− 50	11·74	0·0112	4·195	− 2·58	99·28
− 40	15·31	0·0114	3·279	0·00	100·46
− 30	19·69	0·0115	2·594	2·62	101·63
− 20	24·99	0·0116	2·075	5·29	102·79
− 10	31·34	0·0118	1·678	7·99	103·92
0	38·87	0·0119	1·370	10·72	105·02
10	47·66	0·0121	1·130	12·87	105·89
20	58·00	0·0122	0·936	15·66	106·93
30	69·97	0·0124	0·781	18·53	107·93
40	83·72	0·0126	0·656	21·50	108·89
50	99·40	0·0128	0·554	24·53	109·78
60	117·2	0·0130	0·470	27·63	110·58
70	137·2	0·0132	0·400	30·79	111·29
80	159·7	0·0135	0·342	34·07	111·93
90	184·8	0·0137	0·293	37·41	112·47
100	212·6	0·0140	0·252	40·78	112·86
110	243·4	0·0143	0·217	44·15	113·09
120	277·3	0·0147	0·187	47·65	113·32

* Saturated liquid at − 40° F. = 0.

TABLE V.—REFRIGERATING CAPACITY OF TYPICAL AMMONIA COMPRESSOR.

(B.Th.U.'s per hour for 1,000 cu. in. per minute of swept volume.)

Condensation and Liquid Temperature ° F.	Evaporation Temperature ° F.													
	–20	–15	–10	–5	0	5	10	15	20	25	30	35	40	45
65	833	980	1130	1330	1540	1780	2050	2360	2680	3060	3450	3890	4350	4850
75	774	913	1070	1240	1445	1670	1920	2200	2520	2850	3260	3660	4130	4630
85	711	846	1000	1165	1355	1565	1800	2065	2370	2700	3080	3480	3910	4410
95	650	780	925	1090	1270	1470	1690	1940	2220	2530	2880	3260	3700	4160
105	584	710	850	1010	1185	1380	1585	1825	2090	2380	2710	3080	3480	3910
115	515	640	775	930	1100	1290	1490	1710	1960	2240	2540	2890	3260	3700

TABLE VI.—REFRIGERATING CAPACITY OF TYPICAL METHYL CHLORIDE COMPRESSOR.

(B.Th.U.'s per hour for 1,000 cu. in. per minute of swept volume.)

Condensation and Liquid Temperature ° F.	Evaporation Temperature ° F.													
	–15	–10	–5	0	5	10	15	20	25	30	35	40	45	50
65	466	549	645	763	892	1025	1180	1350	1520	1700	1910	2120	2340	2580
75	434	514	607	720	847	975	1125	1290	1450	1635	1830	2040	2260	2500
85	401	479	570	680	801	925	1070	1230	1385	1565	1750	1960	2180	2410
95	370	444	533	637	753	878	1010	1165	1320	1495	1680	1890	2090	2320
105	338	410	495	595	708	825	957	1110	1255	1425	1600	1810	2010	2230
115	306	374	460	554	665	775	905	1045	1190	1350	1530	1730	1930	2140
125	274	340	423	513	617	729	847	983	1120	1285	1460	1650	1840	2050
135	242	305	385	472	570	678	797	925	1060	1210	1380	1570	1760	1960

This table applies to small machines, having a swept volume of, say, up to 20,000 cu. in. per minute. Larger machines may have an appreciably higher output.

TABLE VII.—REFRIGERATING CAPACITY OF TYPICAL FREON-12 COMPRESSOR.

(B.Th.U.'s per hour for 1,000 cu. in. per minute of swept volume.)

Condensation and Liquid Temperature ° F.	Evaporation Temperature ° F.													
	–15	–10	–5	0	5	10	15	20	25	30	35	40	45	50
65	582	678	790	920	1070	1225	1390	1565	1750	1940	2160	2380	2600	2840
75	533	625	730	855	995	1145	1300	1470	1650	1835	2050	2260	2480	2710
85	485	569	670	790	920	1065	1215	1375	1550	1730	1930	2140	2360	2590
95	435	515	610	723	847	985	1125	1280	1450	1625	1815	2010	2230	2460
105	385	460	553	655	775	905	1045	1190	1350	1520	1705	1900	2100	2330
115	336	406	493	590	700	825	955	1090	1210	1410	1585	1780	1980	2200
125	286	351	433	525	627	745	865	995	1140	1305	1470	1655	1850	2070
135	237	296	373	457	553	665	780	905	1040	1200	1360	1535	1730	1945

This table applies to small machines, having a swept volume of, say, up to 20,000 cu. in. per minute. Larger machines may have an appreciably higher output.

TABLE VIII.—CLEARANCE VOLUMETRIC EFFICIENCY.

Compression Ratio.	Clearance Percentage.				
	2	3	4	5	6
2	98·3	97·4	96·5	95·6	94·7
3	96·7	94·9	93·2	91·4	89·9
4	95·1	92·4	90·0	87·4	85·1
5	93·6	90·2	87·0	83·6	80·5
6	92·1	88·0	84·0	80·0	75·9
7	90·5	85·7	80·8	76·3	71·4
8	89·0	83·5	77·7	72·7	67·1
9	87·5	81·2	74·5	69·0	62·8
10	86·0	79·0	71·3	65·4	58·7

TABLE IX.—APPROXIMATE FRICTION HORSE-POWER (TWIN CYLINDER COMPRESSORS).

Swept Volume. Cu. in. per min.	Friction h.p.
50,000	2·5
100,000	4·5
150,000	6
200,000	8
300,000	11
400,000	14
500,000	17
750,000	24·5
1,000,000	31
1,250,000	36·5
1,500,000	42

TABLE X.—INDICATED HORSE-POWER FOR A SWEPT VOLUME OF 10,000 CU. IN. PER MINUTE. AMMONIA.

Evaporator Gauge. ° F.	Condenser Gauge. ° F.				
	60	70	80	90	100
− 40	0·81	0·89	0·97	1·05	1·125
− 30	0·91	1·01	1·115	1·22	1·325
− 20	1·005	1·125	1·255	1·385	1·51
− 10	1·08	1·23	1·385	1·53	1·69
0	1·125	1·31	1·49	1·68	1·865
10	1·125	1·35	1·57	1·79	2·015
20	1·08	1·34	1·61	1·87	2·13
30	0·95	1·27	1·58	1·89	2·20
40	0·74	1·11	1·48	1·85	2·22
50	0·42	0·85	1·28	1·71	2·14

TABLE XI.—INDICATED HORSE-POWER FOR A SWEPT VOLUME OF 10,000 CU. IN. PER MINUTE. METHYL CHLORIDE.

Evaporator Gauge. ° F.	Condenser Gauge. ° F.				
	60	70	80	90	100
− 40	0·48	0·52	0·565	0·61	0·655
− 30	0·54	0·595	0·65	0·705	0·755
− 20	0·585	0·655	0·725	0·79	0·855
− 10	0·62	0·705	0·79	0·87	0·95
− 0	0·645	0·755	0·86	0·95	1·035
10	0·645	0·765	0·885	1·00	1·115
20	0·595	0·74	0·885	1·025	1·165
30	0·52	0·69	0·865	1·035	1·205
40	0·405	0·605	0·805	1·000	1·195
50	0·23	0·465	0·695	0·92	1·145

TABLE XII.—INDICATED HORSE POWER FOR A SWEPT VOLUME OF 10,000 CU. IN. PER MINUTE. FREON-12.

Evaporator Gauge. ° F.	Condenser Gauge. ° F.				
	60	70	80	90	100
− 40	0·53	0·57	0·61	0·655	0·695
− 30	0·59	0·61	0·695	0·75	0·80
− 20	0·64	0·705	0·77	0·835	0·90
− 10	0·68	0·76	0·845	0·925	1·005
0	0·685	0·79	0·90	1·00	1·100
10	0·675	0·80	0·93	1·05	1·165
20	0·65	0·79	0·93	1·07	1·21
30	0·565	0·73	0·90	1·07	1·24
40	0·455	0·65	0·84	1·04	1·24
50	0·29	0·51	0·735	0·960	1·185

TABLE XIII.—VALUES OF EXPONENT IN PV^n = CONSTANT (ADIABATIC COMPRESSION).

Evaporator Gauge. ° F.	Ammonia.		Methyl Chloride.		Freon-12.	
	Condenser Gauge. ° F.					
	60	100	60	100	60	100
− 40	1·29	1·285	1·255	1·24	1·11	1·095
0	1·295	1·29	1·255	1·24	1·09	1·08
40	1·29	1·28	1·25	1·235	1·075	1·055

TABLE XIV.—PROPERTIES OF SATURATED CARBON DIOXIDE, CO_2.

Temperature ° F.	Pressure (Absolute). Lb. per sq. in.	Specific Volume. Cu. ft. per lb.		Enthalpy.* B.Th.U.'s per lb.	
		Liquid.	Vapour.	Liquid.	Vapour.
(1) − 69·9	75·1	0·01360	1·1570	− 13·7	135·9
− 60	94·7	0·01384	0·9270	− 9·2	136·6
− 50	118·2	0·01409	0·7492	− 4·7	137·2
− 40	145·8	0·01437	0·6113	0·00	137·8
− 30	177·8	0·01466	0·5029	4·5	138·2
− 20	214·9	0·01498	0·4168	9·1	138·5
− 10	257·3	0·01532	0·3472	13·9	138·7
0	305·5	0·01570	0·2904	18·8	138·9
10	360·2	0·01614	0·2437	24·0	138·7
20	421·8	0·01663	0·2049	29·4	138·3
30	490·8	0·01719	0·1722	35·4	137·8
40	567·8	0·01787	0·1444	41·7	136·7
50	653·6	0·01868	0·1205	48·4	135·0
60	748·6	0·01970	0·0994	55·5	132·1
70	853·4	0·02112	0·08040	63·7	127·5
80	968·7	0·02370	0·06064	73·9	118·7
(2) 87·8	1066·2	0·03454	0·03454	97·0	97·0

* Liquid at − 40° F. = 0. (1) Triple point. (2) Critical point.

TABLE XIVA.—ENTHALPY OF CARBON DIOXIDE LEAVING CONDENSER.
(B.Th.U.'s per Lb.)

Temp. F.	Pressure. Lb. per sq. in. Absolute.													
	700	800	900	1000	1050	1100	1150	1200	1250	1300	1350	1400	1450	1500
40	40·1	39·9	39·6	39·4	39·3	39·2	39·1	39·1	39·0	38·9	38·9	38·8	38·7	38·6
50	47·3	46·3	45·5	45·0	44·7	44·4	44·2	44·0	43·9	43·8	43·6	43·5	43·3	43·2
60		53·7	52·3	51·3	50·9	50·6	50·3	50·0	49·8	49·5	49·3	49·1	49·0	48·8
70			61·3	59·2	58·6	57·9	57·5	57·0	56·5	56·0	55·7	55·4	55·2	55·0
75				64·5	63·2	62·0	61·3	60·7	60·2	59·8	59·2	58·7	58·4	58·1
80				70·5	68·6	66·9	65·6	64·6	63·9	63·3	62·7	62·3	61·9	61·5
82					71·9	69·2	67·6	66·4	65·5	64·8	64·3	63·7	63·3	62·9
84					76·0	72·0	69·9	68·4	67·3	66·6	65·9	65·3	64·8	64·3
86					81·5	75·5	72·7	70·8	69·4	68·5	67·7	67·0	66·4	65·9
88							76·3	74·8	71·8	70·4	69·6	68·6	68·0	67·5
90							81·0	77·3	74·5	72·7	71·7	70·5	69·8	69·3
92								81·8	77·7	75·2	73·6	72·5	71·7	71·1
94								87·0	81·3	78·0	76·2	74·5	73·6	73·0
96									85·4	81·2	79·0	76·8	75·7	75·0
98										85·0	82·0	79·2	77·9	76·9
100										89·1	85·0	81·9	80·1	78·9
102										93·6	88·5	84·7	82·4	81·0
104										99·0	91·9	87·5	84·7	83·0

Carbon Dioxide Refrigerating Machines.

CO_2 occupies a unique position in the group of fluids used for refrigeration in that its critica temperature, 87·8° F., often comes in the working range; also its specific heat is high, and its latent heat relatively low.

Its efficiency with condensing water at about 50° F. is less than that of the ammonia machine and falls away considerably with an increasing condenser temperature. There is, however, no break in the curves of capacity at the critical temperature.

CO_2 is odourless, and in mild concentrations can be inhaled without danger, but with concentrations of about 10 per cent. it may cause unconsciousness if inhaled continuously, and at higher percentages will produce suffocation.

The quantities used in machines are, however, such that even if the whole charge were to escape in any ordinary situation it is hardly likely to produce a dangerous concentration, and on this account the Board of Trade regulations permit CO_2 machinery to be installed in the engine room or similar space on board ship, whereas ammonia machines must be housed in a special deck-house.

CO_2 does not attack any metals, so that condensers can be constructed of copper piping, and these advantages have given it a predominant position for marine refrigeration.

Although the pressures are high, the diameters of all parts are small, so that there are no constructional difficulties.

The figures given in Tables XV and XVI are for ordinary CO_2 compressors of recent construction. They represent the average results that are obtained with compressors of moderate size, but performance and power naturally vary to some extent with the characteristics of different makes.

To obtain the best results in working, a CO_2 machine should run with the suction gas in a dry saturated state, or even slightly superheated. When leather pistons and gland packings were used, this was impracticable as the leathers were burnt by the temperature reached with dry compression. All modern CO_2 machines have piston rings and metallic gland packings, so that there is now no difficulty in running a CO_2 machine to give the best results that can be obtained.

With high condensing temperature the performance of a CO_2 machine can be appreciably improved by permitting the cooling of the liquid by its own partial evaporation to take place in two stages working as described below.

In this system the refrigerant splits up into two separate cycles. In the first or main cycle the liquid from the condenser passes through a regulating valve into a receiver where it is cooled by the evaporation of the liquid which is passing through the second cycle. The cooled liquid in the first cycle passes through a second regulating valve to the evaporator and thence to the compressor.

In the second cycle the liquid from the condenser passes simultaneously with the liquid in the first cycle into the receiver. It is then completely evaporated and in so doing cools the liquid in the first cycle. The vapour formed passes at a higher pressure than the vapour formed in the evaporator in the first cycle either to the main compressor or to an auxiliary compressor.

TABLE XV.—REFRIGERATING EFFECT OBTAINED PER CU. FT. SWEPT VOLUME OF COMPRESSOR IN B.TH.U.'S WITH DIFFERENT CONDENSER AND EVAPORATOR TEMPERATURES.

CARBON DIOXIDE.

Temperature of Evaporation. ° F.	Suction Gauge Pressure in Lb. per sq. in.	Condenser Gauge.		
		70° F.	86° F.	105° F.
		Corresponding Gauge Pressure. Lb. per sq. in.		
		838·7	1028	1292
– 20	200·2	150	123	93·7
– 10	242·6	193	159	123
0	290·8	241	200	156
5	317·2	269	224	175
10	345·5	301	251	196
20	407·1	372	310	243
30	476·1	456	381	299

TABLE XVI.—COMPRESSOR B.H.P. PER CU. FT. SWEPT VOLUME PER MINUTE WITH DIFFERENT CONDENSER AND EVAPORATOR TEMPERATURES. CARBON DIOXIDE.

Temperature of Evaporation. ° F.	Suction Gauge Pressure in Lb. per sq. in.	Condenser Gauge.		
		70° F.	86° F.	105° F.
		Corresponding Gauge Pressure. Lb. per sq. in.		
		838·7	1028	1292
– 20	200·2	1·49	1·65	1·79
– 10	242·6	1·62	1·85	2·06
0	290·8	1·72	2·02	2·31
5	317·2	1·75	2·09	2·42
10	345·5	1·77	2·15	2·53
20	407·1	1·76	2·22	2·71
30	476·1	1·66	2·23	2·83

Characteristics of Refrigerants.

Ideally, a refrigerant should possess, among others, the following qualities :

High evaporator duty per cubic foot of vapour.
Moderate working pressures and pressure ratios.
Low delivery temperature.
Critical and triple points outside the working range.
Inert, non-toxic, non-corrosive, stable, non-inflammable.
Inexpensive and readily obtainable.

No refrigerant combines all these qualities.

Table XVII shows how the various refrigerants in common use comply with these requirements (except as regards toxicity and inflammability, which are dealt with separately).

TABLE XVII.

	Carbon Dioxide.	Ammonia.	Methyl Chloride.	Sulphur Dioxide.	Freon-12.	Freon-22.
B.Th.U.'s per cu. ft. *	209	58·2	33·6	22·0	34·4	56·0
Pressures . .	High	Moderate	Rather low	Rather low	Rather low	Moderate
Discharge temperature	Low	High	Moderate	Moderate	Low	Moderate
Coefficient of Performance *	2·56	4·79	4·90	4·73	4·72	4·47
Leakage tendency .	Normal	Normal	Normal	Normal	High	
Stability	Good	Good	Good	Good	Good	Good
Corrosion . .	Low	Low †	Low	High ‡	Low	Low
Cost (1948) . .	Low	Moderate	Moderate	Moderate	High	High

Refrigerants classified in *descending order* of inflammability and explosivity :

(1) Hydrocarbons, Methyl formate } Lower limit of concentration 1 per cent. to 3 per cent. Upper limit 6 per cent. to 10 per cent. Violence of explosion similar to that of coal gas.
(2) Ethyl chloride $3\frac{1}{2}$ per cent. to 12 per cent.
(3) Methyl chloride 8 ,, ,, 17 ,,
(4) Ammonia 16 ,, ,, 24 ,,
(5) F-21, Methylene Chloride (Carrene No. 1) } Almost negligible risk.
(6) The other Freons, Carbon Dioxide, Sulphur Dioxide } Non-inflammable.

Refrigerants classified in *descending order* of toxicity :

(1) Sulphur dioxide	Highly toxic.
(2) Ammonia	,, ,,
(3) Methyl formate	
(4) Methyl chloride	Moderately toxic.
(5) Ethyl chloride, Methylene Chloride	
(6) F-21, F-113	Slightly toxic.
(7) F-11, Carbon dioxide	
(8) F-22	
(9) F-12, F-114	Non-toxic.

Other characteristics of commonly used refrigerants are as follows :

Ammonia.—This is still the most widely used refrigerant, especially for industrial and commercial applications. It does not mix with lubricating oils to any considerable extent, and as liquid ammonia is much lighter than oil, the latter can be readily drained off from an ammonia evaporator.

Ammonia shows slight traces of decomposition in both compression and absorption plants, but not to an extent sufficient to cause any serious trouble.

It does not attack metals, with the exception of copper and its alloys: phosphor bronze gudgeon pin bushes are, however, quite commonly used in ammonia compressors, and are not appreciably affected in the absence of moisture. Probably a film of oil acts as a protection in some measure.

Leakage of ammonia may be readily detected by Nessler's reagent. Ammonia is very soluble in water.

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* Under U.S.A. Standard Ton Rating Conditions, 5° F. evaporation and 86° F. condensation. (C.o.P. on reversed Carnot cycle for these conditions : 5·74).

† Except for copper and its alloys.

‡ Especially in the presence of even small quantities of moisture.

Methyl Chloride.—This refrigerant is used to a considerable extent in small plants, its comparatively low working pressures simplifying construction. It is, however, tending gradually to be replaced by Freon-12.

Methyl chloride does not corrode the ordinary engineering metals in the absence of moisture, but when this is present corrosion is to be expected, particularly in the case of aluminium, zinc, and magnesium alloys. These materials should, therefore, never be used in methyl chloride plants, and every care should be taken to exclude moisture from the refrigerant circuit.

Liquid methyl chloride is miscible with oil and can only be separated from the latter by a distillation process.

Many organic materials are dissolved by liquid methyl chloride, and synthetic rubbers used for shaft seals must be carefully chosen.

Leaks of methyl chloride are best detected by the use of a soap and water solution applied to suspected points. The presence of traces of oil also often provides an indication of a leak.

Freon-12 (Dichlorodifluoromethane).—This modern refrigerant is being increasingly used, especially when safety is of paramount importance. As compared with methyl chloride, its disadvantages are a greater leakage tendency and the necessity for larger pipe sizes to avoid undue pressure drop (this arises because whereas the refrigerating duty obtainable from a given plant is practically the same with the two refrigerants, F-12 has about three times the density of methyl chloride, though only one-third of the latent heat). F-12 also requires considerably more condenser surface than methyl chloride, and, usually, slightly more power for a given compressor size. It is worth noting, therefore, that whereas a plant designed for Freon-12 can always be used for methyl chloride, conversion from methyl chloride to F-12 is not always possible without considerable modifications to the plant.

Freon-12 has no corrosive action on any of the usual metals, but is a solvent for some organic substances. Gaskets, shaft seals, etc., for F-12 may contain Neoprene or Chloroprene rubber, but not natural rubber.

It is quite stable under ordinary conditions, but will decompose, forming poisonous products, at high temperatures (above 1,000° F.).

F-12 is miscible with lubricating oil, being similar in this respect to methyl chloride.

It has only a slight odour, is non-inflammable and quite non-toxic, although, of course, if present in too large a quantity it will cause suffocation simply through exclusion of oxygen.

It has a higher leakage tendency than most refrigerants; leaks are detected by the change in the colour of the flame of a suitable ('Halide') torch. The same method is also sometimes used for methyl chloride, but not without risk, in consequence of the inflammable nature of the latter.

Calcium Chloride Brine.

TABLE XVIII.—SPECIFIC GRAVITY, CONCENTRATION, FREEZING POINT, AND HEAT CAPACITY.

Specific Gravity	1·10	1·12	1·14	1·16	1·18	1·20	1·22	1·24	1·26	1·28	1·30	
Percentage $CaCl_2$	11·4	13·6	15·7	17·8	19·8	21·8	23·7	25·6	27·5	29·4	31·2	Temp. ° F.
Freezing Point ° F.	19·1	15·1	10·6	5·4	−0·4	−6·4	−13·2	−20·7	−29·6	−41·0	−56·8	
Heat Capacity B.Th.U.'s per gallon per ° F.											8·33	−50
											8·325	−40
										8·27	8·325	−30
									8·27	8·29	8·325	−20
								8·34	8·31	8·31	8·335	−10
						8·485	8·42	8·375	8·34	8·33	8·345	0
					8·61	8·53	8·46	8·405	8·37	8·345	8·355	10
			8·875	8·76	8·66	8·565	8·50	8·435	8·405	8·37	8·37	20
	9·185	9·045	8·915	8·795	8·69	8·605	8·525	8·47	8·435	8·39	8·38	30
	9·21	9·065	8·945	8·825	8·72	8·635	8·56	8·50	8·46	8·425	8·405	40
	9·23	9·095	8·97	8·85	8·745	8·66	8·59	8·535	8·495	8·45	8·425	50
	9·25	9·115	8·99	8·875	8·77	8·685	8·62	8·56	8·525	8·485	8·45	60
	9·27	9·125	9·01	8·895	8·79	8·71	8·645	8·59	8·555	8·525	8·49	70

Eutectic conditions :—Sp. gr. 1·303, concentration 31·4 per cent., temperature — 59·8° F.

NOTES :—

(1) Various authorities differ regarding the data for conditions near the eutectic point. The figures given here are due to R. S. Jessup.

(2) 'Specific Gravity' refers to the ratio of the density of the brine at 60° F. to that of water at the same temperature.

(3) The increase of density of calcium chloride brine at temperatures below 60° F. may be taken as very nearly 0·19 per cent. for each 10° F. reduction in temperature.

TABLE XIX.—WEIGHT OF COMMERCIAL CALCIUM CHLORIDE (70 PER CENT. $CaCl_2$) TO BE USED IN MAKING 1 GALLON OF BRINE OF VARIOUS DENSITIES.

Specific Gravity	1·20	1·22	1·24	1·26	1·28	1·30
Pounds per Gallon	3·74	4·13	4·53	4·95	5·38	5·80
Minimum working temperature recommended . .	5	0	−10	−20	−30	−40

The use of a secondary refrigerant or brine is often adopted when refrigeration is required for a number of cold rooms or other applications, as it permits these to be controlled individually without complication of the primary refrigerant circuit; and it is almost universally used for ice making, except in quite small units. It is also frequently employed as a form of 'refrigeration storage.' Calcium chloride is the salt used in the great majority of cases, and is suitable for temperatures down to − 40° F.

In deciding the density of the brine, account must be taken of the fact that fairly weak solutions are more corrosive than strong ones; for this reason, brine of a lower specific gravity than 1·20 should not be used.

Another important point in connection with the corrosive tendency of brine is the *p*H value. This should normally be kept between 8·5 and 10, by the addition of caustic soda when necessary; for this purpose a cresol red indicator paper, turning a violet colour at a *p*H value of 8·6, is useful.

Where zinc (or galvanised) parts are to be in contact with the brine, however, it is better to limit its *p*H value to 8·6, which can be done by adding the alkali cautiously until the test paper changes colour.

Cold Storage and Ice Making.

TABLE XX.—SHOWS THE TEMPERATURES MOST SUITABLE FOR THE STORAGE OF VARIOUS COMMODITIES.

Commodity.	Temperature. ° F.	Commodity.	Temperature. ° F.
Apples	See special table	Honey	40
Bacon (storing)	35–40	Hops	32
,, (curing)	40	Ice	28
Bananas	55	Ice Cream (bulk storage)	15
Beef (fresh)	35	,, (bulk hardening)	5
,, (chilled)	30	,, (brick storage)	0
Beer	55	,, (brick hardening)	−10
,, (lager)	32	Lamb (frozen)	15
Butter	40	Margarine	35
,, (frozen)	15	Milk	40
Cheese	35–45	Mortuary (usually)	36
Cream	40	Mushrooms	36
Eggs (fresh)	32	Mutton (fresh)	36
,, (frozen)	10–15	,, (frozen)	15
Fish (fresh)	32	Pork (fresh)	34
,, (smoked or dried)	25	Poultry (fresh)	32
,, (frozen)	−5 or below	,, (frozen)	15
Flowers (cut)	40–45	Rabbits (fresh)	32
Furs	25–35	Vegetables (most)	36
Fruit (most)	32–36	Wines	45–50
Game (frozen)	15	Yeast	34 or higher

STORAGE OF APPLES AND PEARS. REFRIGERATED GAS STORAGE.

Apples.

It has been found that many varieties of apples when stored at a temperature a little above freezing point (say 34° F.) develop an internal breakdown—if not in the store, then shortly after removal from it; while storage at higher temperatures is not capable of prolonging the life of the fruit sufficiently.

This difficulty has been overcome, as a result of the researches of Dr. Kidd and Dr. West, by using somewhat higher temperatures than 34° F., combined with storage in an atmosphere containing a considerable percentage of carbon dioxide with either a corresponding or a greater

reduction in the percentage of oxygen. The conversion of oxygen into carbon dioxide is effected by the apples themselves, and in cases where it is desired to reduce the oxygen content by more than the percentage of carbon dioxide required (*i.e.* when the combined percentages of carbon dioxide and oxygen are to total less than the percentage of oxygen in the atmosphere, 21 per cent.), the excess carbon dioxide is removed by circulating the air through a scrubber containing caustic soda or milk of lime.

Table XXI shows the temperature and the percentage of carbon dioxide and oxygen required for the principal varieties of apples. The first four varieties in the list must not be 'gas-stored.' It may also be noted that the variety Worcester Pearmain is usually kept in an ordinary cold store at 34° F.

TABLE XXI.—STORAGE CONDITIONS FOR APPLES.

Variety.	Carbon Dioxide. Per cent.	Oxygen. Per cent.	Temperature. ° F.	Remarks.
Annie Elizabeth	—	21	34	Must not be 'gas-stored.'
Blenheim Orange	—	21	34	
King Pippin	—	21	34	
Newton Wonder	—	21	34	
Bramley's Seedling	8–10	13–11	40	$CO_2 + O_2$ percentage = percentage of O_2 in atmosphere (21 per cent.). No scrubbing required.
Lord Derby	8–10	13–11	40	
Stirling Castle	8–10	13–11	40	
Cox's Orange Pippin	5	2½	39–40	$CO_2 + O_2$ percentage less than that of O_2 in atmosphere; excess CO_2 must be removed by scrubbing.
Ellison's Orange	5	2½–5	39–40	
King Edward VII	5–10	2½	37–40	
Lane's Prince Albert	5	2½–5	39–40	
Laxton's Superb	10	2½	40	
Monarch	5	2½–5	39–40	
Worcester Pearmain	5	2½–5	34–35	

Pears.

The benefits of refrigerated gas storage as compared with ordinary cold storage are even more marked in the case of pears than in that of apples. Not only is the storage life greatly prolonged, especially if the percentage of oxygen is reduced, but the time for which the pears may be kept after removal from the store is approximately doubled.

TABLE XXII.—STORAGE OF PEARS—COMPARISON OF ORDINARY COLD STORAGE AND REFRIGERATED GAS STORAGE.

Variety.	Carbon Dioxide. Per cent.	Oxygen. Per cent.	Temperature. ° F.	Storage Life. Months.
Conference	—	21	34	3½
	10	11	34	7
	5	2½	34	10
Doyenne du Comice	—	21	37	3½
	10	2½	31·5	5
Williams	—	21	34	1½–2
Bon Chrétien	10	11	34	7

Ice Making.

The method most usually employed for making ice is by immersing galvanised cans, holding up to 2 or 3 cwts. each, in a tank of brine. The cans are of various standard weights, and taper slightly to enable the ice to slide out freely. The greatest thickness commonly used is 11 ins.

The cans are arranged in rows placed on frames so that a row can be lifted at a time. The tank contains an evaporator coil compartment, and the brine is circulated through this and between the cans by means of a propeller.

The brine temperature required depends on the thickness of ice and the freezing time allowed and is given approximately by the equation $t = \frac{7d^2}{32 - \theta}$, where t is the time in hours, d the maximum thickness in inches, and θ the brine temperature in ° F. This applies where the brine is circulated at 30 ft. per minute between the cans. Brine temperatures usually held are from 10° to 15° F. Lower temperatures involve a risk of cracking the ice ; the temperature at which this happens depends to some extent on the analysis of the water.

Ordinary water frozen in a block without special precautions yields ice which is opaque owing to the minute bubbles of dissolved air entrapped in it ; the usual way of producing clear ice is to keep the water in motion while it freezes by the injection of a stream of air. This can be done at a pressure of 15 to 30 lbs. per sq. in. gauge through nozzles in the bottoms of the cans, the air delivery pipe running underneath a row of cans (high pressure air agitation system) or at about 2 lb. per sq. in. through drop tubes inserted from the top into the centre of the cans. In this case the drop tubes are removed shortly before the ice is frozen. Either system requires roughly ¼ c.f.m. of free air per cwt. of ice made.

As the ice formed spreads inwards from the side of the can, the salts dissolved in the water concentrate in the centre. The central core may be removed and filled with treated water to produce a perfectly clear block of ice.

When a row of cans has been frozen, it is immersed in a thawing tank containing warm water to release the blocks from the cans. These are then placed in a frame which tips them so that the blocks slide out ; the empty row of cans is filled in one operation from a filling tank.

The refrigeration required in this country for a fairly large tank, insulated with 6 ins. of slab cork, with brine at 15° F. and using water at 65° F., is about 215 B.Th.U.'s for each pound of ice.

Several special methods for making ice otherwise than in cans also exist.

In the ' Flakice ' system, ice is frozen in a thin layer on a revolving drum which dips into water and is refrigerated internally by brine. The ice is scraped off automatically.

Another American system, the ' Pakice ', uses a corrugated cylinder refrigerated externally, the ice being cut off continuously as it is formed. The crystals produced may be cemented into briquettes.

Fairly clear ice is produced in small quantities for use in drinks by freezing water in trays fitted with spacing grids so that the ice obtained is in cube form. The trays are placed in a unit comprising a series of shelves refrigerated by direct expansion. Ice 1¼ in. thick should be frozen in about 3 hours with evaporation at 5° to 10° F.

Insulation.

The thermal conductivities given in Table XXIII are laboratory values, and when calculating the heat leakage into a cold store, a liberal margin should be added to them to allow for imperfections in construction and the use of wood or other framework. With the usual insulating materials and methods of construction, the allowance required is usually from 30 to 50 per cent.

For insulation on board ship, a special allowance for the presence of steel frames, etc., must be made. Usually a value of about 0·07 B.Th.U.'s per hour per sq. ft. for a thickness of 10 ins. is in satisfactory agreement with results.

The calculation of heat flow through a compound wall is facilitated by the resistance concept —an analogy with Ohm's Law in electricity.

$$Q = \frac{\Delta t}{R_1 + R_2 + R_3 + \dots}$$

where Q = heat flowing in B.Th.U.'s per sq. ft. of insulation per hour, Δt = temperature difference between the two sides of the wall, R_1, R_2, etc., are the resistances of the several layers. The resistance is the reciprocal of the conductance, *i.e.* $R = \frac{d}{K}$, where d = thickness in inches and K = thermal conductivity in B.Th.U.'s per hour per sq. ft. per ° F. for 1 in. thickness (or d may be in feet and K expressed for 1 ft. thickness).

Example : Calculate the heat flowing through a 10-in. concrete wall insulated with 2-in. slab cork and lined with 1-in. deal. External temperature 70° F. ; internal temperature 50° F.

$$R_1 \text{ (concrete)} = \frac{10}{6\cdot7} = 1\cdot49$$

$$R_2 \text{ (cork slab)} = \frac{2}{0\cdot35} = 5\cdot72$$

$$R_3 \text{ (wood)} = \frac{1}{1\cdot1} = 0\cdot91$$

$$R_1 + R_2 + R_3 = 8\cdot12$$

$$Q = \frac{20}{8\cdot12} = 2\cdot46 \text{ B.Th.U.'s per hour per sq. ft.}$$

(Note that in this example the tabulated K of 0·27 for cork slab has been increased by 30 per cent. as explained above.)

For accuracy, surface resistances would have to be taken into account; but in most cases they are comparatively small, and it is usual to neglect them, the estimate of heat leakage being in consequence somewhat on the safe side. In the case of thin partitions, especially if made of materials of low insulating value, surface resistances should be included in the calculation.

It is good practice to base the calculation of heat load on the outside surface of a chamber; this again gives a small margin of safety.

The usual thickness of insulation used in this country—based on a calculation of the cost of the insulation and the refrigerating machine, and of the cost of running the latter—is about as follows:—

Temperature, ° F.	*Thickness, Ins.*
30 to 40	4
20 to 30	5
0 to 20	6
− 20 to 0	8

Cork slab is the most commonly used insulating material, combining the virtues of low thermal conductivity, ease of handling and permanence.

For special applications in which it is important to cool a chamber as quickly as possible, an insulating material of lower heat content is required, such as glass wool, Kapok, or one of the plastics.

TABLE XXIII.—THERMAL CONDUCTIVITY OF INSULATING MATERIALS AT 50 TO 60° F. (B.Th.U.'S PER HOUR PER SQ. FT. PER ° F. FOR 1 IN. THICKNESS.)

Material.	Density. Lb. per cu. ft.	K.
Aluminium foil (multiple layers, spaced)	3	0·25
Asbestos, tightly packed	44	1·6
„ loosely „	30	1·1
„ cement sheeting	96	2
Balsa wood	6–10	0·33–0·4
Brick, light	—	5
„ heavy	—	9
Cane fibre board (celotex)	13·5	0·4
Cellular materials:		
Concrete	16–20	0·5–0·6
Ebonite (onazote)	3·5–5·5	0·18–0·28
Phenol-formaldehyde plastic	5	0·2
Polystyrene plastic	3–6	0·17–0·34
Rubber (soft)	7–14	0·27–0·38
Concrete (gravel aggregate)	146	6·7
*Cork slab	8	0·27
„ granulated	5·5–7·5	0·3–0·35
Glass:		
†Glass window pane: Single	—	1·13
† „ „ „ Double	—	0·53
† „ „ „ Triple	—	0·42
† „ „ „ Quad.	—	0·35
Glass silk (fibres 1–10,000 in. dia., not felted)	3–9	0·22–0·28
„ wool (' fibreglass ')	—	0·3
Kapok	0·5–1	0·23–0·26
Kieselguhr	30	0·55–0·62
Plaster board	60	1·1
Slag wool	15	0·3
Soil, clay, 14 per cent. moisture:		
Loosely packed	—	2·6
Loaded at 1 cwt. per sq. ft.	—	5·0
„ „ 1 ton per sq. ft.	—	8·3
Loam over sand and gravel 3 ft. deep (varies with season)	—	7·5–11
Wood, across grain:		
Deal	—	1·1
Oak	—	1·1
Pitch pine	—	1·05
Spruce	—	0·75–1
Teak	—	0·8
Wood, three-ply	—	0·95

* Cemented with natural binder under heat and pressure. Typical figures.
† For the window, not for 1 in. thickness.

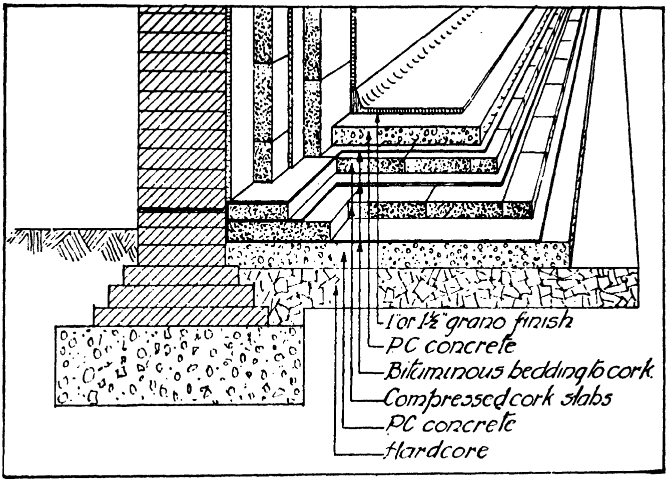

1" or 1½" grano finish
PC concrete
Bituminous bedding to cork
Compressed cork slabs
PC concrete
Hardcore

FIG. 3.

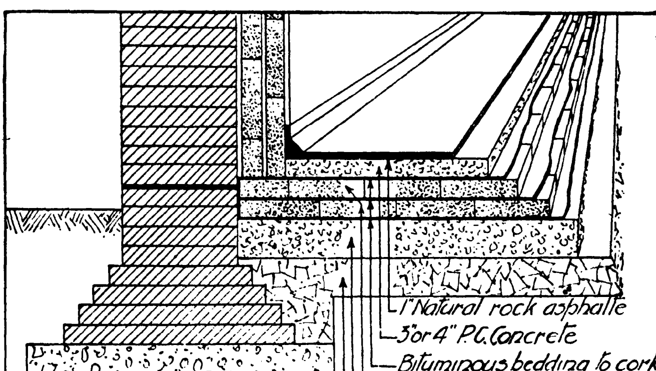

1" Natural rock asphalte
3" or 4" P.C. Concrete
Bituminous bedding to cork
Compressed cork slabs
P.C. Concrete
Hardcore

FIG. 4.

INSULATION ON BOARD SHIP.

The insulation of cold chambers on board ship follows the same principles and method that apply to insulation work as carried out on land.

Owing to the way in which the ship's side and deck beams break up the surface, and to the absence of pure rectangular spaces, cork slabs are not so simple to use as loose filling materials. The use of cork slabs is extending.

Machinery.

COMPRESSORS.

Single stage compression is usual in this country for temperatures down to about $-15°$ or $-20°$ F., although Continental practice tends towards the use of compound compression for considerably higher temperatures than this.

The reciprocating type of compressor is used in the overwhelming majority of cases, the exceptions being the occasional employment of a multistage centrifugal compressor, usually for large duties and using a refrigerant of high specific volume, and the small rotary compressor sometimes used for domestic refrigerators.

Most compressors for ammonia, methyl chloride or freon are made single acting, and are of enclosed construction, with a crankcase at suction pressure and a gland of packed type or with smooth metallic rubbing faces (a carbon face is used in one pattern) fitted to the crankshaft. Cylinders and crankcase are generally made of cast iron.

Lubrication, except in quite small compressors of, say, 12 h.p. or less, is by a pump which circulates oil from the crankcase to the bearing surfaces including the gland. Preferably, though not usually, the oil supply to the cylinders should be independent.

Frequently, the compressors are of uniflow design, having trunk type pistons and, usually, automatically operating ring valves, the suction valves being fitted on the top of the piston. A safety head held down by a heavy spring is almost universally used; this carries the delivery valves and is capable of lifting and giving a greatly increased flow area, should it be subjected to an exceptionally high pressure. The risk of damage to the compressor in the event of a quantity of liquid refrigerant or oil entering the cylinder is thus greatly reduced. The provision of a safety head is particularly important in automatically operated compressors.

Vertical compressors usually have from 2 to 4 cylinders; a greater number, up to 16, may be arranged radially in some modern designs.

Cylinders of ammonia compressors are frequently water-jacketed, although the reduction in delivery temperatures thereby produced is not great. A water jacket is quite unnecessary in the case of compressors used for Freon-12.

The trunk pistons are usually made of cast iron, and except in small sizes (about 2 in. diameter downwards) are fitted with piston rings and, in the lower part, oil scraper rings.

Small compressors are frequently fitted with suction valves consisting of thin flexible strips of metal, with delivery valves either of similar design or of disc type.

Pistons in these small machines may be ringless, with benefit to the power absorbed, provided that the finish of pistons and cylinder walls is sufficiently fine, and that the clearance between piston and cylinder is not too great (it should not appreciably exceed 0·00025 in. per in. of diameter).

Carbon dioxide compressors are machined from high carbon steel forgings, owing to the high working pressures; stop-valves and other fittings are also machined from the solid. These compressors are often made double acting. The compressor rod gland is provided with a lantern supplied with oil at a pressure rather higher than the compressor delivery pressure. This is done by a special lubricator consisting of a cylinder with piston, oil being pumped by a hand pump (via a check valve) into the cylinder on the rod side of the piston and delivered to the lantern from this same side. The other side of the piston is subject to the CO_2 delivery pressure. Since the piston is balanced between the two forces, and the area on the rod side is the smaller, it follows that the pressure on this side is greater.

CONDENSERS.

The following types of condensers are used:

1. *Water-cooled.*

(*a*) *Submerged.*—This type, consisting of coils of piping contained in a tank through which water flows, is obsolescent, being replaced by:

(*b*) *Horizontal shell and tube.*—This widely used design consists of a cylindrical shell in which the refrigerant condenses on a number of tubes through which the water flows in one or more passes. The tubes are fixed to the end plates by expanding, welding, or brazing. Several such shells may be used if required. This type of condenser has taken the place of (*a*) owing to the much higher heat transfer coefficients obtainable and the consequent gain in compactness. Typical figures obtainable in practice for a water velocity of 4 ft. per second (which is normal) are:—

Ammonia =	215 B.Th.U.'s per	hour	per sq. ft.	per ° F.	
Methyl chloride =	160	,,	,,	,,	,,
Freon-12 =	135	,,	,,	,,	,,

(c) *Vertical shell and tube.*—In this case the water flows down the inner surface of the tubes in a single pass, without filling them. Its main advantage is probably economy of floor space.

With regard to the water quantity required for shell and tube condensers, this is usually calculated to give a rise of about 13° F. There is no doubt, however, that while this is good practice where water is re-circulated over a cooling tower, the use of considerably smaller quantities of water will result in economy of operation at normal costs of power and water where the latter is run to waste.

(d) *Double pipe condensers*, with water flowing through the inner pipe and refrigerant condensing in the annulus. These are now little used, being largely superseded by (b).

2. *Evaporative.*

(a) *Atmospheric.*—The type of condenser usually called by this name consists of piping arranged in vertical stacks, water being re-circulated over these from a tank beneath. They take up considerable space, and must be placed in a position exposed to the atmosphere but protected from the wind. The heat removed in a condenser of this type is used in evaporating the water, each pound of which is therefore capable of providing about 1,050 B.Th.U.'s cooling effect, so that the water consumption is only a few per cent. of that necessary for an ordinary water-cooled condenser. The condensation temperature obtained is, however, considerably higher and depends on the wet bulb temperature of the air.

(b) *Forced draught.*—This type of condenser is basically similar to (a), but provides more effective evaporative cooling by incorporating forced circulation of air over the pipes. Less surface is required, since the rate of diffusion of water vapour into the air is increased. The quantity of air to be circulated may be reckoned at about 1 cu. ft. per minute for every 50 B.Th.U.'s per hour of condenser duty to be removed.

Space may be further economised by the use of gilled instead of plain piping.

3. *Air-cooled.*

This type is made only in small sizes, as otherwise it becomes unduly large and also requires an unreasonably large volume of air. For small automatic machines up to 3 or 4 h.p., however, it is almost universal. It consists of one or more stacks of gilled tubing connected by headers at top and bottom; air is circulated over them by a propeller type fan. A liquid receiver is fitted.

EVAPORATORS.

These are of two main types : (a) air coolers, and (b) coolers of brine or water.

(a) When the evaporator cools air directly, the plant is said to operate on the direct expansion system.

In some cases, piping in the form of grids is arranged on the walls and/or ceiling of the room to be cooled. This arrangement is used mainly in cases where the heat load to be dealt with consists almost entirely of that passing through the insulation, and no appreciable cooling of the commodities stored is involved. It has the disadvantage of using a large quantity of piping, and defrosting is liable to be a rather messy operation.

A better method of cooling a store is in most cases to use a cooler consisting of a number of grids arranged in the form of a battery over which air is circulated by means of a fan. Air flow should be at right-angles to the pipes, in which case a heat transfer coefficient about three times that found with natural air circulation can be obtained. Except for quite small rooms, it is usual to arrange ducting to distribute the air fairly uniformly ; but it is also possible to dispense with the use of ducting in quite large rooms by installing one or more coolers each having above it a series of fan runners mounted on a horizontal shaft and projecting a stream of air across the ceiling, the air returning to the cooler near the floor.

In cases where the evaporation temperature is above 32° F., (as in air conditioning applications) the air cooler is frequently made of gilled tubing, resulting in a great saving in space occupied, and in refrigerant charge. This type of cooler is also sometimes used at lower temperatures where the stopping periods of the plant can be sufficiently numerous and extended to allow defrosting to take place.

A ' wet ' or brine spray cooler is suitable for use when the evaporation temperature is below 32° F. and considerable quantities of moisture have to be removed from the air. In this type of apparatus brine is recirculated through sprays and over the cooler, which not only prevents the formation of frost but also increases the effective cooling surface and so makes it possible to reduce the amount of piping by 50 per cent. or more.

The disadvantage is the necessity of continuously or periodically re-concentrating the brine in consequence of its dilution by moisture condensed from the air.

(b) The most usual type of evaporator for cooling brine or water to-day is the horizontal shell and tube pattern, very similar in design to the corresponding condenser, but usually fitted with a vertical branch of large diameter from which the vapour is drawn off, and which keeps the velocity of the vapour sufficiently low to avoid entrainment of liquid. Regulation of a plant fitted with such an evaporator may be effected by either a high pressure or a low pressure float regulator.

Submerged coil evaporators of simple design are also still used to some extent.

Another form of evaporator useful when considerable storage of water or brine is required consists of a rectangular tank containing rows of vertical tubes connected to headers at their upper and lower ends. The water or brine is circulated across the tubes at about 1 ft. per second.

When water has to be cooled to a temperature very near to its freezing point, a shell and tube cooler is not suitable, because of the risk of freezing the water in the tubes and bursting them. In such a case a cooler of the Baudelot type may be used; this is a surface cooler consisting of vertical coils over which the water is circulated at the rate of 1 to 2 galls. per minute per foot of pipe in the uppermost tier.

TEST PRESSURES.

Commonly used test pressures for the various parts of refrigerating plant are as follows:—

	Hydraulic. Lb. per sq. in.	Air Lb. per sq. in.
Carbon Dioxide (all parts)	3,000	1,500
Ammonia:		
Cylinders, and other parts subject to condenser pressure	600	350
Crankcases, etc.	300	175
Condenser and evaporator coils	1,500	500
Shell and tube condensers and evaporators	500	300
Methyl Chloride and *Freon*-12:		
Cylinders, and other parts subject to condenser pressure	350	200
Crankcases, etc.	200	150
Condensers and evaporators	350	200

Heat Transfer.

The importance of obtaining high coefficients of heat transfer lies in the fact that to obtain the best performance from the machine its evaporation and condensation temperatures must be kept as close as possible to those of the substance being cooled and the cooling medium respectively.

Most cases of heat transfer met with in refrigeration work are concerned with a pipe having air, water or brine on the outside and either (*a*) evaporating or condensing refrigerant, or (*b*) water or brine used as a cooling medium, on the inside. The total resistance to heat transfer is made up of fluid film resistances on the inner and outer surfaces, resistances of dirt and scale films, and resistance of metal, and the relative values of these resistances must be considered in assessing the effect of any change in conditions. In many cases the effect of the metal resistance is negligible, but this may not be so where the film coefficients of heat transfer are high, as in some condensers. Again, in air coolers or air-cooled condensers, the internal resistance, though not negligible, is usually so much less than that on the air side that the latter has by far the greater influence. Thus, although the film coefficient of condensing Freon-12 is much lower than that for methyl chloride, it is found that there is little difference in overall heat transfer in an *air-cooled* condenser for either of these two refrigerants. In a water-cooled condenser, on the other hand, the resistances of the water side and the refrigerant side are comparable, and the overall figure is therefore much lower in the case of Freon-12.

AIR COOLERS.

In cooling and dehumidifying air by means of a refrigerated surface, the sensible heat removed from the air has to be transferred through a film of non-turbulent air, the resistance of which governs the rate of heat transfer. The condensation of water vapour to liquid form, however, takes place only at the surface of the water film, not in the air itself, and therefore meets with no appreciable resistance. Hence it is correct to design the cooler surface on the basis of sensible heat transfer only. The proportion of the total duty that is sensible heat varies with the circumstances, but for normal cold store work probably averages about 80 per cent. In some air-conditioning applications it may be much less. The figures for air coolers given in Table XXIV apply to sensible heat transfer.

Once the sensible heat removed by a given cooler is known, the dehumidification effected can be calculated by the aid of two principles: (1) the ratio of moisture removal to temperature reduction remains constant as the air passes through the cooler, provided that the surface temperature is constant, which may be taken to be the case with a direct expansion cooler of plain pipe; (2) if the cooler surface were indefinitely prolonged, the air would eventually reach the surface temperature and would then be saturated.

In the case of extended surface (gilled) coolers, however, the problem is complicated by the fact that the surface temperature is considerably higher than that of the refrigerant, and varies in the direction of the air flow across the cooler.

TEMPERATURE DIFFERENCE.

When sensible heat is transferred, the temperature difference between the donor and the recipient of the heat is not in general the same in different parts of the apparatus. In this case it

is usual to use the logarithmic mean temperature difference for calculations; *i.e.* if D and d are the temperature differences at inlet and outlet (not necessarily respectively, but D representing whichever is the greater) -

$$\text{Logarithmic M.T.D.} = \frac{D - d}{2 \cdot 303 \log_{10} D/d}.$$

This is always less than the arithmetical T.D., which should not therefore be used if D is more than say $1 \cdot 3\, d$; for lower ratios of $\frac{D}{d}$ the difference between the two T.D.'s may be neglected for practical purposes. The use of the Log. M.T.D. is correct only if the specific heats are constant with the temperature variation involved, and if the coefficient of heat transfer is also constant. These assumptions are sufficiently close to the truth to be warrantable in most cases, but where they are not, the calculation must be broken down into suitable stages.

TABLE XXIV.—SOME TYPICAL HEAT TRANSFER COEFFICIENTS.

Air Coolers (Cross Flow).	Max. Air Velocity, ft. per sec.			Notes.
	10	12	14	
No snow	6·7	7·5	8·25	Based on External surface.
Considerably frosted	4·5	4·9	5·1	,, ,, ,,
With brine sprays	14	15	16	,, ,, ,,
Grids (natural air circulation)	1·75 to 2·0 (plain)			,, ,, ,,
	1·25 to 1·5 (gilled)			,, ,, ,,
	Brine or Water Cooling.			
Coil in stagnant water	35			Based on mean surface.
Coil in stagnant brine (0° F.)	24			,, ,, ,,
	Gall. per min. water per ft. of top tier.			
	1	1·5	2	
Baudelot (direct expansion) cooling water	65	95	120	,, ,, ,,
	Max. water velocity, ft. per sec.			
	$\frac{2}{3}$	1	$1\frac{1}{3}$	
Vertical tube cooling water	72	86	100	,, ,, ,,
Horizontal shell and tube.	Velocity, ft. per sec.			
	2	4	6	
Water	90	118	132	,, ,, ,,
Brine (0° F.)	60	90	110	,, ,, ,,

EFFECT OF VARIATION IN VELOCITY.

In the great majority of the cases of heat transfer dealt with by the refrigerating engineer, the flow of the fluids concerned is turbulent; in such cases the surface coefficient of heat transfer varies with a power of the velocity, *i.e.* (approximately) $h \propto v^{.6}$ for fluids flowing at right angles to pipes and $h \propto v^{.8}$ for fluids flowing along or through pipes. It must be borne in mind that such variations apply only to the particular film coefficient concerned, not to the overall coefficient.

EFFECT OF VARIATION IN PIPE DIAMETER.

For flow across pipes, $h \propto \frac{1}{d^{.4}}$ approximately; for flow through pipes, $h \propto \frac{1}{d^{.2}}$.

EFFECT OF VARIATIONS IN PROPERTIES OF FUIDS.

(*a*) *Viscosity.*—For flow across pipes, $h \propto \frac{1}{\mu^{.3}}$ approximately; for flow through pipes, $h \propto \frac{1}{\mu^{.45}}$ approximately.

(*b*) *Density.*—Flow across pipes, $h \propto \rho^{.6}$. Flow through pipes, $h \propto \rho^{8}$.

When considering the effect of changes in density, as in the case of air at various pressures, it is helpful to remember that the heat transfer coefficient actually depends on mass velocity, *i.e.* on the product ($v\rho$).

Automatic Refrigerating Machines.

Domestic refrigerators working on the vapour compression system are now always automatically controlled, as are the great majority of small machines (up to 4 or 5 h.p.) used for commercial and industrial applications.

Ammonia has been almost entirely replaced for these small plants by methyl chloride or Freon-12. Sulphur dioxide was used to a considerable extent not long ago but is now seldom met with.

The compressor has most usually 2 or 3 cylinders, and presents no special features of design as compared with larger ammonia machines, other than simplicity of construction (an oil circulating pump is not used, lubrication being mainly by splash, but in some models helical grooves cut in the crankshaft convey the oil to the bearing surfaces) and the universal use of a packless crankshaft gland. The latter varies in design, but its essentials comprise a rubber, composition, or white metal sleeve fitting tightly to the crankshaft to prevent leakage along the latter, and two rubbing faces, one fixed to the gland cover and the other rotating with the shaft. The rubbing faces may be of bronze and steel, or one may be of carbon. A spring is used to press them together, and some measure of flexibility is essential to ensure perfect contact. This may be obtained by a metal bellows fixed to the cover at one end and to one rubbing member at the other, the spring being between the two end pieces, or the rubber sleeve may be prolonged in the form of a bellows ending in the rotating member, one end of the spring being in contact with this assembly and the other with a collar on the shaft.

Usually no oil separator is fitted, the oil circulating through the system and returning again to the crankcase; the velocity of the vapour in the evaporator and suction line must be sufficient to carry the oil through. In most cases the direct expansion system is used, but it is not advisable to try to obtain accurate temperature control of more than two or three chambers on this system using only one compressor.

REGULATION AND CONTROLS.

A thermostatic regulator is generally used, its function being to ensure that the vapour leaving the evaporator is slightly superheated. This results from the fact that the needle valve is balanced between a force exerted by the pressure at the evaporator outlet and one corresponding to the temperature there, the latter being produced by a suitable fluid—usually that with which the system is charged—contained in a phial attached to the suction line. Should several evaporators be connected to the same machine, each will have its own thermostatic regulator.

A single cold chamber or other unit is controlled by a thermostat, stopping and starting the machine according to the temperature; the usual difference between starting and stopping temperatures in a commercial instrument is about 4° F. Another method often used in the U.S.A. is to control by a pressure switch ('pressostat'); variations in evaporation temperature (and thus pressure) following those of chamber temperature, it is possible to set the pressure switch so that the chamber temperature is maintained between required limits, at least until some considerable change in external conditions occurs. This method is not much favoured in Britain, where the 'pressostat' is used mainly as a safety device to stop the compressor should unduly low evaporation temperatures be reached.

Where two or three units have to be automatically controlled from one machine, this control can be effected by fitting a solenoid-operated stop-valve in the liquid line leading to each regulator. The controls are wired so that each thermostat is capable of starting the machine and opening the liquid stop-valve corresponding to it own chamber; when all thermostats are 'cut out,' the machine stops.

Should two units of widely different temperatures have to be dealt with by one machine, it is necessary, in addition to the two regulators, solenoid valves, and thermostats, to provide a constant back pressure valve in the suction line from the higher temperature unit and a check valve in that from the lower temperature unit. The back pressure valve prevents the temperature of the evaporator to which it is connected from being undesirably low, while the check valve prevents the charge of refrigerant in the higher temperature evaporator from passing into the lower temperature evaporator and thereby warming it during the stopping periods of the machine; in the absence of the check-valve this would happen as a result of the difference in pressure existing.

Most small automatic plants (other than those installed on board ship) have air-cooled condensers; but when a water-cooled condenser is fitted, a pressure switch arranged to cut out on rise of pressure and connected to the delivery side of the compressor should be fitted, so as to stop the machine should the failure of the water supply give rise to an unduly high pressure. Such a safety device is often omitted in the case of an air-cooled condenser with a fan driven by the compressor motor; but should the condenser be remote from the compressor and have its own fan and motor, stoppage of the latter would lead to excessive pressure, which should be guarded against by the provision of a pressure cut-out switch.

When a water-cooled condenser is used with an automatic plant, an automatic water valve to control the water supply is also fitted. This may be simply a solenoid-operated valve, opening when the machine starts and closing when it stops; or it may be a modulating valve operated by the condenser pressure, opening when the latter reaches a given value.

Automatic control is now often used in conjunction with quite large compressors. In such cases it is important that the plant shall be so arranged that there is no chance of liquid being drawn into the compressor, as it is dangerous for this to be allowed to happen frequently, even though safety heads are fitted. This is ensured by what is known as the flooded or dry compression system, which also provides an excellent heat transfer coefficient between refrigerant and pipe wall. The essential feature of this system is that the refrigerant circulates through the coils at a rate several times as fast as that at which it is evaporated. The coils are fed with liquid refrigerant from a reservoir vessel, a mixture of liquid and vapour returning to the latter from the coils. The circulation may be by gravity only, being ensured by the buoyancy of the vapour formed; or in the case of long leads from the vessel to the coils being involved, a pump may be used. In either case, the vessel, besides feeding the coils, acts in its upper part as a liquid separator, the velocity of the vapour here being low enough to ensure that drops of liquid will not be entrained.

The flooded system is widely used on large plants, whether automatic or not.

Electrolux System.

This, used for domestic refrigerators, is a special application of the absorption cycle. In the basic form of the latter, the liquid refrigerant, usually ammonia, absorbs heat in an evaporator and is converted into vapour just as in a compression system. After leaving the evaporator, however, it enters an absorber, where it is dissolved in water. The strong solution is pumped to a generator at high pressure, in which by the application of heat the ammonia vapour is expelled, leaving a weak solution which returns via a pressure reducing valve to the absorber. The expelled vapour is liquefied in a condenser and returns through a regulating valve to the evaporator.

The Electrolux system is fitted with several auxiliary devices to improve efficiency, but its most important special feature is that it has no expansion valve and no circulating pump, and the pressure in all parts of the system is approximately the same. This is achieved by taking advantage of the fact that the pressure of the vapour of a liquid is independent of the presence or absence of a gas, non-condensible at the pressure and temperature prevailing, mixed with it. The evaporator of the Electrolux system contains hydrogen (prevented from entering the condenser by a liquid seal) and since the total pressure in evaporator and condenser is the same, the pressure of the ammonia in the evaporator is much lower, and the required evaporation temperature is thus obtained. The circulation of ammonia vapour and hydrogen to the absorber, and the return of hydrogen to the evaporator, are produced automatically by virtue of the difference in densities.

The weak solution of ammonia is transferred from the generator to the absorber as follows: the bubbles of vapour liberated by the action of heat (which may be by means of gas or electricity) carry the solution up to a separating vessel, from which the vapour passes to the condenser while the solution flows by gravity to the absorber.

Rating, Testing and Operation of Refrigeration Plant.

Rating.

The refrigeration produced by a plant is often stated in Tons Refrigeration. The Ton Refrigeration is the number of B.Th.U.'s which must be removed from 1 ton of water at 32° F. in order to convert it into ice at that temperature. In nearly all cases the U.S.A. ton of 2,000 lb. is referred to, and the latent heat of freezing of water being 144 B.Th.U.'s per lb., 1 ton of refrigeration is equivalent to $\frac{2,000 \times 144}{24}$ or 12,000 B.Th.U.'s per hour.

The refrigerating duty of a given machine may be quoted in tons under any conditions of evaporation and condensation, provided that these are clearly specified. There is, however, a

standard set of conditions, *i.e.* evaporation 5° F. (− 15° C.) and condensation 86° F. (30° C.); these are known as Standard Ton Rating conditions, and it is usually at these conditions that different refrigerants, for example, are compared. For completeness, the temperatures of the liquid leaving the condenser should also be stated, and it is important that this should be done in the case of carbon dioxide.

Lloyd's definition of the Ton Refrigeration is similar to that given above, except that the English ton of 2,240 lb. is used, so that the ton refrigeration is equivalent to $\frac{2,240 \times 144}{24}$ or 13,440 B.Th.U.'s per hour.

It should be borne in mind that the duty specified as obtainable at stated evaporation and condensation temperatures defines the output of the compressor only. The evaporation and condensation temperatures actually obtaining in practice, and hence the refrigerating duty produced, will depend not only on the temperature of the air or other substance being cooled and on that of the cooling water, but also on the evaporator and condenser surfaces provided and the coefficients of heat transfer obtainable from them.

Testing.

There are several ways of ascertaining the refrigerating duty obtainable from a given compressor, of which these may be mentioned :

(1) A brine cooling evaporator may be used and the rate of flow of the brine and the drop in temperature measured (see Table XVIII for heat capacity of brine). For equilibrium conditions, heat must be put into the brine in another part of the circuit, *e.g.* by an electrical heater or by steam-heated coils, at the same rate as that at which it is removed, and by measuring the heat input here, a check on the refrigerating duty may be obtained.

(2) The brine may be kept stationary instead of being circulated, and the rate of heat input adjusted until the brine temperature remains constant. The heat input being then measured by electrical instruments in the case of electric heaters, or by measuring the rate of condensation in the case of steam heaters, gives the refrigerating duty. The disadvantage of this method is that there is no independent check on the result.

(3) The rate of flow of liquid refrigerant arriving at the regulator may be measured by one of several methods, and the refrigerating duty then calculated by the aid of the enthalpy tables.

Brake horse-power may be ascertained by measuring the electrical input to the motor, provided that the efficiency of the latter is precisely known. It is better, however, especially in the case of small compressors, to use a dynamometer.

Operation.

Before a plant is put into operation, it should be tested for leaks under air pressure, and then evacuated of air, using the compressor as a pump for this purpose. The refrigerant is then charged into the inlet side of the evaporator, the charge being completed while the plant is actually running.

Before starting a non-automatic plant, care should be taken to make sure that the delivery stop-valve is open, that the oil level in the compressor is correct, and that the gauge valves are open ; also that the water is running through the condenser, and the cylinder jackets if any, and that the air, water or brine to be cooled is in circulation. In some cases a by-pass between the suction and delivery sides of the compressor is provided so that the motor does not have to start under load.

In judging the running of the machine, the most important indications are those given by the gauges. These are marked, in addition to the pressure scales, with scales showing the corresponding saturation temperatures for the refrigerant in use ; these scales show the actual temperatures of evaporation and condensation respectively (except in so far as the pressures at the gauge may differ from those existing in the evaporator and condenser as a result of pressure drop in the pipe lines).

The object of the operator of the plant is at all times to have the evaporator gauge reading as high as it can be consistent with the avoidance of liquid being drawn into the compressor, for on this depends the production of the greatest possible refrigerating effect ; and the condenser gauge as low as possible consistent with the presence of a sufficient charge of refrigerant and with the avoidance of an unduly high consumption of water, for in this way the power absorbed will be kept to a minuimm.

Regulation and refrigerant charge are to a considerable extent interdependent, and both must be correct.

For a plant working at an evaporation temperature below 32° F., the rule of thumb method of regulating, so that the frost line on the suction pipe extends up to but not on to the compressor, is excellent. A more scientific method of regulation is in accordance with the temperature of the refrigerant in the suction line. If this is equal to the evaporator gauge reading, it is a virtual certainty that unevaporated liquid is returning to the compressor, and the regulator should therefore be closed in until several degrees of superheat—usually at least 10° F.—are indicated by the thermometer. If there are several evaporator circuits to be regulated, the suction temperature method is the only one which will ensure maximum efficiency. These remarks apply where hand regulators are fitted ; float regulators are automatic in operation. Thermostatic regulators are

also automatic, but may require careful initial setting to provide a superheat suitable to the particular conditions.

Refrigerant charge must be sufficient to ensure that the refrigerant arriving at the regulator is entirely in liquid form. An excellent way to ascertain whether this is so is by the use of a sight glass in the liquid line: it should be completely full of liquid. Otherwise, it is often difficult to be sure that a plant is fully charged; rule of thumb methods, such as that the condenser gauge should be a given number of degrees higher than the water inlet or outlet temperature, are of limited application.

Where a condenser capable of providing some sub-cooling of the liquid—*e.g.* a horizontal shell and tube condenser with some tubes near the bottom of the shell—is fitted, a thermometer indicating the temperature of the liquid leaving the condenser will give a good indication; if its reading is slightly below the condenser gauge, the plant is sufficiently charged.

Of course, if the duty of the plant can be measured and proves to be equal to the known capacity at the working conditions, that in itself is sufficient proof that the charge is adequate.

In general, excess of charge will do no harm until the point is reached at which an appreciable proportion of the condenser surface is submerged by liquid and therefore unable to fulfil its function, in which case the condenser gauge will be higher than it should be. If a liquid receiver is fitted to the condenser, it is possible to run with a charge considerably greater than the minimum without ill effect. It is always better to work with slightly too much than with too little refrigerant.

If the regulator is of the high pressure float type, however, almost the whole charge must be in the evaporator, and the amount is, therefore, rather critical; if it is insufficient the evaporator gauge will be abnormally low and the suction superheat high, while if it is too great the machine will run ' wet,' *i.e.* unevaporated liquid will reach the compressor.

If a plant works under fairly constant temperature conditions, and the condenser gauge reading for correct working has been ascertained, the indication of this gauge will in future be a useful guide. If it is abnormally low, and the plant is not ' running wet,' there is insufficient charge. An abnormally high condenser gauge may be due to excess of charge, or on the other hand it may be caused by the presence of ' non-condensible ' gas, usually air, in the system. This may occur through leakage in at the gland if the evaporation pressure is below that of the atmosphere. It is remedied by purging, *i.e.* allowing gas to escape through a connection at the top of the condenser. It must be borne in mind that air cannot be expelled without loss of refrigerant vapour also. This loss can be minimised by first allowing the mixture of vapour and air from the condenser to pass into a purging vessel, where by the use of a coil at the evaporation temperature some of the vapour is condensed, leaving a mixture with a higher proportion of air.

Apart from maintaining the correct charge of refrigerant and correct regulation, and keeping the system free from air, operation will include attention to the oil level in the crankcase (oil separators may be provided with automatic means of returning oil to the crankcase, or this may have to be done by hand) and to the compressor gland. A suitable oil for use in the crankcase will be specified by the makers of the compressor, and it is essential to adhere to their recommendations—the oil must have sufficient ' oiliness,' yet not freeze in the evaporator.

Refrigeration on Board Ship.

The use of refrigerating machinery on board ship may be roughly subdivided into four main groups:—

(1) Small plants to cool provision rooms and to make small quantities of ice.

(2) Large passenger vessels with large provision rooms and numbers of cold cupboards throughout the ship and perhaps some refrigerated cargo.

(3) Large vessels used primarily for refrigerated cargo.

(4) Air conditioning.

CO_2 refrigerating machines are largely used for all classes and sizes of machines and evaporative liquid cooling of the CO_2 is sometimes employed as it results in economy in working particularly when the condenser water temperature is high.

In Classes (1) and (3), the machines are usually driven by an electric motor.

In Classes (2) and (4), most of the older steam-driven ships have the main refrigerating machinery driven by a compound tandem steam engine which incorporates its own steam condenser in the machine base. In motor-driven ships the drive is either by an electric motor or occasionally by a diesel engine direct coupled to the crankshaft of the refrigerating machine or through speed-reducing gears. The almost complete electrification of modern ships, whether driven by steam turbines or oil engines calls for a more compact design of refrigerating machinery which is electrically driven. Some of the larger ships have an insulated capacity of upwards of 600,000 cu. ft. and a typical arrangement of the refrigerating machinery is shown in fig. 8 (p. 747).

Cooling of Chambers.

It is usual to cool the chambers by means of cold brine circulated through 1¼-in. brine pipes. These pipes can be arranged on the roof and sides of the cold chambers, and this is the system used on most of the old vessels.

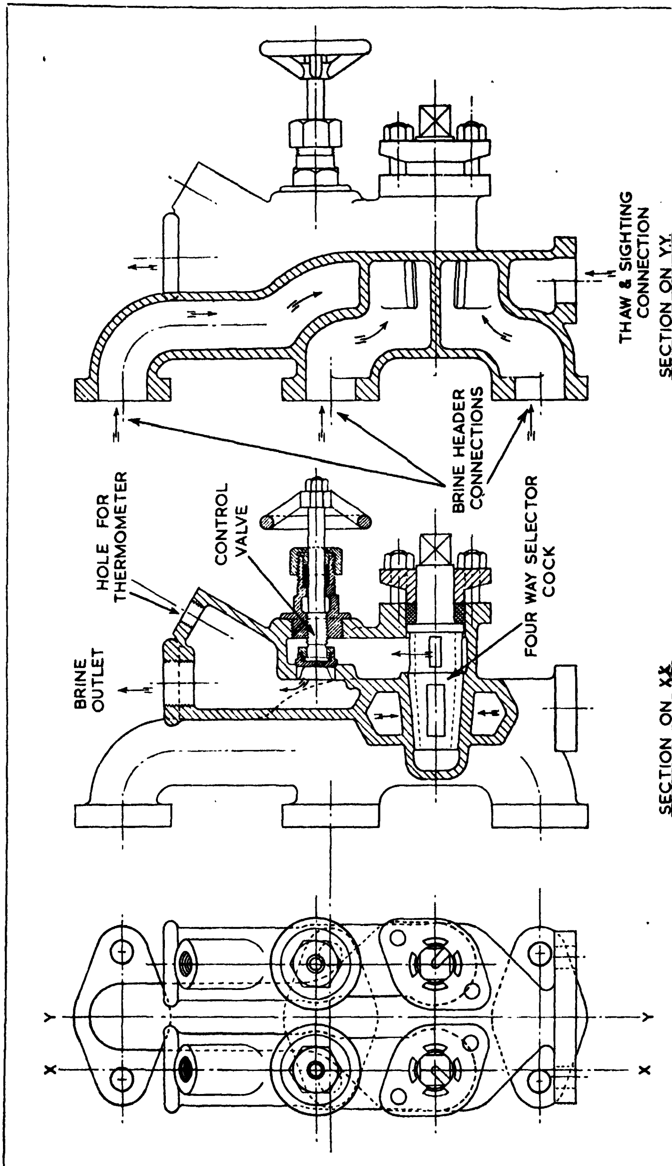


FIG. 5.—Duplex Four-way Selector Cock.

The present practice, however, is to arrange the pipes in the form of an air cooler, or battery, at one end of the cold chamber. Air is cooled by passing it over the cold pipes in the air cooler by means of a fan and then distributing the cold air throughout the space as desired.

It is sometimes found desirable to use both methods.

It is now usual to use a closed brine system where the brine is drawn from the evaporator by a pump, circulated through one or more circuits in the chamber to be cooled, returned to a closed header, and from this to the evaporator. The whole system is under a head of brine for which purpose a balance tank is fitted just above the highest level of the brine in the system. Thus there is no point where air can leak into the brine system which can therefore be kept free of air.

The multi-temperature brine system: When fruit, chilled and frozen meat are all carried in the same vessel, it is necessary to provide brine at a temperature best suited to each produce carried. Thus a modern ship may require brine at zero F. for freezing, brine at 20° F. for chilling and brine at any desired higher temperature for fruit. For thawing coolers and grids still higher temperatures are required. In large installations different evaporators are used for each temperature. The arrangement is shown diagrammatically in fig. 7, p. 746.

In both cases the brine pipes are arranged in a number of circuits of limited length, so that the return temperatures shall only show a small rise, and each of these circuits is separately controlled.

In some installations, the brine returning from the piping in the chill and fruit spaces is pumped directly back to the chilling delivery header and the heat absorbed by the brine in its passage through the chilling circuits is eliminated by means of an injection of freezing brine into the chilling system, the excess of brine thus carried in the chilling system being automatically returned into the freezing system.

Fig. 5, p. 744, shows the sectional arrangement of a duplex four-way selector cock and fig. 6, below, shows a section through the brine feed and return headers with the selector cocks mounted

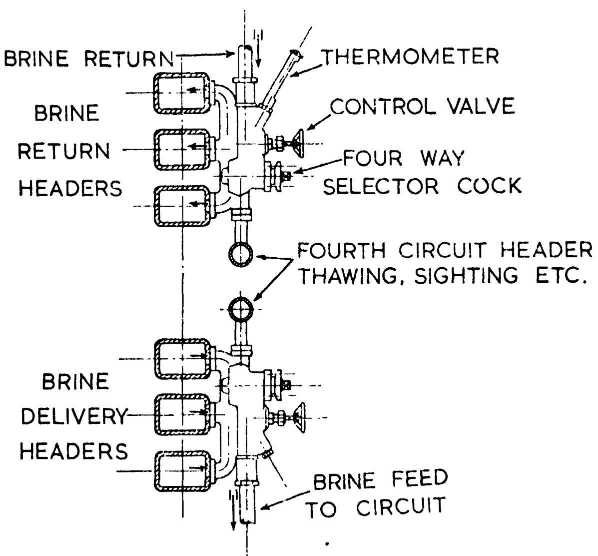


FIG. 6.—Arrangement of Brine Headers.

thereon. The two halves are quite separate as far as flow of brine is concerned, each double cock serving two circuits, and they are made in one casting for convenience. The two halves have individual control or shut-off valves and selector cocks, the headers being common to both.

Thermometers are fitted on the brine return fittings, but in order to keep the delivery and return cocks interchangeable, the bosses for the thermometers are plugged on the delivery cocks.

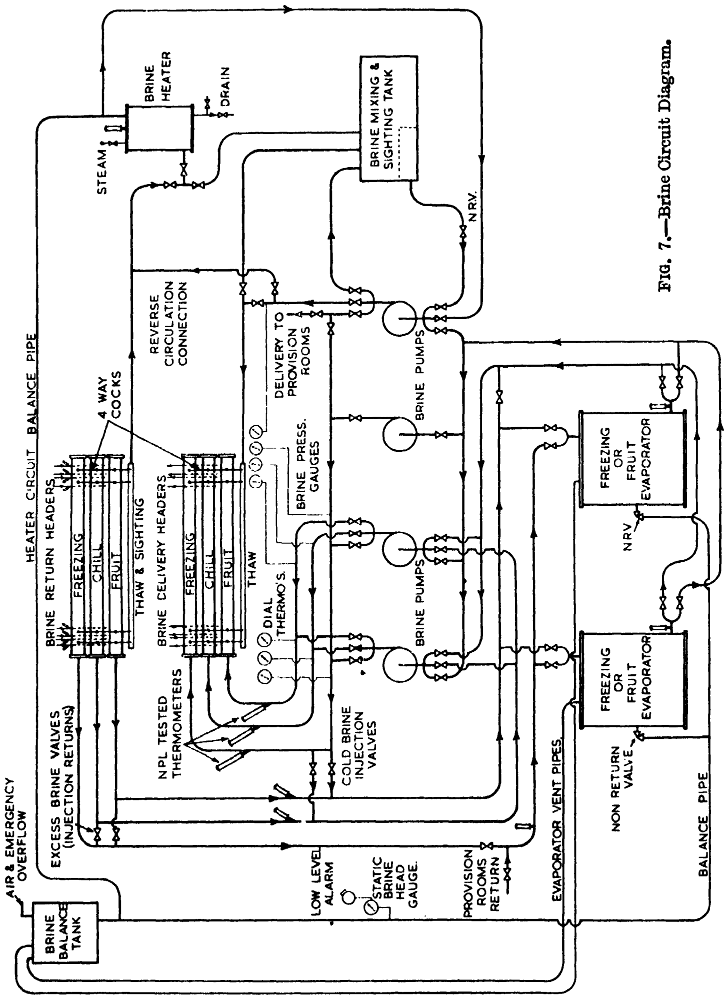


FIG. 7.—Brine Circuit Diagram.

One of the four circuits, usually the thawing circuit, is arranged for auxiliary duties, sighting, brine making, etc.

A connection is taken from the thawing return to the brine-making tank.

If there is a doubt as to whether any circuit is flowing freely or is being choked by foreign matter, the return cock of that section can be set to discharge into a tank where the flow can be observed.

It will be noticed on reference to fig. 7 that a tank is fitted to contain the surplus brine due to increase in volume caused by higher temperatures when thawing down the refrigerated chambers after unloading cargo.

A large modern installation may involve up to about 200 separate brine circuits, and the control room for a closed brine system is shown in fig. 9.

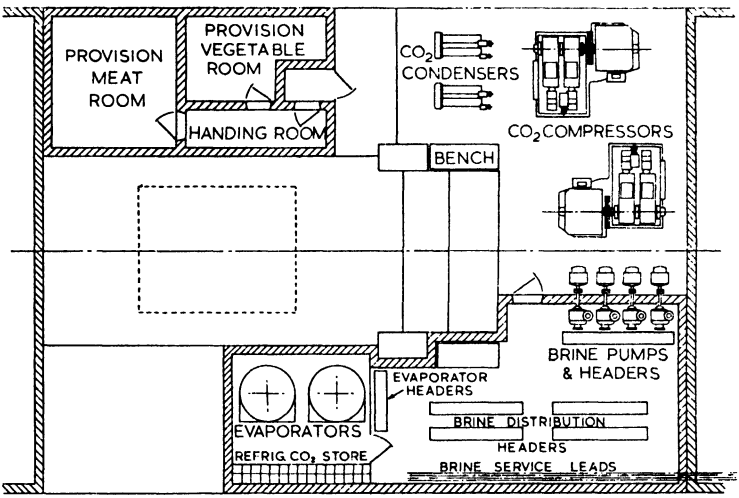


FIG. 8.—Refrigerating Plant on Board Ship.

The refrigerating machinery for these installations always consists of at least two units, each complete with its condenser and evaporator, and capable of independent working, each being of sufficient capacity to maintain the required temperatures in the cargo spaces at sea if the other is out of action. In most large plants there are three machines, two of which have ample power to maintain the cargo, with one independent unit in reserve.

The carriage of fruit, which is a living organism, demands air circulation to remove the heat given off by it due to respiration and to cool down the fruit, as, in many cases, it is delivered warm to the ship. A supply of fresh air must be introduced in order to control the CO_2 content of the air in the space. The air is cooled in a battery or cooler, and is distributed through the holds by ducts on the walls and buried in the insulation between the beams of the deck. A typical arrangement of an air-cooled hold is shown in fig. 10 which is taken, by permission, from the article by Messrs. Ormiston & Farmer on 'The Running and Maintenance of Marine Machinery,' published by the Institute of Marine Engineers. With these vertical air systems no dunnage is required except on the floor, where it serves to take the air away to the suction openings at the sides. The air ducts on the sides of the space are divided horizontally, the upper half being the delivery and the lower the suction. When the system is at work the air is drawn into the suction duct through openings near the deck, passes over the coolers, and is discharged into the delivery duct, from which it passes into the ducts already referred to in the deck insulation. These overhead ducts are perforated with 2-in. holes, and the air is discharged downwards through them, through the cargo, and finds its way into the suction duct to repeat the process. The square of the hatch is refrigerated by air blown through pipes as shown.

The system of air distribution shown in fig. 10 was introduced by J. & E. Hall, Ltd., and is now fitted to most modern ships. In this the sides of the hold are completely screened. This screening is divided vertically at frequent intervals so that the air passes from the bottom of the hold to the main suction duct and so to the coolers. The cooled air is distributed from main delivery ducts on the underside of the deck above, and particular care is taken to distribute cold air in front of the bulkhead. This system is used for the carriage of all types of refrigerated produce such as meat in frozen and chilled condition, fruit and cheese, and no dunnage is required except on the floor.

FIG. 9.

Another system of cooling with which many of the older ships for mixed cargo are fitted, is called the screened grid system. The grids on the side of the hold are shielded by timber screens which form a duct, with openings for air at intervals. The air is circulated by a fan which draws it from the duct at the end and delivers it to the ducts at the side of the hold. It is now out of date except for the smallest cargo spaces.

Heat Pumps

The use of a vapour compression refrigerating plant as a means of heating is a fairly obvious application, since its function is to remove heat from one substance and transfer it to another, and in fact the heating of houses by this means was suggested by Lord Kelvin as early as 1852.

There is no essential difference either in the principal components or in the method of operation between a refrigerating plant and a heat pump; the term '*reversed* refrigeration cycle,' often applied to the latter, is quite misleading. The only difference is that in the refrigerating plant the user's interest is in the evaporator or heat absorption side of the installation, whereas in the heat pump it is the condenser or heat rejection side that is important, and the pressure here must be adapted to the temperature at which the heating medium is required. It is, however, quite possible for the same installation to be in use simultaneously as a refrigerating plant and as a heat pump. Indeed, the most economical application of the heat pump occurs where the refrigeration

can be utilised, either simultaneously, or at a different time, as when a building is heated in winter and air-conditioned in summer.

It is natural that the idea of the heat pump should be attractive, particularly when fuel is as scarce as it is at the present time, because by its aid several units of energy in the form of heat can be obtained for the expenditure of one unit of electrical energy to drive the compressor.

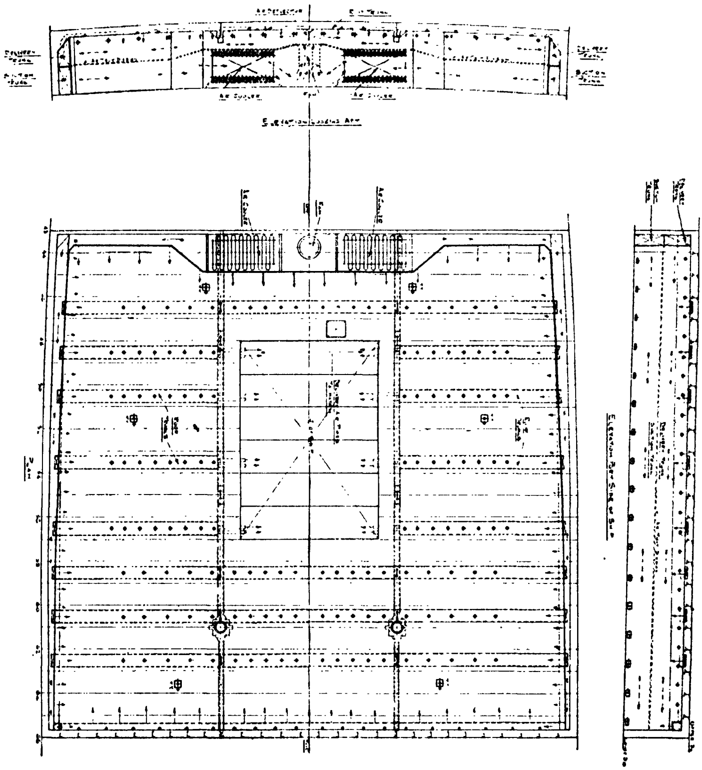


FIG. 10.—Arrangement of Tween Deck with Twin Battery and Vertical Air Circulation.

There are, however, a number of facts which tend to discourage the adoption of the heat pump, *i.e.* (1) The high first cost of the plant ; (2) the efficiency of production of electrical energy is low, so that one unit of energy in the form of electricity requires very much more coal than the same amount of energy as heat produced by the direct combustion of coal.

(3) The advantage ratio (heat produced ÷ energy consumed) of a heat pump depends on operating conditions : in order that it should be high, the condensation temperature must be low, which involves the use of large heating surfaces, and the evaporation temperature must be high, whereas in fact (4) the source of heat is usually coldest just when most heat is required. The two most readily available sources are rivers or lakes, and the atmosphere.

Heat pumps have been used mostly in the U.S.A., and also in Switzerland, where coal is very scarce, water power for the production of electricity is abundant, and lakes form a readily available source of heat. Even in such cases, however, it has usually been found necessary to supplement the heat pump by a conventional heating system during the coldest weather.

Table XXV gives an approximate idea of the advantage ratio obtainable with either ammonia or Freon-12 under various working conditions. The figures are based on actual results, and give the ratio of heat produced to actual energy input to the electric motor.

TABLE XXV.—ADVANTAGE RATIO OF HEAT PUMP.

Evaporator Gauge. ° F.	Condenser Gauge and Liquid. ° F.			
	110	120	130	140
30	3·5	3·1	2·75	2·45
40	4·15	3·55	3·1	2·65
50	5·0	4·25	3·65	3·1
60	6·15	5·1	4·3	3·6
70	7·8	6·25	5·1	4·2

See also Descriptive Section XXXV,
Heenan & Froude Ltd.

SECTION XXXVI

AIR COMPRESSION (pp. 753–766)

(Contributed by R. L. Quertier, B.Sc., M.I.Mech.E., A.M.I.C.E., A.C.G.I.)

SECTION XXXVI

AIR COMPRESSION

(Contributed by R. L. Quertier, B.Sc., M.I.Mech.E., A.M.I.C.E., A.C.G.I.)

TABLE SHOWING THE WORKING OF SIX SELECTED WATER PUMPING WINDMILLS DURING TRIALS OF 150 HOURS, BY THE ROYAL AGRICULTURAL SOCIETY.

Name of Engine.	Diam. of Wheel in Feet.	Total No. of Gallons Pumped.	Total Pressure Head in Feet.	Total Revolutions of Wheel.	Total No. of Double Strokes of Pump.	Transmission.	Ratio of Wheel to Pump.	Particulars of Pumps. Diam. in Ins.	Particulars of Pumps. Stroke in Ins.	Single or Double Acting.	Average Water H.P. per Minute.
Canadian-Imperial	16	79,000	200	308,000	42,000	rack gear	5	4	22	double	·53
Henry Sykes, Ltd.	16	49,000	200	210,000	210,000	direct crank	1	2·5	8	„	·33
J. W. Titt . .	16	46,000	200	285,000	285,000	„ „	1	3·25	6·12	„	·30
Thomas & Son .	16	41,000	200	307,000	122,000	geared „	2·5	4	8	single	·275
R. Warner & Co. .	16	40,000	200	330,000	160,000	„ „	2	3·5	5	double	·268
J. W. Titt . .	16	36,000	200	230,000	88,000	„ „	2·5	4·5	8	„	·242

TABLE SHOWING PERFORMANCES OF WATER-PUMPING WINDMILLS, SHOWING QUANTITY IN IMPERIAL GALLONS PUMPED AGAINST A TOTAL HEAD OF 200 FEET, AS REGISTERED BY METER, PER DAY OF 10 HOURS.

Average Wind Velocity in miles per hour for each day.	Results of Two Windmills with 8 ft. diameter wheels.		Results of Six Windmills with 12 ft. diameter wheels.		Results of Seven Windmills with 16 ft. diameter wheels.		Results of Three Windmills with 20 ft. diameter wheels.		Result of Windmill with 30 ft. diameter wheel.	
	Highest.	**Lowest.**	**Highest.**	**Lowest.**	**Highest.**	**Lowest.**	**Highest.**	**Lowest.**	**Highest.**	**Lowest.**
5	—	—	650	320	1,200	350	—	—	940	—
8	—	—	1,500	900	3,000	600	1,600	800	4,000	—
10	—	—	2,000	950	4,000	1,600	3,500	1,200	7,000	—
12	—	—	2,400	1,000	4,200	2,000	4,100	3,800	8,000	—
15	400	—	4,000	1,250	5,600	3,750	6,000	4,000	13,000	—
18	700	—	3,500	1,700	5,200	4,500	—	—	19,000	—
20	900	700	4,200	1,400	6,930	4,200	—	—	21,000	—
22	800	—	4,500	1,500	7,600	3,300	—	—	23,000	—
24	—	—	4,950	1,600	10,400	3,600	—	—	24,000	—
	CORRECTED AVERAGE CAPACITY IN GALLONS PER HOUR RAISED 50 FEET.									
15	250		1,000		1,500		2,200		5,000	

AIR COMPRESSION

P = absolute pressure.
p = gauge ,,
v = volume of gas at pressure P.
B = barometric pressure, in the same unit as P and p.
T = absolute temperature.

Let suffix 1 be initial condition.
,, ,, 2 be final condition.

$P = p + B = p + 14 \cdot 7$ lbs./sq. in. for normal atmospheric air.

Isothermal compression and expansion.—In an isothermal change of conditions the temperature remains constant, *i.e.* T = constant.

hence PV = constant = RT (Gas Law)
or $P_1 v_1 = P_2 v_2$.

$$\text{and } \frac{P_1}{P_2} = \frac{v_2}{V_1}.$$

The horsepower required to compress and deliver gas isothermally (or the horsepower obtained by expansion and delivery of the gas) is

$$\text{H.P.}_{is} = \frac{144 \times P_1 \times \text{F.A.D.}}{33{,}000} \log_e P_2/P_1.$$

Where F.A.D. is the volume flow of air at initial conditions of P_1 and T_1 in cub. ft. per min. with P in lbs. per sq. in. abs.

This equation can be used for determining the air horsepower for water-cooled compressors of two or more stages with intercoolers between stages.

Adiabatic compression and expansion.—An adiabatic change of condition is one in which no heat is added or subtracted from the gas, and during which no losses (due to friction and eddies) occur. It conforms to the equation Pv^{γ} = constant.

When = ratio of specific heats at constant pressure and volume.
= $1 \cdot 4$ for air at normal temperatures.

Pv^{γ} = constant.

$$\text{hence } \frac{P_1}{P_2} = \left(\frac{v_2}{v_1}\right)^{\gamma} = \left(\frac{T_1}{T_2}\right)^{\frac{\gamma}{\gamma - 1}}.$$

The horsepower required to compress and deliver a gas adiabatically (or the horsepower obtained by expansion and delivery) is

$$\text{H.P.}_{ad} = \frac{144 \times P_1 \times \text{F.A.D.}}{33{,}000} \cdot \frac{\gamma}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma - 1}{\gamma}} - 1\right]$$

if the compression is done in several stages adiabatically with intercoolers between stages the equation becomes

$$\text{H.P.}_{ad} = \frac{144 \times P_1 \times \text{F.A.D.} \times N}{33{,}000} \cdot \frac{\gamma}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma - 1}{N\gamma}} - 1\right]$$

where N is the number of stages.

In practice, however, a certain amount of heat is given out during compression, the amount depending chiefly on the cooling arrangements and the speed of rotation. In this case the index of compression is altered, hence Pv^{n} = constant
where n is less than γ

For slow speed water jacketed machines n will be approximately
$1 \cdot 2 - 1 \cdot 25$ for air

For high speed air cooled machines n will be approximately
$1 \cdot 30 - 1 \cdot 35$ for air

These values for n should be used in place of γ for practical calculations.

Other forms of horsepower equation can be deduced from those given ; the most common to the aeronautical engineer is

$$\text{H.P.} = \frac{Kp\,J.\,W.\,\Delta T}{33{,}000}$$

when Kp is the specific heat at constant pressure.
J is Joule's equivalent.
W is gas flow in lbs. per min.
ΔT is the temperature rise, *i.e.* $T_2 - T_1$

for air $Kp = 0 \cdot 24$. J = 1,400 when ΔT is in degrees centigrade.

$$\therefore \text{H.P. approx.} = \frac{W.\,\Delta T}{100}.$$

Increase of Volume of Air by Increase of Temperature.

If a volume, V_1, of air at atmospheric pressure and at $t_1°$ F. be raised to $t_2°$ F., the air being allowed to expand freely, then the volume V_2 at $t_2°$ F. will be

$$V_2 = V_1 \frac{461 + t_2°}{461 + t_1°}.$$

Example.—If volume V_1 of air at 32° ($t_1°$) be 1,000, then volume V_2 at 525° ($t_2°$) will be :

$$V_2 = 1{,}000 \frac{461 + 525}{461 + 32} = 1{,}000 \times \frac{986}{493} = 2{,}000.$$

The following table gives the increase in volume from 32° F. :—

Increase of Volume of Air by Increase of Temperature from 32° F.

F.°	V.	F.°	V.	F.°	V.	F.°	V.	F.°	V.
32	1,000	49	1,034	66	1,069	83	1,103	100	1,138
33	1,002	50	1,037	67	1,071	84	1,105	110	1,158
34	1,004	51	1,039	68	1,073	85	1,107	120	1,178
35	1,006	52	1,041	69	1,075	86	1,110	130	1,199
36	1,008	53	1,043	70	1,077	87	1,112	140	1,219
37	1,010	54	1,045	71	1,079	88	1,114	150	1,239
38	1,012	55	1,047	72	1,081	89	1,116	160	1,260
39	1,014	56	1,049	73	1,083	90	1,118	170	1,280
40	1,016	57	1,051	74	1,085	91	1,120	180	1,300
41	1,018	58	1,053	75	1,087	92	1,122	190	1,320
42	1,020	59	1,055	76	1,089	93	1,124	200	1,341
43	1,022	60	1,057	77	1,091	94	1,126	210	1,361
44	1,024	61	1,059	78	1,093	95	1,128	212	1,365
45	1,026	62	1,061	79	1,095	96	1,130	300	1,544
46	1,028	63	1,063	80	1,097	97	1,132	400	1,746
47	1,030	64	1,065	81	1,099	98	1,134	500	1,949
48	1,032	65	1,067	82	1,101	99	1,136	525	2,000

Table of Brake Horse Powers Required at the Compressor Shaft.

B.H.P.

Free Air Delivered in Cubic Feet Min	25	50	60	70	80	90	100
400	46	65	72	78	84	88	92
300	35	48	53	58	63	67	70
250	30	41	44	48	52	56	59
200	24	33	36	39	42	44	46
150	19	25	28	30	32	34	35
125	16	23	24	25	27	29	30
100	12	18	20	22	23	24	25
75	11	14	16	17½	19	20	21
60	8	11	12½	14	15	15½	16
50	7	10	11	12	13	13½	14
40	6	8	9	9½	10	10½	11
30	4	6	6	6½	7	7½	8
20	3	4½	4½	5	5	5½	5½
10	1½	2½	2½	2¾	2¾	3	3

B. H. P.

Delivery Pressure Lbs./Sq. ins. (Gauge).

The table is only a guide. The B.H.P. will vary a little according to design and piston speed.

Flow of Air through an Orifice in Cubic Feet of Free Air per Minute Flowing from a Round Hole in Receiver into the Atmosphere.

Dia. of Orifice in Ins.	Gauge Pressure (lbs. sq. in.).																
	2 lbs.	5	10	15	20	25	30	35	40	45	50	60	70	80	90	100	125
$\frac{1}{64}$	0·038	0·0597	0·0842	0·103	0·119	0·133	0·156	0·173	0·19	0·208	0·225	0·26	0·295	0·33	0·364	0·40	0·486
$\frac{1}{32}$	0·153	0·242	0·343	0·418	0·485	0·54	0·632	0·71	0·77	0·843	0·914	1·05	1·19	1·33	1·47	1·61	1·97
$\frac{1}{16}$	0·647	0·965	1·36	1·67	1·93	2·16	2·52	2·80	3·07	3·36	3·64	4·2	4·76	5·32	5·87	6·45	7·85
$\frac{1}{8}$	2·435	3·86	5·45	6·65	7·7	8·6	10	11·2	12·27	13·4	14·5	16·8	19	21·2	23·5	25·8	31·4
$\frac{1}{4}$	9·74	15·4	21·8	26·7	30·8	34·5	40	44·7	49·09	53·8	58·2	67	76	85	94	103	125
$\frac{3}{8}$	21·95	34·6	49	60	69	77	90	100	110·45	121	130	151	171	191	211	231	282
$\frac{1}{2}$	39	61·6	87	107	123	138	161	179	196·35	215	232	268	304	340	376	412	502
$\frac{5}{8}$	61	96·5	136	167	193	216	252	280	306·8	336	364	420	476	532	587	645	785
$\frac{3}{4}$	87·6	133	196	240	277	310	362	400	441·79	482	522	604	685	765	843	925	—
$\frac{7}{8}$	119·5	189	267	326	378	422	493	550	601·3	658	710	822	930	1004	—	—	—
1	156	247	350	427	494	550	645	715	785·4	860	930	—	—	—	—	—	—
1¼	242	384	543	665	770	860	1000	—	—	—	—	—	—	—	—	—	—
1½	350	550	780	960	—	—	—	—	—	—	—	—	—	—	—	—	—
2	625	985	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—

(*Ingersoll-Rand Co., Ltd.*)

Delivery Gauge Press. lbs./sq. in.	Isothermal. Volume Ratio.	Isothermal. M.E.P. lbs./sq. in. g.	Isothermal. H.P. per cu. ft./min.	Adiabatic. Volume Ratio.	Adiabatic. M.E.P. lbs./sq. in. g.	Adiabatic. H.P. per cu. ft./min.	Adiabatic. Temp. Ratio.	Adiabatic. Temp. from 60°.
1	0·936	0·97	0·00423	0·954	0·973	0·00424	1·02	70
2	0·880	1·87	0·00816	0·913	1·91	0·00833	1·038	80
3	0·831	2·68	0·0127	0·876	2·85	0·0124	1·055	89
4	0·786	3·51	0·0153	0·843	3·68	0·0160	1·073	98
5	0·746	4·30	0·0187	0·812	4·55	0·0198	1·088	106
6	0·710	5·06	0·0221	0·784	5·35	0·0233	1·104	114
7	0·677	5·76	0·0251	0·758	6·12	0·0267	1·119	121
8	0·648	6·35	0·0277	0·735	6·85	0·0299	1·134	130
9	0·620	7·00	0·0305	0·713	7·60	0·0331	1·148	137
10	0·595	7·63	0·0333	0·692	8·27	0·0361	1·162	145
11	0·572	8·23	0·0359	0·671	8·97	0·0391	1·176	151
12	0·551	8·80	0·0384	0·655	9·66	0·0421	1·189	158
13	0·531	9·28	0·0405	0·638	10·29	0·0448	1·201	165
14	0·512	9·82	0·0423	0·622	10·91	0·0476	1·214	171
15	0·495	10·34	0·0451	0·607	11·57	0·0505	1·226	177
20	0·424	12·62	0·0551	0·543	14·46	0·0631	1·282	207
25	0·370	14·60	0·0637	0·494	17·06	0·0744	1·334	234
30	0·329	16·35	0·0714	0·454	19·43	0·0848	1·380	258
35	0·296	17·90	0·0781	0·421	21·61	0·0943	1·423	280
40	0·269	19·31	0·0843	0·393	23·71	0·103	1·463	301
45	0·246	20·60	0·090	0·370	25·61	0·112	1·501	320
50	0·227	21·78	0·095	0·349	27·42	0·119	1·537	339
55	0·211	22·87	0·100	0·331	29·16	0·127	1·570	356
60	0·197	23·89	0·1040	0·315	30·76	0·134	1·602	373
65	0·184	24·84	0·1082	0·301	32·32	0·141	1·632	389
70	0·174	25·74	0·1120	0·288	33·81	0·147	1·662	403
75	0·164	26·58	0·1157	0·277	35·20	0·153	1·689	418
80	0·155	27·34	0·1190	0·266	36·59	0·160	1·716	432
85	0·147	28·14	0·1225	0·257	37·90	0·165	1·741	445
90	0·140	28·86	0·1257	0·248	39·20	0·171	1·766	458
95	0·134	29·54	0·1285	0·240	40·41	0·176	1·789	470
100	0·128	30·20	0·1314	0·233	41·60	0·181	1·813	483
120	0·109	32·60	0·142	0·205	45·2	0·198	1·88	457
150	0·089	35·59	0·155	0·177	51·0	0·223	1·99	516
200	0·068	39·50	0·172					
300	0·047	45·30	0·197					
400	0·035	49·20	0·214					
500	0·029	52·70	0·229					
750	0·019	58·10	0·253					
1000	0·0115	62·20	0·271					
2000	0·0073	72·30	0·315					
3000	0·0049	78·25	0·341					
5000	0·0029	85·70	0·374					

Notes on Graphs and Table showing Loss in Air Mains due to Friction.

(Graphs, see pages 760 and 761; Table, see page 758.)

The use of the graphs and table enables the loss in an air main to be quickly ascertained.

They are based on a comprehensive series of tests made in South Africa.

The results obtained must be taken as approximate, as it is impossible in such a simple way to allow for differences of pipe surface, elbows, tees, valves, etc.

Case 1.

***To find discharge in cubic feet per minute*, given that the initial pressure is 100 lbs. per sq. in. and the drop in pressure through 8,000 ft. of 4-in. main is 25 lbs. per sq. in.**

The drop in pressure for 100 ft. will be 25/80, that is, 0·3125 lb. per sq. in.

From the graph it will be seen that the intersection of the lines representing a drop 0·3125 lb. per sq. in. and 12·56 sq. ins.(the area of a 4-in. pipe) gives 970 cub. ft. of free air approximately.

The mean pressure in the pipe will be (100 plus 75) divided by 2, equals 87·5 lbs. per sq. in., or, say, 102·5 lbs. per sq. in. absolute.

From the table, under the heading ' to find free air per minute,' the multiplier corresponding to 102·5 lbs. per sq. in. absolute is 1·13 approximately.

Thus the discharge will be 970 × 1·13, or 1,100 *cub. ft. of free air per minute.*

Case 2.

To find the drop in pressure and the final pressure when a main is 8 ins. diameter and 600 ft, in length and passes 4,000 cub. ft. of free air with an initial pressure of 100 lbs. per sq. in.

From the graph the intersection of the line representing 4,000 cub. ft. of free air per minute and 28·3 sq. in. area, gives a pressure drop of 0·6 lb. per sq. in. approximately for 100 ft. run, or 3·6 lbs. per sq. in. for 600 ft., the length of the main.

A first approximation of the mean pressure is (100 plus 96·4) divided by 2, equals 98·2 lbs. per sq. in. or 113·2 lbs. per sq. in. absolute.

From the table, under the heading ' to find the pressure drop,' the multiplier corresponding to 113·2 may be interpolated as 0·71 approximately.

The corrected pressure drop will be 2·56 lbs. per sq. in., and therefore *the terminal pressure will be 97·4 lbs. per sq. in.*

LOSS IN AIR MAINS.

Density lbs. per cub. ft.	Corresponding Air Pressure lbs. sq. in. abs.	MULTIPLIERS. *Case* 1. To find Discharge.	*Case* 2 *and* 4. To find Drop in Pressure or Length.	*Case* 3. To find Area of Pipe Section.
0·01	1·9	0·150	42·1	4·46
0·02	3·9	0·222	20·5	3·34
0·03	5·8	0·270	13·8	2·86
0·04	7·7	0·310	10·4	2·55
0·05	9·6	0·347	8·3	2·33
0·06	11·6	0·380	6·9	2·16
0·08	15·4	0·438	5·2	1·94
0·10	19·3	0·492	4·2	1·78
0·15	29·0	0·604	2·8	1·50
0·20	39·0	0·700	2·05	1·33
0·25	48·0	0·776	1·66	1·22
0·30	58·0	0·853	1·38	1·14
0·40	77·0	0·982	1·04	1·02
0·418	80·0	1·000	1·30	1·00
0·50	96·0	1·095	0·830	0·93
0·60	116·0	1·210	0·690	0·86
0·70	135·0	1·295	0·590	0·81
0·80	154·0	1·390	0·520	0·77
0·90	173·0	1·475	0·465	0·74
1·00	193·0	1·555	0·420	0·71
1·50	288·0	1·900	0·276	0·60
2·00	385·0	2·190	0·205	0·53
2·50	480·0	2·450	0·167	0·49
5·00	960·0	3·470	0·083	0·37

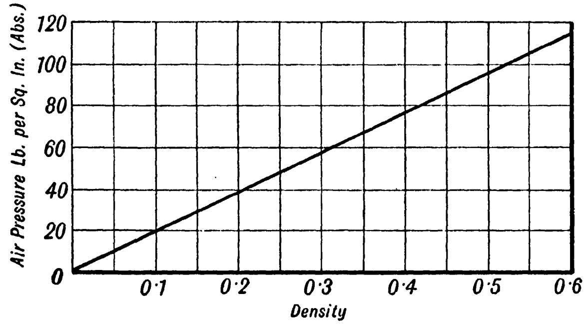


FIG. 1.

Case 3.

To find the diameter of an air main 8,000 ft. long which discharges 2,000 cub. ft. of free air per minute under an initial pressure of 200 lbs. per sq. in. and a pressure drop of 40 lbs. per sq. in. in the length of the main.

The drop in pressure per 100 ft. of main is 0·5 lb. per sq. in.

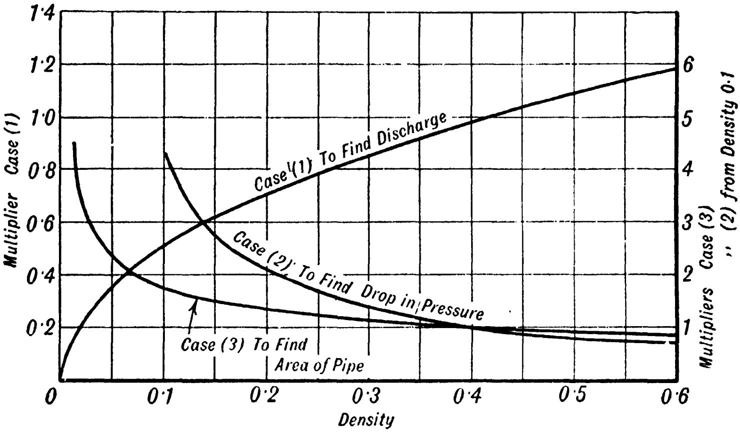


Fig. 2.

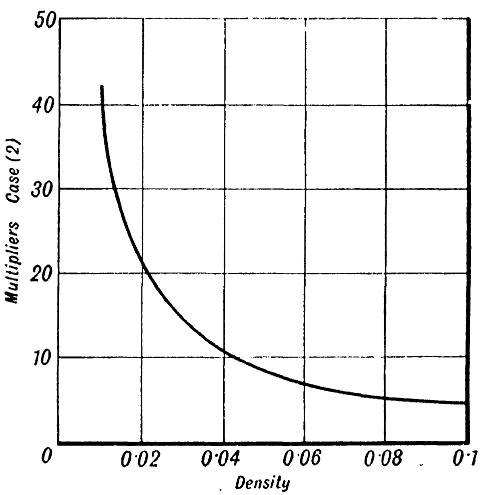
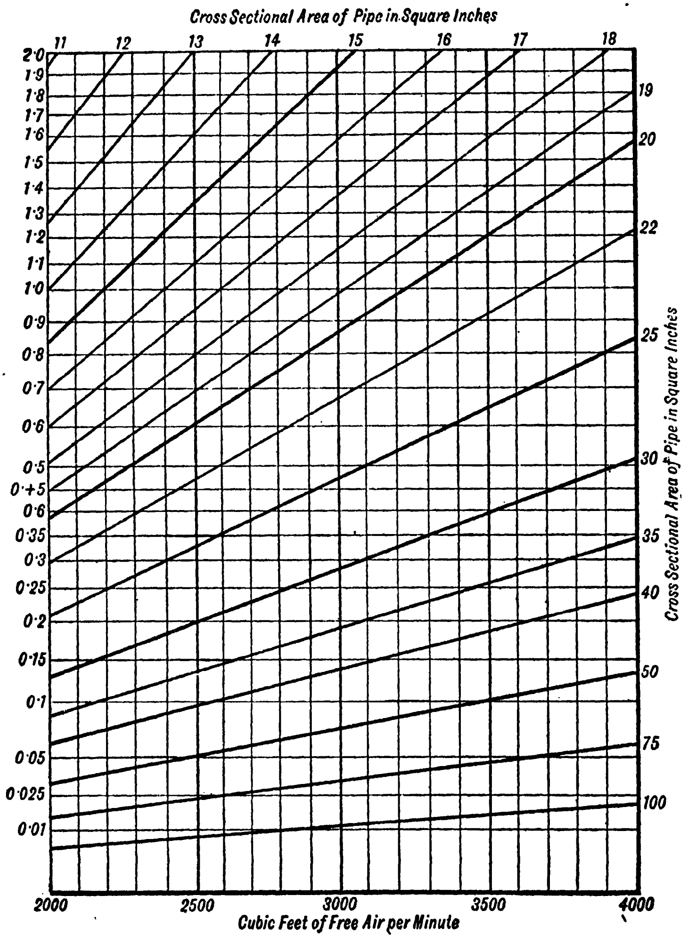


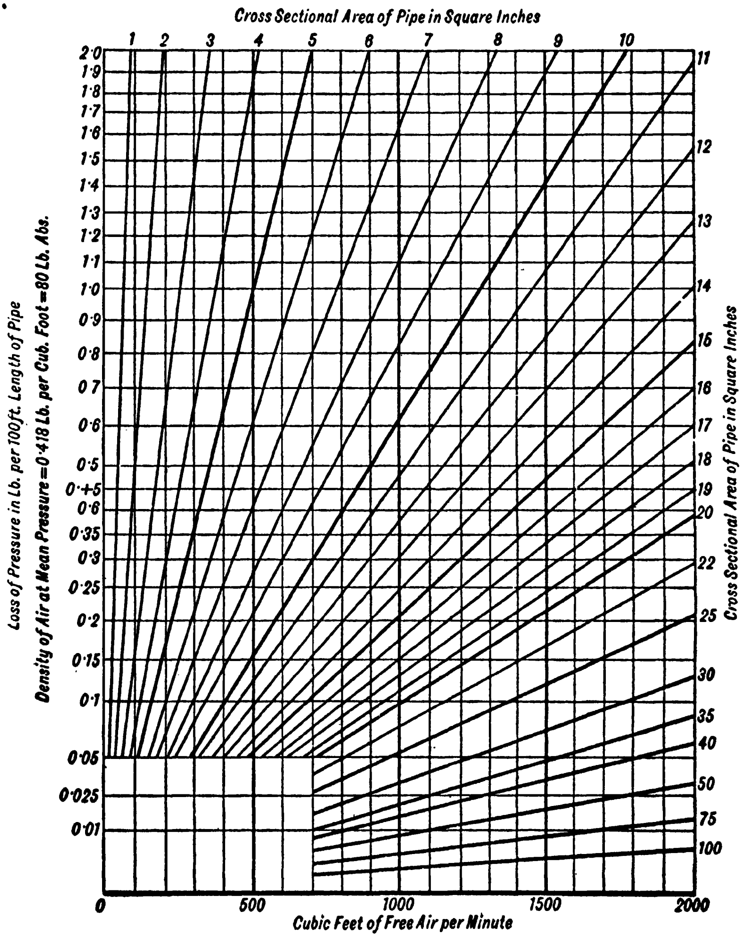
Fig. 3.

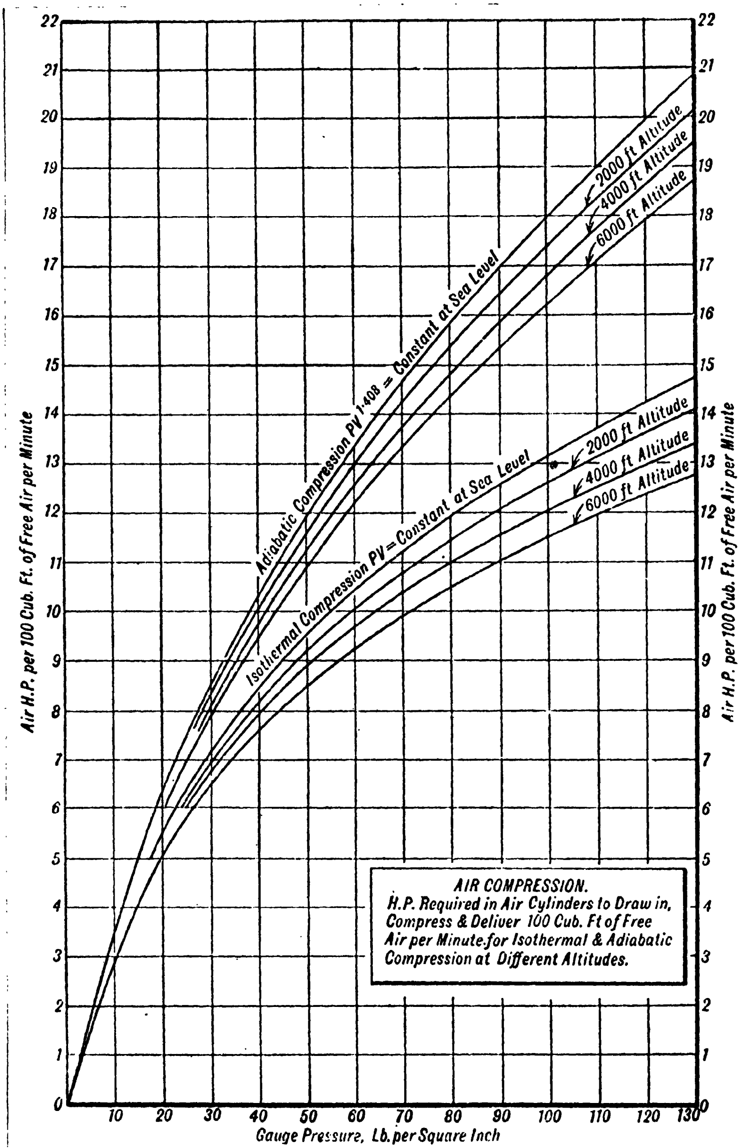
From the graph, the intersection of the lines representing 2,000 cub. ft., and a pressure drop of 0·5 lb., gives an area of 18·15 lbs. per sq. in. approximately.

From the table, under the heading 'to find area of pipe section,' the multiplier corresponding to 180 plus 15 equals 195 lbs. per sq. in. absolute is 0·70 approximately.

Hence the diameter of the main will be 0·70 × 18·15 equals 12·7 sq. ins., *so that a 4-inch main would meet the case.*


Cross Sectional Area of Pipe in Square Inches
11
12
13
14
15
16
17
18
19
20
22
25
30
35
40
50
75
100
Cross Sectional Area of Pipe in Square Inches
Loss of Pressure in Lb. per 100 ft. Length of Pipe
Density of Air at Mean Pressure = 0·418 Lb. per Cub. Foot = 80 Lb. Abs.
2·0
1·9
1·8
1·7
1·6
1·5
1·4
1·3
1·2
1·1
1·0
0·9
0·8
0·7
0·6
0·5
0·+5
0·6
0·35
0·3
0·25
0·2
0·15
0·1
0·05
0·025
0·01
2000
2500
3000
3500
4000
Cubic Feet of Free Air per Minute


Cross Sectional Area of Pipe in Square Inches
1 2 3 4 5 6 7 8 9 10
11 12 13 14 15 16 17 18 19 20 22 25 30 35 40 50 75 100
Cross Sectional Area of Pipe in Square Inches
2·0 1·9 1·8 1·7 1·6 1·5 1·4 1·3 1·2 1·1 1·0 0·9 0·8 0·7 0·6 0·5 0·45 0·4 0·35 0·3 0·25 0·2 0·15 0·1 0·05 0·025 0·01
Loss of Pressure in Lb. per 100 ft. Length of Pipe
Density of Air at Mean Pressure = 0·418 Lb. per Cub. Foot = 80 Lb. Abs.
0 500 1000 1500 2000
Cubic Feet of Free Air per Minute



(Belliss & Morcom, Ltd.)

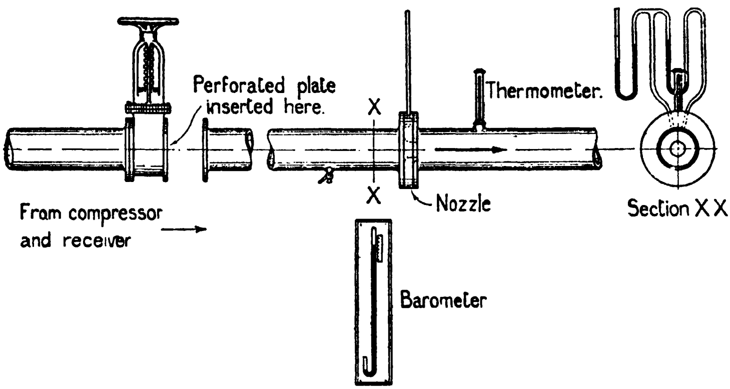
Testing of Air Compressors.

In 1929 a committee was set up by the Heat Engine Trials Standing Committee of the Institution of Civil Engineers to investigate the measurement of air flow. Resulting from the work of this committee and later from a recommendation of the British Compressed Air Society, a committee was appointed by the British Standards Institution to formulate a British Standards method of measurement of air flow and the free air delivered by compressors.

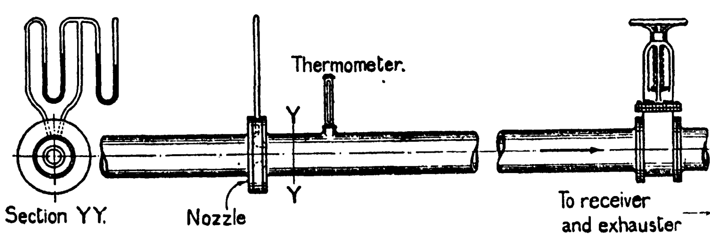
The work of this committee has resulted in the publication by the British Standards Institution of the Pamphlet No. 726–1937 which gives all the information necessary for measuring the free air delivered by an air compressor.

Definition: Free air delivered (F.A.D.). The volume of free air at the intake conditions expressed in cubic feet per minute that a compressor will take in, compress and deliver at the stated delivery pressure.

The apparatus used for testing an air compressor or alternatively an exhauster is shown in fig. 4. It consists, if testing a compressor, of a suitable pipeline leading from the compressor



Arrangement for Testing a Compressor.

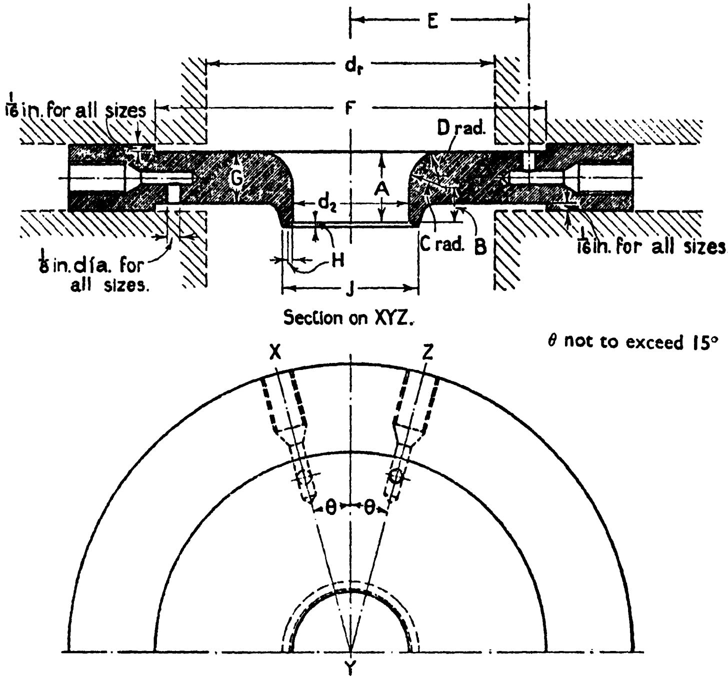


Arrangement for Testing an Exhauster.

FIG. 4.

and receiver to a straight through valve beyond which is inserted a perforated plate $\frac{1}{8}$ in. thick, having perforations $\frac{1}{8}$ in. in diameter spaced at $\frac{1}{4}$ in. centres; this plate is intended to steady the flow and is followed by the upstream pipe, the nozzle and downstream pipe, arranged as shown in fig. 4.

Fig. 5 shows the form of nozzle, size 1 in. or over. Smaller nozzles are exactly similar in the throat, but have to be modified in their general dimensions—see B.S.I. 726—1937, from which Table A is taken.



Nozzles 1-in. or Over.

FIG. 5. (See Table, p. 765.)

When testing a compressor having a steady flow, such as a turbo compressor, a receiver is unnecessary between the compressor and the measuring apparatus, but with a reciprocating machine a receiver of sufficient size must be inserted to damp out pulsations.

METHOD OF TEST.

By suitably adjusting the straight through valve immediately before the nozzle, the correct working pressure is maintained on the compressor.

The following readings are taken when making a test :—

P_i—absolute pressure at intake of compressor in inches of mercury. (Generally the atmospheric pressure prevailing at the time.)

t_i—the intake temperature in degrees Fahrenheit.

Readings taken at the measuring apparatus :—

h—pressure drop across the nozzle in inches of water. Measured by the manometer shown, fig. 4, section XX.

m_2—the absolute pressure at the downstream side of the nozzle. This is obtained by adding or subtracting from the atmospheric pressure, as given by the barometer, the water gauge reading obtained from the second leg reading of the manometer shown, fig. 4, section XX.

t—the temperature of the air in ° F. at the downstream side of the nozzle

TABLE A

DIMENSIONS OF NOZZLES.

ALL DIMENSIONS ARE IN INCHES AT THE NORMAL TEMPERATURE OF 68°F.

Nominal Size.		d₂.	A.	B.	C.	D.	Tolerance* on Profile	E.	F.	G.	H.	J.
Nozzle Dia.	Suggested pipe Dia. d_1.											
3/8	7/8	0·375 ± 0·0005	0·227 ± 0·0025	0·113	0·124	0·075	± 0·005	3/4	1 3/4	—	1/64	—
5/8	1 1/2	0·625 ± 0·0005	0·378 ± 0·0025	0·188	0·206	0·125	± 0·005	1 1/32	2 3/8	—	1/64	—
1	2 1/2	1·000 ± 0·001	0·605 ± 0·003	0·300	0·330	0·200	± 0·005	1 17/32	3 3/8	7/16	1/32	1 11/64
1 1/2	3 1/2	1·500 ± 0·001	0·908 ± 0·004	0·450	0·495	0·300	± 0·007	2 1/32	4 3/8	7/16	1/16	1 3/4
2 1/2	6	2·500 ± 0·002	1·513 ± 0·005	0·750	0·825	0·500	± 0·010	3 11/32	7	7/16	3/32	2 59/64
4	10	4·000 ± 0·002	2·420 ± 0·005	1·200	1·320	0·800	± 0·010	5 9/16	11 1/2	1/2	1/8	4 43/64
6	15	6·000 ± 0·004	3·630 ± 0·007	1·800	1·980	1·200	± 0·010	8 1/8	16 3/4	5/8	3/16	7
10	24	10·000 ± 0·007	6·050 ± 0·010	3·000	3·300	2·000	± 0·010	12 3/4	26	3/4	5/16	11 1/2
15	36	15·000 ± 0·010	9·075 ± 0·015	4·500	4·950	3·000	± 0·010	18 7/8	38 1/4	3/4	1/2	16 1/2

* The curved surface must not depart from the nominal profile by more than this tolerance at any point between the cylindrical throat and the upstream face of the nozzle.

The dimensions given in fractions of an inch are not required to a high degree of accuracy.

Intermediate sizes of nozzles when made shall conform to intermediate tolerances and shall be proportioned on the dimensions given for the 1 in. or 10 in. nozzles

From the foregoing observations the F.A.D. is found from the formula:—

$$\text{F.A.D.} = K\,(T_i/P_i)\ \sqrt{h}\ \sqrt{m_2}/T$$

in which K = the constant for the nozzle (see Table B).

$$T_i = 459{\cdot}6 + t_i.$$

where t_i = intake temperature in ° F.

P_i = absolute pressure at intake in inches of mercury.

h = pressure drop across nozzle in inches of water.

m_2 = absolute pressure on the downstream side of the nozzle in inches of mercury.

This is obtained by adding or subtracting from the atmospheric pressure, as given by the barometer, the water-gauge reading obtained from the appropriate manometer. The water-gauge readings are converted to inches of mercury by dividing by 13·6.

$$T = 459{\cdot}6 + t$$

where t = temperature at the nozzle in ° F.

The nozzle may also be inserted in a pipe line where the pressure downstream of the nozzle is above that of the atmosphere. It is important to ensure steady flow at both upstream and downstream sides of the nozzle, and that the recommendations set out in detail in B.S.I. No. 726–1937 should be carried out.

The following table shows the standard nozzle sizes from ⅜ in. up to 15 ins., covering a range of F.A.D. from 6 cub. ft. per min. to 30,000 cub. ft. per min. Intermediate and larger nozzles can be used provided they follow the general form of fig. 4.

TABLE B.

Nozzle diameter inches.	Suitable pipe-line internal diameter inches.	Approximate F.A.D.			Constant K.
		h = 0.4″	h = 4″	h = 40″	
⅜	⅞	—	6	18	0·73
⅝	1½	—	16	50	2·03
1	2½	—	42	130	5·18
1½	3½	30	90	300	11·65
2½	6	80	260	800	32·4
4	10	210	660	2 100	82·8
6	15	470	1 500	4 700	187
10	24	1 300	4 100	13 000	518
15	36	3 000	9 250	30 000	1 165

See also Descriptive Section XXXVI.

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SECTION XXXVII

SANITARY ENGINEERING—SEWERAGE—SEWAGE TREATMENT AND DISPOSAL—SANITATION OF BUILDINGS

(pp. 769–798)

(Contributed by L. B. Escritt, A.M.I.C.E., M.I.S.E., M.R.San.I., Hon.M.Inst.S.P., F.G.S.)

SECTION XXXVII

SANITARY ENGINEERING—SEWERAGE—SEWAGE TREATMENT AND DISPOSAL—SANITATION OF BUILDINGS

Contributed by L. B. Escritt, A.M.I.C.E., M.I.S.E., M.R.San.I., Hon.M.Inst.S.P., F.G.S.

(Author of *Sewerage Engineering*; *Regional Planning*; *Surface Drainage*; *Sewerage Design and Specification*; *The Municipal Engineer*; joint author of the revised edition of *The Work of the Sanitary Engineer*.)

SANITARY ENGINEERING.

The term ' Sanitary Engineering ' is used loosely to include all engineering works designed and executed in the interests of public health. More specifically, it is applied to the sanitation of buildings. This section covers that part of sanitary engineering in the wider sense, which relates to drainage, including ' Sewerage,' ' Sewage Treatment and Disposal ' and ' Sanitation of Buildings.'

Reference to Publications.

In the following text the reader is referred to British Standard Specifications, published by The British Standards Institution, 28 Victoria Street, London, S.W. 1, thus: (B.S. 1438). These specifications give particulars of the qualities and dimensions of the materials used.

Sanitary works are carried out in accordance with Statutory Rules and Orders (S.R. & O.) and By-laws made under Acts of Parliament. The main relevant publications of these kinds are mentioned. In addition, local Acts and By-laws should be studied.

For more detailed information on sanitary engineering law, theory and practice than can be included in the present work, the reader is referred to the *revised edition* of *The Work of the Sanitary Engineer* (Macdonald & Evans, 8 John Street, Bedford Row, London, W.C. 1), from which the illustrations herein are reproduced.

Sanitary Enactments.

Legislation on sanitation was amended and largely consolidated by the *Public Health Act*, 1936, which is now the most important enactment on the law of public health. London has its own particular code in the *Public Health (London) Act*, 1936. Water supply (the law of which is a concern of the sanitary engineer), has recently received modern interpretation by the *Water Acts*, 1945, and 1948, designed ' to make provision for the conservation and use of water resources and for water supplies and for purposes connected therewith.' Other statutory provisions relating to particular aspects of sanitary administration are the *Public Health (Drainage of Trade Premises) Act*, 1937, the *Land Drainage Act*, 1930, the *River Boards Act*, 1948, and the *Rivers Pollution Prevention Acts*, 1876 and 1893. There are additional local Acts operative only in particular districts.

Under the *Public Health Act*, 1936:

' " Drain " means a drain used for the drainage of *one building* or of *any buildings* or yards of appurtenant to buildings *within the same curtilage*.'

' " Sewer " does not include a drain as above defined, but otherwise includes all sewers and drains used for the drainage of buildings and yards appurtenant to buildings.'

The above legal definitions are accepted and used by sanitary engineers as technical terminology and are so used in the following text.

SEWERAGE.

Discharge of Sewers.

The capacities of sewers can be calculated when the dimensions, hydraulic gradients and character of the internal surfaces are known: various formulæ have been evolved for this purpose of which the most accurate are the empiric formulæ based on experiments.

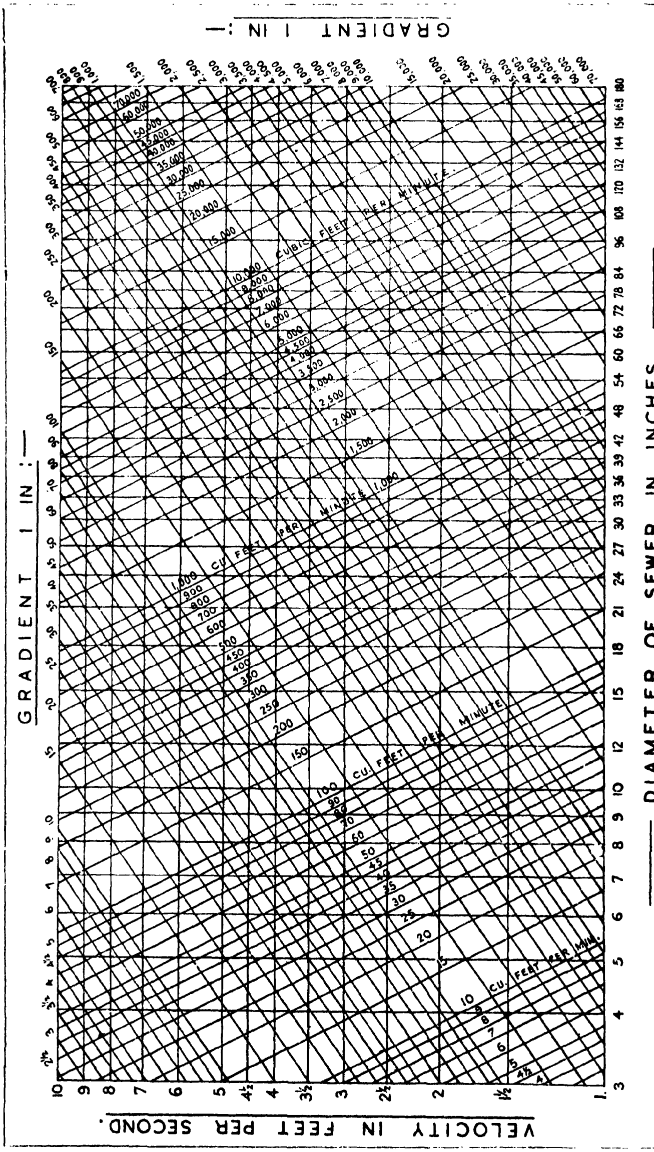


FIG. 1.—Flow in Slimy Sewers according to Barnes' Formula.—Method of using diagram: at the point of intersection of the diagonal line giving gradient and the vertical line giving diameter of pipe intersect also the diagonal line giving discharge and the horizontal line giving velocity.

For general use in determining the flows in well-used sewers of all materials, Barnes' formula 'for flow in slimy sewers' is perhaps the most accurate. It reads as follows:

$$V = 107\, m^{0\cdot7}\, i^{0\cdot5}$$

where:

V = velocity of flow in feet per second.

m = hydraulic mean depth in feet (*i.e.* the cross sectional area of flow divided by the 'wetted perimeter' of contact between the water and its bed.

i = the vertical fall divided by the length (measured along the length of the pipe, not horizontally from end to end of pipe).

Crimp and Bruges' formula is a satisfactory approximation which has been very largely used in this country. It reads as follows:

$$V = 124\, m^{0\cdot667}\, i^{0\cdot5}$$

the notation being as before. The form of Crimp and Bruges' formula that is of general use for determining the discharge of *circular pipes* is:

$$Q = \frac{3\cdot072 D^{\frac{8}{3}}}{\sqrt{I}}$$

where:

Q = discharge in cubic feet per minute.

D = diameter of pipe in inches.

I = length divided by fall.

For finding as accurately as possible the discharge of new cast-iron rising mains, Barnes' formula for flow in new asphalted cast iron pipes should be used:

$$V = 174\cdot1\, m^{0\cdot769}\, i^{0\cdot529}$$

This formula also gives a reasonable approximation for flow in steel pipes.

General Principles of Sewerage.

Apart from rising mains and inverted siphons, all sewers are designed as if they were open watercourses, the hydraulic gradient being at or below the crown line of the sewer. Thus the pipes are not considered to be surcharged and the hydraulic gradient is virtually the gradient of the crown of the sewer when the sewer is flowing full, or the gradient of the invert when there is very little flow. On this basis, sewers of different diameters connecting together are generally arranged with their crowns, not inverts, level.

The fall of the invert should be continuously in the direction of flow throughout the length of the sewer and the gradient should be sufficient to ensure a velocity of at least $2\frac{1}{2}$ ft. per second when the sewer is flowing full, in order that solids such as detritus and fæcal matter shall be swept along with the flow. Table I gives the minimum gradients for sewers flowing nearly half full at least once a day.

TABLE I.—MINIMUM GRADIENTS FOR BEST WORKING CONDITIONS IN CIRCULAR SEWERS. VELOCITY OF FLOW $2\frac{1}{2}$ FEET PER SECOND.

Diameter of Sewer (Ins.)	Gradient 1 in:	Discharge (Cubic Ft. per Minute.)
6	150	29·8
7	190	40
9	265	66·1
12	385	118
15	520	184
18	660	266
21	820	360
24	970	473
27	1100	608
30	1300	740
33	1500	889
36	1650	1069
39	1850	1249
42	2050	1446
45	2250	1659

When sewers receive very little flow, the velocity of flow is less than when they are flowing full, and therefore their gradients should be increased accordingly to bring their velocities up to 2½ ft. per second. Table II gives the proportional velocities, discharges and depths in sewers flowing partly full.

TABLE II.—DISCHARGES AND VELOCITIES IN SEWERS FLOWING PARTLY FULL.

Proportional Depth.	Proportional Velocity.	Proportional Discharge.
0·05	0·257	0·005
0·10	0·401	0·021
0·15	0·517	0·049
0·20	0·615	0·088
0·25	0·701	0·137
0·30	0·776	0·196
0·35	0·843	0·263
0·40	0·902	0·337
0·45	0·951	0·417
0·50	1·000	0·500
0·55	1·031	0·586
0·60	1·072	0·672
0·65	1·099	0·756
0·70	1·120	0·837
0·75	1·134	0·912
0·80	1·140	0·977
0·85	1·137	1·030
0·90	1·124	1·066
0·95	1·095	1·075
1·00	1·000	1·000

Velocities in sewers (other than in sewers of small diameter not flowing more than fractionally full) should be kept within bounds so as to prevent scour. Generally, the velocity of flow should not exceed 10 ft. per second and preferably 6 ft. per second, unless the sewers are constructed of cast-iron or other very durable material. Table III gives the gradients at which velocity reaches 10 ft. per second.

TABLE III.—ABSOLUTE MAXIMUM GRADIENTS FOR CIRCULAR SEWERS. VELOCITY OF FLOW 10 FT. PER SECOND.

Diameter of Sewer (Ins.)	Gradient 1 in.	Discharge (Cub. Ft. per Minute.)
6	10	115
7	12	159
9	17	261
12	25	464
15	33	732
18	42	1055
21	51	1444
24	61	1885
27	73	2359
30	83	2930
33	94	3550
36	105	4237
39	120	4906
42	130	5743
45	145	6537

Sewerage Systems.

There are two principal systems of sewerage, known respectively as the 'combined' and the 'separate' systems. In the former, one set of sewers is provided for the removal of both the soil

sewage and also the rainwater; in the separate systems two sewers are provided in every street, one for soil sewage and the other for rainwater. Both systems have their advantages, but these are counterbalanced by certain disadvantages.

The combined system has the advantage of simplicity, but it involves large and deep sewers with small volumes of sewage flowing in them in dry weather; and where the sewage has to be pumped and purified this system produces a larger volume of sewage than the separate system.

On the other hand, a separate system of sewers becomes very complicated, especially in large towns, and there is a great risk of soil drains being connected to the surface water sewers. The latter discharge direct into the nearest river or stream, and can therefore be made of smaller size than in the combined sewerage system, which concentrates all liquid upon one point. The drainage of all streets and yards is taken direct into the surface water sewers.

There is also the 'partially separate' system, in which the greater part of the rainwater is passed to surface water sewers but the run-off from back roofs and yards is permitted to be discharged to the soil sewers.

Soil Sewerage.

The quantity of sewage discharged from a built-up area is approximately the water supply to that area plus infiltration of subsoil water to leaky sewers and drains. For the design of new soil sewerage systems it suffices to ascertain or estimate the water demand of the area, in gallons per head of population per day, and make the sewers capable of taking at least four, but nor more than six, times the flows so estimated. Generally, the water demands can be obtained from the water authority. Otherwise, they may be taken as being as follows:

Small villages	10–15 gallons per head per day.
Medium to small provincial towns	20–30 ,, ,, ,, ,, ,,
Large towns	30 gallons per head per day and upwards.

The water demand plus the dry-weather infiltration is known as the 'dry-weather flow.'

The water demand and the flow of sewage vary throughout the day, being about twice the average rate of flow round about midday, and being approximately nil in the early hours of the morning. There are also daily variations: on Mondays and Tuesdays, the flow is above the average, for these are washing days; on Sundays the flow is low. The figure of four times dry-weather flow makes allowance for all these variations, plus a moderate amount of infiltration water. A large factor of safety, up to six times dry-weather flow, is desirable when infiltration is known to be high or where future extensions of the system cannot be accurately estimated.

The method of estimating the flow in any particular sewer is to count the number of houses, multiply this by the number of persons per house and by the number of gallons per head per day dry-weather flow. The total dry-weather flow so obtained is multiplied by four or six, as the case may be.

The number of persons per house may be obtained from the local authority, or otherwise estimated on averages. Prior to 1938, the number of persons per house had been falling from 5 to about 4, or 3·7 in the outer London suburbs, and 3 in some housing estates. At present the figure is higher, owing to the shortage of houses.

Surface-water Sewerage.

The flows in separate surface water sewers are considered to consist purely of rainwater; the flows in combined sewers include soil sewage, but during storm time the flow of soil sewage is so small compared with the rainfall run-off that it can be neglected. In the above respect, partially separate systems are dealt with on their own merits.

The Lloyd-Davies method of designing surface water sewers is now generally accepted. In the recognised application of this method, the sewer in question is designed to take the run-off from the drainage area resulting from a storm which is considered to produce the greatest momentary run-off that can occur once a year. Heavier storms are considered to be accommodated by surcharge of sewers, and exceptionally heavy storms of rare occurrence are permitted to cause flooding, on the grounds that the damage done does not justify the construction of extremely large sewers capable of taking any flow likely to occur.

Storms of short duration are more intense than storms of longer duration. The more intense a storm is, the less frequently does it occur. The statistics of short storms of high intensity in Great Britain are expressed by Bilham's formula, as follows:—

$$N = 1 \cdot 25t(r + 0 \cdot 1)^{-3 \cdot 55}$$

where:

N = number of occurrences of storms of this intensity in 10 years.
t = time of duration in hours.
r = rainfall in inches during t.

The storm which produces the greatest run-off from any drainage area is that which has a duration equal to the 'time of concentration,' the time of concentration being the time taken for the water to run from the most distant part of the drainage area to the point at which the flow is desired to be known. This time of concentration is calculated by finding the velocity of flow

through each component length of the main sewer and multiplying by the length. To this is added a 'time of entry' of about 3 minutes, to allow for water to run over roofs and ground and find its way to the sewers.

For many years a rough estimate of the intensity of rainfall was made by use of the Ministry of Health formulæ, as follows:—

$$R = \frac{30}{t + 10}$$ for storms of duration varying from 5 to 20 minutes.

$$R = \frac{40}{t + 20}$$ for storms of duration varying from 20 to 100 minutes.

where:

R = inches of rainfall per hour.

t = duration of storm in minutes.

These formulæ are limited in application and are not strictly accurate, and the figures given in Table IV are recommended for determining the intensities of storms for the purposes of surface-water sewer design.

TABLE IV.—INTENSITIES OF STORMS FOR SEWER DESIGN PURPOSES.

Duration of Storm in Minutes.	Rainfall Intensity in Inches per Hour.	Duration of Storm in Minutes.	Rainfall Intensity in Inches per Hour.
5	2·16	48	0·53
5·5	2·05	50	0·52
6	1·95	52	0·50
6·5	1·85	54	0·49
7	1·76	56	0·48
7·5	1·69	58	0·47
8	1·62	60	0·46
8·5	1·56	62	0·45
9	1·50	64	0·44
9·5	1·46	66	0·43
10	1·41	70	0·42
11	1·33	75	0·40
12	1·26	80	0·38
13	1·20	85	0·37
14	1·14	90	0·35
15	1·09	95	0·34
16	1·05	100	0·33
17	1·01	105	0·32
18	0·98	110	0·31
19	0·94	120	0·295
20	0·91	130	0·28
21	0·88	140	0·27
22	0·86	150	0·26
23	0·83	160	0·25
24	0·81	170	0·24
25	0·79	180	0·23
26	0·77	190	0·22
27	0·76	200	0·21
28	0·74	220	0·20
29	0·72	240	0·19
30	0·71	260	0·18
31	0·69	280	0·17
32	0·68	315	0·16
33	0·67	350	0·15
34	0·65	390	0·14
35	0·64	440	0·13
36	0·63	500	0·12
37	0·62	570	0·11
38	0·61	660	0·10
39	0·60	780	0·09
40	0·59	950	0·08
42	0·57	1150	0·07
44	0·56	1500	0·06
46	0·54		

The impervious area is found as follows. Firstly, the impermeability factor of the area is determined by measuring the roof and paved areas of a sample area and multiplying these by impermeability factors for each type of surface, as given in Table V.

TABLE V.—IMPERMEABILITY OF SURFACES.

Type of Surface.	Impermeability Factor.
Watertight roof surfaces	0·70 to 0·95
Asphalt and other dense pavements in good order	0·85 to 0·90
Stone, brick and wood-block pavements with tightly cemented joints	0·75 to 0·85
Stone, brick and wood-block pavements with open or uncemented joints	0·40 to 0·70
Cobblestone pavements	0·40 to 0·50
Macadam roadways	0·25 to 0·60
Gravel roads and walks	0·15 to 0·30
Parks and open spaces, lawns, meadows, etc.	0·05 to 0·30
Wooded areas	0·01 to 0·20

Thus, the impermeability factor for the drainage area can be found, and this, multiplied by the gross acreage, gives the 'impervious area' in acres.

The run-off from the drainage area can then be found by the Lloyd-Davies formula, as follows:

$$Q = 60 \cdot 5 \times Ap \times R$$

where:

Q = cubic feet per minute run-off from area.
Ap = impervious area, or gross area in acres multiplied by impermeability factor.
R = inches of rainfall per hour.

The Lloyd-Davies method is essentially one of trial and error for, until the diameter of sewer has been decided, the velocity of flow cannot be known; until the velocity of flow is known the time of concentration cannot be known; until the time of concentration is known the intensity of rainfall cannot be known and the diameter of sewer cannot be calculated. The designer therefore selects a size of sewer, calculates the run-off according to the Lloyd-Davies formula, selects a new size of sewer, and so on, until the appropriate size has been found.

The calculation is made for each small component area of the total drainage area so as to determine the sizes of the branch sewers. But the flows in the main sewers are not the total flows of the branch sewers so determined, and must be calculated afresh, because the main sewers are influenced by longer times of concentration, and therefore the applicable rainfall intensities are smaller.

The above is the outline of the Lloyd-Davies method. To this have been added considerable elaborations of theory to make allowance for irregular distribution of impervious area.

Soakaway Systems for Surface Water.

Where the subsoil is permeable, roof water is discharged to soakaways, and in such areas water from road surfaces may also be soaked away provided there are no particular risks to water supplies. If, owing to the use of soakaways for all surface-water, it is desired to exclude surface-water from the soil sewers, a 'nominal' surface-water sewer must be provided, otherwise property owners have the right to discharge surface-water to the soil sewers. The method adopted in such circumstances is to construct soakaways under the road at intervals of about 300 ft. and to connect them by 6-in. diameter linking sewers to which are connected gullies. These linking sewers also serve as overflows in the event of any soakaway becoming surcharged. An extra-large soakaway is constructed at the lowest point on every linked system of soakaways.

Soakaways consist of circular chambers corbelled in at the top or covered with slabs and provided with heavy manhole covers. The walls are of 9-in. open-steined brickwork; or else of alternate rings of four courses of brickwork in header bond in cement mortar and four courses of dry brickwork in header bond, the vertical joints being kept open with pebbles. The walls are given foundations of concrete, but generally the floors are not paved.

Soakaways should not be filled in with rubble or other material, as this robs them of storage capacity. The storage capacity usually allowed is the equivalent of ½-in. of rainfall over the impervious area drained. Any storage below the natural level of the subsoil water is not included.

Ventilation of Sewers.

Sewers are ventilated for the purposes of relieving air-pressure so as to permit free flow of sewage and to prevent the breaking of the seal of traps on the fittings in buildings. Ventilation is provided also to remove dangerous or objectionable gases or vapours.

The gases found in sewers are partly the results of decomposition and include methane (which is explosive) and hydrogen sulphide (which is very poisonous and which also attacks Portland cement). Petrol vapour, due to petrol finding its way to the sewers from roads and garages, or even from private houses, is not only explosive but poisonous. Coal gas and dangerous gases from the subsoil can also find their way to the sewers.

The means of obviating all these dangers are :

(1) Designing sewers to self-cleansing gradients so that sludge does not settle and methane and hydrogen sulphide are not formed by the decomposition of septic sediment.

(2) Control to prevent discharge of petrol, etc., to the sewers. (This includes enforcing of the provision of petrol interceptors at garages and the control or discharge of trade effluents.)

(3) The use of full safety precautions by men entering sewers. See Ministry of Health publication : *Accidents in Sewers : Report on the Precautions Necessary for the Safety of Persons entering Sewers and Sewage Tanks.*

(4) Adequate ventilation to remove any gases or vapours which may have collected.

In areas where the local authorities do not require intercepting traps between the drains of premises and the sewers, the sewers are very adequately ventilated by the stack-pipes at the heads of all soil drains, and therefore no other ventilation is necessary. Where interceptors are used, the drainage systems to premises are ventilated between fresh-air-inlets and stack-pipes, but the sewers, being isolated by the traps, need additional ventilation. This is usually provided by ventilating columns placed at the heads of all branch sewers and at key points or main intersections on the sewerage system. Such columns should be carried to well above eaves level of nearby buildings. Ventilation is given by ventilating manhole covers also, where these can be used without causing nuisance.

The direction of flow of air in sewers cannot be predicted because it is mostly due to local differences of barometric pressure and the wind : theoretically it is not in the same direction as the wind. Less important causes of air flow are the ebb and flow of the sewage displacing air, the drag of the sewage flowing downhill and the tendency for warm moist air to flow up the gradient.

Flushing of Sewers.

The top ends of branch sewers generally receive little flow and therefore some provision should be made for flushing them. An economical arrangement is to provide a disc-valve in the manhole at the head of every branch sewer in order that the manhole may be filled with water from a tank vehicle or hydrant to a mark on the side of the manhole and the water released by opening the valve.

When sewers are unavoidably laid at inadequate gradients at which it is expected that silting will take place, or when inverted siphons are constructed, flushing tanks should be installed. An automatic flushing tank consists of a storage tank incorporating a flushing siphon which discharges to the nearest manhole. Water is delivered to a separate chamber isolated from the flushing tank by a trap. Company's water is metered and the flow regulated by a bib-tap or needle valve. An arrangement should be made for air to find its way to the flushing tank, otherwise the siphon will not function properly.

A suitable capacity of flushing tank is one-tenth of the total cubic capacity of the length of sewer to be flushed.

Construction of Sewers.

Small diameter sewers are constructed of salt glazed ware pipes (B.S. 65) ; salt glazed glass (vitreous) enamelled fireclay pipes (B.S. 540), the last being more commonly used in the North of England and in Scotland ; concrete pipes, reinforced or otherwise, with socketted joints or ogee joints (B.S. 556) ; cast-iron pipes (B.S. 78) ; and spun cast-iron pipes (B.S. 1211).

Stoneware and fireclay pipes are used for the majority of small diameter sewers : concrete pipes are less expensive for diameters of about 12 ins. upwards. Cast-iron pipes are used for rising mains, siphons, or for sewers laid in bad ground, or as an alternative to stoneware or concrete pipes surrounded with concrete.

The following are the normal minimum requirements of the Ministry of Health as regards concrete protection of salt glazed ware, fireclay or concrete pipes (numbered as on Ministry of Health Form K29) :

'(2) All pipes and tubes in heading or with 20 ft. or more of cover in trenches to be surrounded with at least 6 ins. of concrete.

TABLE VI.—PROPORTIONS OF CAST-IRON SEGMENTAL SEWERS.

Internal dia. of sewer	5′ 0″	5′ 3″	5′ 6″	6′ 0″	6′ 6″	7′ 0″	7′ 6″	8′ 0″	8′ 6″	9′ 0″	9′ 6″	10′ 0″
Internal dia. of iron	5′ 5″	5′ 8″	5′ 11″	6′ 5″	6′ 11″	7′ 5″	7′ 11″	8′ 5″	8′ 11″	9′ 5″	9′ 11″	10′ 5″
External dia. of iron	6′ 1″	6′ 4″	6′ 7″	7′ 1″	7′ 7½″	8′ 1½″	8′ 7½″	9′ 2″	9′ 8″	10′ 2½″	10′ 8½″	11′ 2½″
Thickness of iron	¾″	¾″	¾″	¾″	¾″	¾″	¾″	¾″	¾″	¾″	¾″	¾″
Depth of flanges	3¼″	3¼″	3¼″	3¼″	3½″	3½″	3½″	3¾″	3¾″	4″	4″	4″
Thickness of flanges at base	1″	1″	1″	1⅛″	1⅛″	1⅛″	1⅛″	1⅛″	1⅛″	1⅛″	1⅛″	1⅛″
Thickness of flanges at edge	⅞″	⅞″	⅞″	1″	1″	1″	1″	1″	1″	1″	1″	1″
Dia. of bolt circle	5′ 8¼″	5′ 11¼″	6′ 2¼″	6′ 8½″	7′ 2½″	7′ 8½″	8′ 2½″	8′ 8¾″	9′ 2¾″	9′ 9″	10′ 3″	10′ 9″
No. of bolts	19	23	23	23	23	27	29	29	34	36	39	39
Size of bolts, dia.	¾″	¾″	¾″	⅞″	⅞″	⅞″	⅞″	⅞″	1″	1″	1″	1″
Size of bolts, length	4″	4″	4″	4″	4¼″	4¼″	4¼″	4¼″	4¼″	4¼″	4¼″	4¼″
No. of ordinary segments (excluding 1 key and 2 tee segments), per ring	3	4	4	4	4	4	4	4	5	5	5	5

'(3) Subject to (2) all pipes and tubes with over 14 ft. of cover to be bedded on and haunched with at least 6 ins. of concrete to at least the horizontal diameter of the pipe or tube. Any splaying of the concrete to be above that level.

'(4) Subject to (2) all pipes and tubes of 18 ins. diameter and over to be bedded on and haunched with at least 6 ins. of concrete to at least the horizontal diameter of the pipe or tube. Any splaying of the concrete to be above that level.

'(5) Subject to (6) all pipes and tubes under 18 ins. in diameter and with less than 14 ft. of cover may be laid without concrete, if the joints are of the socket or collar type (but concrete tubes with OG joints are permissible when laid as in (2), (3), (4) or (6).

'(6) All pipes and tubes with less than 4 ft. of cover under roads, or 3 ft. not under roads, to be surrounded with at least 6 ins. of concrete.'

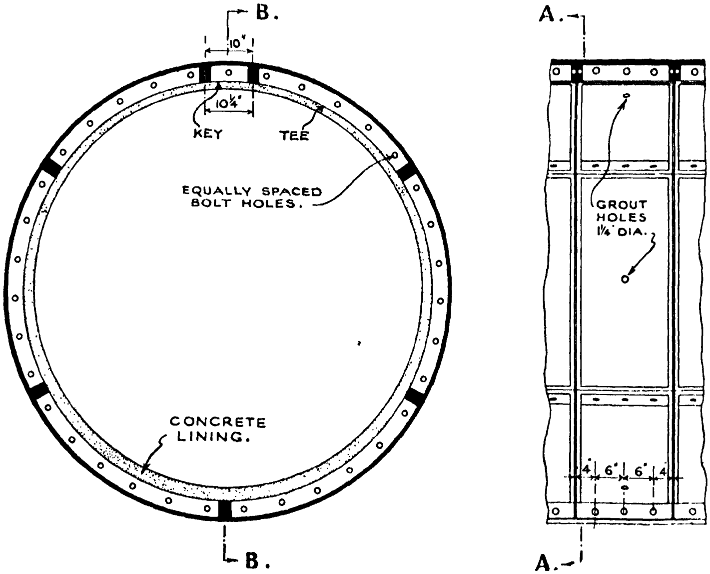


FIG. 2.—Cross and Longitudinal Sections of Cast-iron Segmental Sewer.

Sewers of large diameter are constructed of concrete pipes, mass concrete, mass concrete lined with brickwork, reinforced concrete, brickwork (special radial bricks sometimes being used), cast-iron pipes, bolted cast-iron segments, bolted pre-cast concrete segments and other special forms of construction. The circular cross-section is generally preferred for all sizes and types of sewer except very large sewers involving the use of brickwork, when special shapes such as culverts with vertical sides and arched crowns and inverts, horse-shoe shaped culverts, U-shaped culverts and other forms are preferred, either for their hydraulic properties or for ease of construction.

Limiting Diameters.

Model Byelaws do not permit private sewers to have diameters of less than 6 ins. and public sewers are never smaller. Sewers are seldom made much larger than 10 ft. diameter, it being usual to lay two or more in parallel when a 10-ft. diameter sewer is inadequate to take the flow.

Manholes.

The Ministry of Health requires that manholes shall be provided at all changes of gradient and direction of sewers of small diameter, and at distances apart not exceeding 120 yds. Model

Byelaws require that manholes on private sewers shall not be more than 100 yds. apart. Between manholes all small diameter sewers—*i.e.* less than 30 ins. diameter—must be laid in straight lines and at even gradients. Larger diameter sewers may be laid to curves between manholes, and manholes are sometimes spaced at longer intervals, *e.g.* 500 ft., but it is good practice to maintain intervals of not exceeding 360 ft. regardless of diameter.

Manholes are means of access to small diameter sewers for rodding purposes and for entering large sewers. The chamber of a manhole on a small sewer should not be less than 4 ft. 6 ins. long by 3 ft. wide (or 3 ft. 6 ins. diameter) and 6 ft. high from the top of the benching, except where headroom is restricted by depth below ground level. The invert of the manhole should be shaped to the invert of the sewer, the sides of the channel brought up vertically to the level of the crown

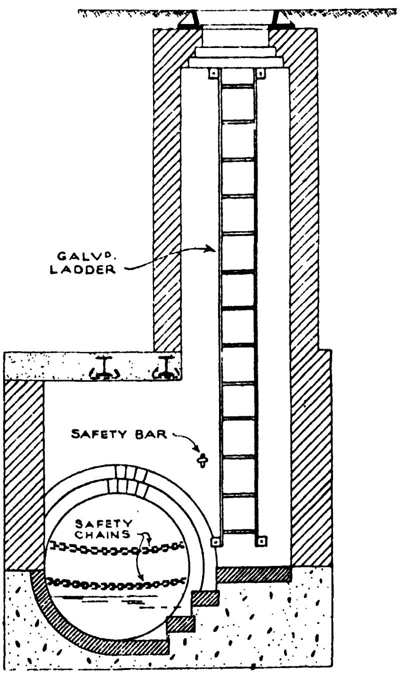


FIG. 3.—Side-entrance Manhole.

of the outgoing sewer, from which point they should turn over sharply to form the benchings. The benchings should slope evenly at about 1 in 6 towards the channels. The chambers of deep manholes should be approached by vertical shafts, the measurements of which should be 2 ft. 3 ins. by 2 ft. 7½ ins., the last measurement being taken from the face of the wall on which is fixed the ladder (or step-irons). Covers should have clear openings of not less than 20 ins. diameter.

Manholes on large sewers should be 'side-entrance' manholes, *i.e.* they should not lead directly down into the sewer but should terminate at a platform to one side of the sewer, the invert of the sewer being approached by a flight of steps or in some similar manner.

Manholes are structures which are inherently strong, and are very frequently designed extravagantly. Table VII gives the adequate thickness of manhole walls according to length of the longest wall and depth below ground or water level.

TABLE VII.—THICKNESS OF MANHOLE WALLS IN MASS CONCRETE AND BRICKWORK.
(Allowing tension of 60 lbs. per sq. in. approximately.)

External Water Pressure.		External Heavy Earth Pressure. (London Clay.)	
Depth below Water Level Feet.	$\frac{\text{Thickness of Wall}}{\text{Length of Wall}}$.	Depth below Ground Level Feet.	$\frac{\text{Thickness of Wall}}{\text{Length of Wall}}$.
11	$\frac{1}{5}$	16	$\frac{1}{6}$
17	$\frac{1}{4}$	23	$\frac{1}{5}$
31	$\frac{1}{3}$	36	$\frac{1}{4}$
		64	$\frac{1}{3}$

SEWAGE TREATMENT AND DISPOSAL.

Strength of Sewage.

The strength of sewage is estimated by chemical tests, but there is no test yet devised which gives an absolutely reliable and accurate indication of the difficulty of treatment. The accepted formulæ for determining strength of sewage are given in the Ministry of Health publication, *Methods of Chemical Analysis as Applied to Sewage and Sewage Effluents*, and are known as McGowan's formulæ. The index of strength as calculated by these formulæ is known as 'strength (McGowan).' The following is a quotation from the above mentioned publication.

'A good approximation can be arrived at for domestic sewage* by estimating the ammoniacal and albuminoid or organic nitrogen and the "oxygen absorbed" from $\frac{N}{8}$ permanganate in 4 hours at 27° C. and then applying the following formula to the figures so obtained:

For sewages:

(Ammon. + Organic N) × 4·5 + (Ox. abs. in 4 hours × 6·5 †).

For septic tank liquors:

(Ammon. + Organic N) × 4·5 + (Ox. abs. in 4 hours × 6·5 †).

For precipitation liquors:

(Ammon. + Organic N) × 4·5 + (Ox. abs. in 4 hours × 6·0 †).

'The subject is dealt with in the Fifth Report of the Royal Commission on Sewage Disposal Appendix IV, p. 1, *et seq.* (1910) (Cd. 4282), to which the reader is referred.'

Other than in Great Britain, and of recent years in Great Britain also, the strength of sewage has been estimated in terms of the five days bio-chemical oxygen demand (or five days B.O.D. value). This is found by determining the quantity of oxygen absorbed when sewage or sewage effluent is diluted with water containing oxygen and incubated at a constant temperature for five days.

There is no direct correlation between strength (McGowan) and B.O.D. value, for although, on the average, B.O.D. value increases in direct proportion to strength (McGowan), individual samples of various sewages show marked differences.

The Royal Commission described a sewage having a strength (McGowan) of 100 as being a medium strength sewage. A typical analysis of such a sewage is as follows:

Ammoniacal Nitrogen	3·53 parts per 100,000.
Albuminoid ,, ...	0·91 ,, ,, ,,
Total organic ,, ...	2·25 ,, ,, ,,
Oxidised ,, ...	trace
Total ,, ...	5·85 ,, ,, ,,
Oxygen absorbed at 27° C. (80° F.) in 4 hours . .	11·27 ,, ,, ,,
Chlorine	9·16 ,, ,, ,,
Solids in suspension	29·40 ,, ,, ,,

A sewage of medium strength probably has, on the average, a B.O.D. value of 35 parts per 100,000, or slightly less.

'* Apart from cases where large volumes of trade wastes, containing *e.g.* starch, glucose, or gas liquor, are present.

'† *Note.*—When N/80 permanganate has been used in the "oxygen absorbed test," this figure would have to be further multiplied by 1·6, in order to give an approximately correct result.'

The strength of sewage varies throughout the day, being at a maximum at about the same time as the maximum rate of flow, *i.e.* at about midday, and being weakest during the night. When samples are taken for the purpose of determining strength, as a prelude to design, they must

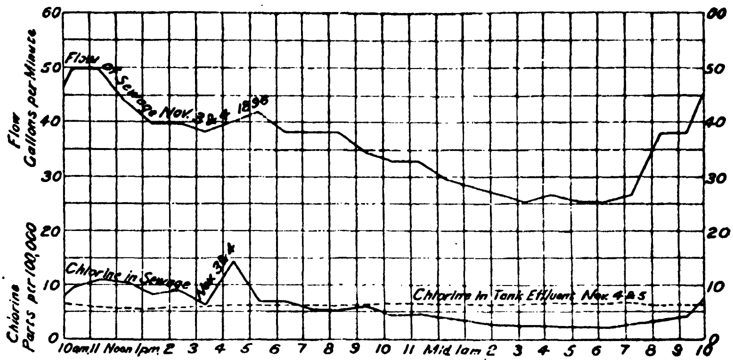


FIG. 4.—Typical Curves showing Variation of Rate of Flow and Strength of Sewage throughout the day.

be taken at regular intervals over twenty-four hours, the size of each sample being in proportion to the rate of flow at the time. The component samples are then thoroughly mixed to form the final sample for analysis.

Standards of Treated Effluents.

In 1913 the Royal Commission issued their 8th Report, in which they summarised their conclusions as follows :

(*a*) The law should be altered so that a person discharging sewage matter into a stream shall not be deemed to have committed an offence under the Rivers Pollution Prevention Act, 1876, if the sewage matter is discharged in a form which satisfies the requirements of the prescribed standard.

(*b*) The standard should be either the general standard or a special standard which will be higher or lower than the general standard as local circumstances require or permit.

(*c*) An effluent in order to comply with the general standard must not contain as discharged more than 3 parts per 100,000 of suspended matter, and with its suspended matters included must not take up at 65° F. (18·3° C.) more than 2·0 parts per 100,000 of dissolved oxygen in five days. This general standard should be prescribed either by Statute or by Order of the Central Authority, and should be subject to modifications by that Authority after an interval of not less than ten years.

(*d*) In fixing any special standard the dilution afforded by the stream is the chief factor to be considered. If the dilution is very low it may be necessary for the Central Authority, either of their own initiative or on application by the Rivers Board, to prescribe a specially stringent standard, which should also remain in force for a period of not less than ten years.

(*e*) If the dilution is very great the standard may, with the approval of the Central Authority, be relaxed or suspended altogether. Our experience leads us to think that as a general rule, if the dilution, while not falling below 150 volumes, does not exceed 300, the dissolved oxygen absorption test may be omitted and the standard for suspended solids fixed at 6 parts per 100,000. To comply with this test no treatment beyond chemical precipitation would ordinarily be needed. If the dilution, while not falling below 300 volumes, does not exceed 500, the standard for suspended solids may be further relaxed to 15 parts per 100,000. For this purpose tank treatment without chemicals would generally suffice if the tanks were properly worked and regularly cleansed. These relaxed standards should be subject to revision at periods to be fixed by the Central Authority, and the periods should be shorter than those prescribed for the general or for the more stringent standards.

(*f*) With a dilution of over 500 volumes all tests might be dispensed with, and crude sewage discharged, subject to such conditions as to the provision of screens or detritus tanks as might appear necessary to the Central Authority.

The standards mentioned in (*e*) and (*f*) are not intended to apply to effluents discharged into streams which are used for drinking water or to streams the water of which, at the point of discharge of the sewage effluent, normally takes up more than 0·2 parts per 100,000 of dissolved oxygen in five days at 60° F. Up to the present time the proposals contained in the Report have not been embodied in any Act of Parliament, and have therefore no official authority, and must be treated as suggestions only.

Quantity of Sewage to be Treated: Separation and Treatment of Storm Water.

Present-day practice is to allow for full treatment of all sewage arriving at the works up to three times dry-weather flow. This applies to the sewage from both separate and combined sewers.

At works treating sewage from combined or partially separate sewerage systems, any flow in excess of three times dry-weather flow is passed to stand-by or storm tanks, where it is settled and stored until such time as, the rate of flow having reduced, the stored sewage can be passed through the works for full treatment. If the quantity of storm water exceeds the capacity of the storm tanks, the storm tanks function as sedimentation tanks and the settled effluent is passed to the outfall without further treatment, or with such treatment, *e.g.* treatment on land, as can be given without undue expense.

Storm tanks normally have a capacity of six hours dry-weather flow and receive the flow from separation weirs, which are placed below the screens and detritus tanks but come before the remainder of the treatment works. The tanks are usually in the form of long flat-bottomed rectangular sedimentation tanks, with weirs at the outlet ends, protected by floating scumboards, and floating arms (also protected by floating scumboards), for the decanting of the tank contents after a storm. Sludge is removed by sweeping to the sludge outlets, either manually or mechanically.

At sewage treatment works which derive their flows from separate systems from which little storm or infiltration water is expected, storm tanks are usually omitted, the whole of the flow arriving at the works being separated after the sedimentation tanks, or passed through the entire works.

Storm overflows for the discharge of crude sewage without treatment of any kind are not permitted at sewage treatment works: once the sewage has arrived at the treatment works, it must be given treatment at least by settlement in storm tanks or the sedimentation tanks.

General Outline of Treatment.

The purpose of sewage treatment is to remove as much as possible of the organic matter from the sewage and to oxidise the remainder, converting it into inorganic matter in order that it does not putrefy, causing a nuisance, or deplete the watercourse, into which it is discharged, of oxygen, thereby destroying fish life. The process of sewage treatment, therefore, consists of the mechanical removal of suspended solids and the oxidising of dissolved solids, with the aid of bacteria and other organisms.

Sewage treatment works normally consist of:

1. Screens.
2. Detritus tanks or channels.
3. Sedimentation tanks.
4. The aeration unit (land treatment, percolating filter treatment or activated sludge treatment).
5. Humus tanks (following percolating filter treatment), or final sedimentation tanks (following activated sludge treatment).

In addition, there are ancillary works, including plant for treating and disposing of the sludge that has been collected in the sedimentation tanks.

Formerly, the design of sewage treatment works and the determination of the proportions of the component units were (in Great Britain) almost entirely based on the recommendations on the Royal Commission on Sewage Disposal and on various figures, other than recommendations, which appeared in the Reports or the Appendices thereto and which, it was believed with or without justification, were the specific requirements of the Ministry of Health. Thus, for many years design became stereotyped and little progress was made. Now, under the influence of American and German practice, etc., and as a result of extensive research, practice has changed considerably and only in part remains the same as in the days of the Royal Commission. The following description of works, and methods of determining capacities, are in accordance with advanced but sound current practice.

Screens.

Opinion on screens varies considerably. Some authorities prefer to use screens with ½- to ¾-inch clear spaces so as to remove as much as possible of the larger organic material before further treatment of the sewage. Others install screens with wide spaces sufficient only for the purpose of protecting pipe-lines and pumping plant from large solids. Sometimes screens are omitted. At some works, screenings are removed by fine screens and, after maceration, returned to the flow of sewage. As an alternative to this process, 'Comminutors' which combine the offices of screen and macerator may be installed.

At small works hand-raked screens are most suitable. These should be set at an angle and so arranged that the screenings may be raked upwards into a rolled steel channel from which they may be shovelled or swept into barrows. At large works, *i.e.* those serving more than 20,000 persons, mechanical screens (of many designs) are installed. The submerged area of a hand-raked screen should not be less than 1½ ft. super per thousand head of population. The sizes of mechanically raked screens should be such that, at the maximum rate of flow, the velocity through the bars is in the region of 1½ ft. per second.

When screenings are not disintegrated and returned to the sewage, they may be passed to sludge digestion tanks (if provided), discharged to trenches and covered with earth, discharged with the sewage sludge, or destroyed by incineration.

Detritus Tanks and Channels.

Detritus tanks are provided for the purpose of removing heavy inorganic sediment such as road grit, gravel and chippings which, if permitted to pass to the sedimentation tanks, might pack and interfere with sludging. Detritus tanks were of more importance in the days of water-bound macadam roads and the prevalence of combined systems of sewerage. At the present time it is considered that they are not necessary at works treating purely domestic sewage from separate systems, although at small works they are still installed mainly for the purpose of housing and operating hand-raked screens. At all large works the installation of detritus tanks is usual. They are a necessity when sewage from combined systems or industrial areas is to be treated.

At one time the orthodox practice in Great Britain was to provide detritus tanks not fewer than two in number and having a total capacity of $\frac{1}{10}$th dry-weather-flow. These were crude pits or chambers arranged for decanting of top water and sludging. They were very unsatisfactory, being too large and therefore settling large quantities of organic material with the inorganic detritus, which often led to offensive conditions. Under the influence of American practice, various forms of constant velocity detritus channels have come to be favoured and will probably replace the simple detritus tank at all except those small works where the detritus tank serves as screen chamber and its size is mainly determined by the proportions of the screen.

The constant velocity detritus channel is based on the principle that at a velocity of flow of 1 ft. per second, comparatively fine siliceous grit will settle and remain settled, while organic particles both large and small are swept along by the current.

The means by which a constant velocity can be maintained are:

1. Manual or automatic-electric control in order that the number of channels in use can be varied in almost direct proportion to rate of flow.
2. Control of depth of flow by a flume or weir at the outlet ends of channels, and specially proportioning the cross-sections of the detritus channels in order that the velocity of flow therein remains constant at all depths of flow.
3. A combination of the above methods.

A much favoured form of constant velocity detritus channel is the channel of parabolic section, the flow through which is controlled by a rectangular standing-wave flume the invert of which is level with the invert of the channel.

The discharge of a rectangular standing-wave flume is approximately according to the formula:

$$\text{Cubic feet per second} = 3 \cdot 09\ B.H.^{\frac{3}{2}}$$

where:

B = width of flume in feet.
H = depth from invert of flume to upstream top water level, in feet.

The width at any point above the invert of the detritus channel is given by the formula:

$$X = \frac{3Q}{2H}$$

where:

X = width of channel at water level in feet.
H = depth of flow in feet.
Q = rate of flow at that depth in cubic feet per second.

In order that the grit may have time to settle the channels should not be short, and preferably the length of each channel should not be less than thirty times the maximum working depth.

At large works, grit can be removed by travelling grit pump or by mechanical dredger. At small works, the channel to be cleansed is allowed to run dry and is then dug out manually. No fewer than two channels should be provided and, generally, each channel should be controlled by its own standing wave flume or weir.

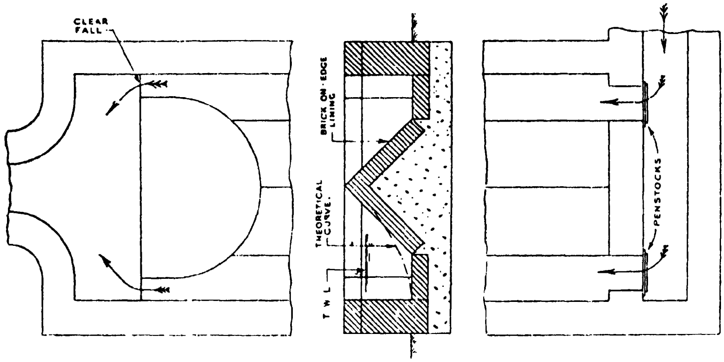


FIG. 5.—Plan and Cross-section of Constant Velocity Detritus Channel for Small Sewage-works.

Quiescent Sedimentation.

Formerly, quiescent sedimentation tanks were largely used. These were worked on the fill-and-draw principle: a tank was filled with sewage, allowed to stand for a sufficient period for settlement, the top water then decanted with the aid of a floating arm, after which the sludge was swept out manually. For this purpose, not fewer than four, and preferably eight tanks were provided, the total capacity of the battery being in the region of 16 hours' dry-weather flow. Now, continuous flow sedimentation, in which the sewage flows continuously through the tanks, is always provided. Quiescent tanks were identical, in design, with storm tanks, as described above.

Continuous Flow Sedimentation.

The detail design of sedimentation tanks is far more important than their capacities. The latter can be varied over a very wide range without causing an excessive difference of efficiency. At a time when English sedimentation tanks were being made to capacities of 10 to 15 (or even more) hours' dry-weather flow, American designers were content with capacities of 1½ to 2 hours' dry-weather flow and were obtaining satisfactory results.

At the present time those English designers who still design in terms of capacity allow 6 or 8 hours' dry-weather flow in many instances. But it is becoming appreciated that the design of sedimentation tanks is not a matter of fixing a storage capacity for retention period, but the proper design of the tank and its components for convenient and efficient operation.

The efficiency of a sedimentation tank is mostly dependent on the design of inlets and outlets. The inlet should be such that it does not produce a jet or swirling action, or set up eddies: the sewage should enter the tank quietly and at a low velocity, and any turbulence should be dissipated in a stilling-chamber. The outlets of all sedimentation tanks should be weirs of such length that the flow over them is very shallow. These weirs should be protected by scumboards which should, however, be placed sufficiently far from them not to cause a local upward velocity.

The capacity of a sedimentation tank has, however, some effect on efficiency, the latter varying more or less in direct proportion to the fourth or fifth root of the former. Of more importance is the surface area, which generally should be not less than 1 ft. super for every 300 gallons dry-weather flow, and preferably 50 per cent. greater, according to current practice.

The types of sedimentation tanks in use include:

1. Rectangular longitudinal flow tanks, 5 or more feet deep and about five times as long as they are wide. These are usually mechanically swept.

2. Pyramidal bottomed tanks with peripheral weirs and central inlets (the bottoms sloping at about 60° to the horizontal), sludged under hydrostatic head.

3. Various designs of flat-bottomed or conical-bottomed circular tanks provided with automatic sludging gear.

Pyramidal-bottomed tanks are very largely used at works of all magnitudes. The considerations in their design are that the surface area shall be adequate and that the capacity for sludge storage shall be well below the central inlet and not occupying more than one-half the capacity of the tank. The quantity of sludge to be stored in the lower part of the tank can be taken as being 2 gallons per head of population in most instances. This is approximately one week's storage.

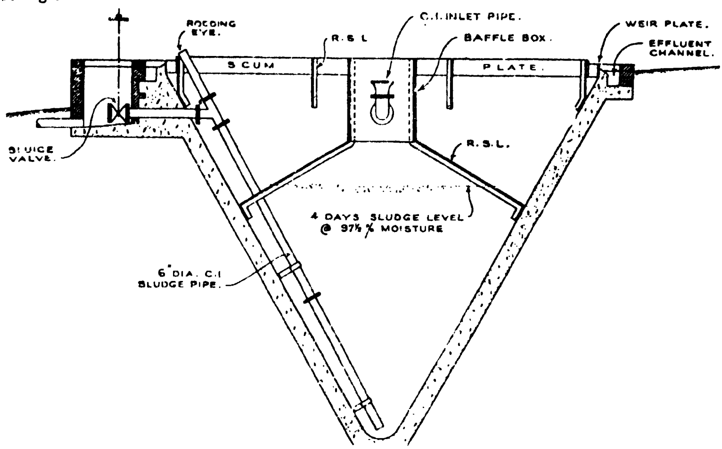


FIG. 6. –Section of Pyramidal-bottomed Sedimentation Tank.

Chemical Precipitation.

Chemical precipitation is rarely employed owing to the expense of, and difficulty in obtaining, chemicals. Generally, it is not necessary except when treating abnormal sewages.

The precipitants chiefly used are lime in a caustic state, 10 to 12 grains per gallon of sewage; lime and sulphate of iron or green copperas in the proportion of 6 grains to 3 grains of lime with 3 grains to 1 grain of copperas; lime and alumina in the proportion of 2 of the former to 1 of the latter, usually about 8 and 4 grains respectively; alumino-ferric, either with or without lime, proportion 10 to 7 grains lime and 5 to 3 grains alumino-ferric – if used without lime, up to 8½ grains per gallon.

Treatment and Disposal of Sewage Sludge.

The quantity and moisture content of the sludge settled in primary sedimentation tanks varies considerably according to the solids content of the sewage, the efficiency of sedimentation and the method of sludge removal. For design purposes it usually suffices to assume that the sludge from pyramidal-bottomed sedimentation tanks sludged under hydrostatic head amounts to ½ gallon per head of population per day and has a moisture content of 97½ per cent. and that the sludge from a mechanically swept tank amounts to ¼ gallon per head per day and has a moisture content of 95 per cent.

The methods of disposal of sludge include :—

1. Discharge into trenches and covering over, or ploughing into land.
2. Distribution by pipe-line to nearby farm land for utilisation as fertiliser.
3. Drying on drying beds.
4. Drying with the aid of filter presses or vacuum filters.
5. Heat drying.
6. Digestion followed by drying.
7. Heat treatment followed by filter pressing.
8. Incineration.
9. Dumping at sea.

Of these methods, drying on beds either of raw sludge or of sludge after digestion, is by far the most common. Sludge drying beds consist of shallow lagoons, with or without concrete floors, according to the site, and bottomed with about 9 ins. of clinker. The clinker is graded from 1 to 1½ ins. diameter of particle and topped with about 2 ins. of ¼-in. to ½-in. material. It is underdrained with agricultural tiles. One superficial yard of bed is required for every five to seven

head of population. At all works there should be four, and preferably not fewer than six individual beds, to permit rotation of the drying process. The sludge is discharged on to the beds to a depth of about 9 ins., left to dry, and removed for further drying on a dump after it has become 'spadeable.' Dried sludge has a moisture content of about 50 per cent. and weighs about 14 cwt. per cubic yard.

Sludge Digestion.

Sludge digestion has the following advantages :—

1. The quantity of sludge to be disposed is reduced.
2. The process renders the sludge inoffensive as regards odour.
3. Digested sludge usually, but not always, is dewatered more easily than raw sludge.
4. Digested sludge when dried is more useful for agricultural purposes than raw sludge.
5. The gas given off during digestion can be used for power production or for other purposes.

Sludge can be digested at day temperature, heated to a temperature of about 85° F. (mesophilic digestion), or heated to about 115° F. (thermophilic digestion). The last method is rare.

The capacities of sludge digestion tanks are as recommended in Table VIII. Two-stage digestion is generally preferred.

TABLE VIII.—CAPACITIES OF SLUDGE DIGESTION TANKS.

Type of Treatment.	Cu. Ft. Capacity per Head of Population.	
	Domestic Sewage.	Containing Trade Waste.
Single-stage digestion at day temperature .	3 to 4	6 to 8
Single-stage digestion at 85° F. . .	1½ to 2	4 to 5
Capacity of primary tank of two-stage digestion at day temperature . .	2 to 2½	3
Capacity of secondary tank of two-stage digestion at day temperature . .	1 to 1½	3
Capacity of primary tank of two-stage digestion at 85° F.	¾ to 1	2 to 2½
Capacity of secondary tank of two-stage digestion at 85° F.	¾ to 1	2 to 2½

The quantity of sludge gas that can be obtained by digestion from tanks of the above capacities varies round about 1 cu. ft. per head of population per day. The calorific value is high, *e.g.* in the region of 630 B.Th.U.'s per cu. ft. The gas can be used for power purposes, the exhaust gases from the gas engines containing usually sufficient heat to heat the sludge in the tanks. There are various methods of heating and stirring the sludge.

Closed, heated and stirred sludge digestion tanks suitable for collection of gas are manufactured by Ames Crosta Mills & Co. Ltd. and Dorr-Oliver Co. Ltd.

Sludge liquor from sludge drying beds and decanted at various levels from digestion tanks is returned to the flow of crude sewage for treatment, for it is very strong.

Land Treatment.

The clarified effluent from sedimentation tanks can be effectively oxidised by treatment on land, and this process can be economical where land is cheap and suitable for the purpose. Land treatment is, however, comparatively little used at the present time.

There are two distinct methods of using land : 'land filtration' and 'broad irrigation.' Where a suitable light friable soil can be obtained, filtration is the process adopted, the sewage passing downward through the soil to underdrains. If, on the other hand, the available land be all of a retentive character, which will not allow of the easy passage of water, the method adopted is to pass the tank effluent over the surface of the land by irrigation and to pick it up again in suitable catchwaters and reapply it to a fresh plot, and subsequently to a third or even a fourth plot before final discharge into the stream : such land should not be underdrained, or only to a very slight extent.

The quantity of liquid which can be dealt with per acre of land depends on the character of the land and the strength of the sewage.

It is not possible to make any close estimate of the amount of land required in any particular circumstances until local experience has been gained. For new installations, the recommendations given by McGowan, Houston and Kershaw in their report to the Royal Commission should be followed.

'To summarise all our results within the limits of a few sentences is impossible, but we may say in conclusion, and speaking in general terms, that we doubt whether even the most suit-

able kind of soil worked as a filtration farm should be called upon to treat more than 30,000 to 60,000 gallons per acre per 24 hours at a given time (750 to 1,500 persons per acre); or more than 10,000 to 20,000 gallons per acre per 24 hours, calculated on the total irrigable area (250 to 500 persons per acre). Further, that soil not well suited for purification purposes, worked as a surface irrigation or as a combined surface irrigation and filtration farm, should not be called upon to treat more than 5,000 to 10,000 gallons per acre per 24 hours at a given time (125 to 250 persons per acre); or more than 1,000 to 2,000 gallons per acre per 24 hours, calculated on the total irrigable area (25 to 50 persons per acre).

It is doubtful if the very worst kinds of soil are capable of dealing even with this relatively small volume of sewage. The population per acre is calculated on 40 gallons of sewage per head per day. It is here assumed that the sewage is of medium strength, and is mechanically settled before going on to the land.'

Percolating Filter Treatment.

Percolating filter treatment is by far the most common method of sewage aeration and is used at works of all sizes and for the treatment of most types of sewage. Percolating filters have replaced the contact beds which were formerly used but which are now applied, in rare circumstances only, to the treatment of trade wastes.

A percolating filter consists of a bed of clinker, broken stone, slag or other suitable medium, having a depth of, generally, not less than 4 or not more than 9 ft. The required qualities of the medium are laid down in B.S. 1438. The floor of the bed is of concrete and is usually laid to a fall of about 1 in 200 and drained with agricultural tiles: sometimes a false floor of filter tiles is laid on the true floor. On this is laid 6 or 9 ins. of large medium or stones, 4 to 6 ins. diameter, and above that the remainder of the medium. For ordinary purposes the medium should be of 1½ to 2-in. grade throughout: for high-rate filters in the region of 3-in. grade. A percolating filter is not a mechanical filter (the name is misleading) and fine material should be avoided, particularly at the surface where it would cause ponding. Filters of ' fine ' material, as described in the Reports of the Royal Commission, are now obsolete.

The tank effluent is distributed over the filter by a sparge or distributor mechanism which is usually actuated by the hydraulic head of the sewage applied to it. Most filters are circular, because this shape accommodates simple rotating distributors: large beds are rectangular and provided with travelling distributors. Tipping tray mechanisms discharging to fixed distributor channels are used for the very small percolating filters incorporated in treatment works for isolated buildings, etc.

TABLE IX.—LOADING OF PERCOLATING FILTERS AFTER CONTINUOUS FLOW SEDIMENTATION.
(Works designed to treat up to three times dry-weather flow.)

Strength (McGowan)	Gallons Dry-weather flow per Cub. Yd. of Filter Medium.
30	222
35	190
40	167
45	148
50	133
55	121
60	111
65	102
70	95
75	89
80	83
85	78
90	74
95	70
100	67
110	60
120	56
130	51
140	48
150	44
160	42
170	39
180	37
190	35
200	33

The capacities of normal percolating filters are still determined more or less in accordance with the recommendations of the Royal Commission, except that whereas formerly it was not unusual to classify sewage as strong, medium or weak, and to allow stereotyped quantities of sewage per cubic yard according to the classification into which the sewage fell, there are now various methods of more accurately determining the amount of medium required. Of these methods, three may be recommended :—

1. Where the sewage is mainly domestic and the strength (McGowan) is known, and percolating filter treatment is preceded by continuous flow sedimentation, the capacity of the beds may be calculated in accordance with Table IX. Larger quantities of medium are required if the sewage contains inhibitory trade wastes. The quantity of sewage that can be treated per cubic yard of medium can be increased by one-third if chemical precipitation has been employed.

2. Where sewage will be mainly domestic but strength is not known, it usually suffices to allow about 0·537 cu. yds. of medium per head of population served.

3. Percolating filters have been designed in America in terms of cubic yards of medium per lb. weight of 5-days' B.O.D. per day. Lbs. B.O.D. per day is :

$$\frac{\text{Gallons per day} \times \text{B.O.D. value (pp. 100,000)}}{10{,}000}.$$

To obtain a Royal Commission effluent, 120 cu. ft. of medium should be allowed per lb. B.O.D. per day.

The above recommendations are based on the requirements that the works shall, at times, be able to treat up to three times dry-weather-flow and a ' Royal Commission effluent ' be obtained.

High Rate Filtration.

The methods of high rate filtration that have been used in this country include alternating-double-filtration, recirculation (in particular the Biofiltration process of Messrs. Dorr-Oliver Co. Ltd.), and the use of enclosed filters.

Alternating-double-filtration consists of the use of two batteries of percolating filters which are so arranged that they work in series, and the filters which are serving as primary stage of treatment can be periodically changed over with those that are serving as secondary stage. Large scale experiments have shown that filters so operated can treat (as regards B.O.D. reduction) much more sewage than normal filters. There is as yet, however, not sufficient information for it to be safe to make very definite statements as to the average capability of alternating double filtration plants compared with normal plants, but it is thought that they may be about three times as efficient.

The process of recirculation involves the return of treated effluent to the flow of untreated tank effluent, or sometimes crude sewage, in quantity equal to, or greater than, the flow of sewage. It would appear that recirculation of effluent in equal quantity to flow of sewage, using normal beds, is very effective. The Dorr process, however, includes methods of recirculation at high rates using special shallow filters and ' Clarifier ' sedimentation tanks, with the result that marked improvement of bed performance is secured.

It is claimed that the methods of alternating double filtration and recirculation involve lower horse-powers than activated sludge processes.

Enclosed percolating filters are deep filters, *i.e.* in the region of 14 ft. deep, roofed over and ventilated by electric fans, which, placed in the apex of each roof, blow downwards through the beds. Enclosed filters do not appear to be quite so efficient as alternating double filtration or recirculation filters, but they can treat twice as much sewage as ordinary percolating filters. The cost of roof construction is expensive.

Humus Tanks.

The oxidation that takes place in percolating filters is effected by bacteria and other organisms which form a slimy coating to the medium. In addition to this flora, other plant life develops, in particular growths of algæ at the surface of the beds. These growths are attacked by worms and larvæ of insects and, being broken up, are seasonally swept out of the percolating filters by the flow. The greatest quantity is noticeable in the spring, when the animal organisms recover from winter inactivity. Included with the vegetable matter are the remains of worms, larvæ, insects, together with dust washed from the medium. This material, known as humus, is collected in humus tanks before the effluent is discharged.

Humus tanks are similar to sedimentation tanks, and may be pyramidal bottomed or flat bottomed. Their surface areas and capacities are generally about half those of the sedimentation tanks for crude sewage. Scumboards are provided, because the humus tends to gas and rise to the surface if not removed shortly after settlement.

Activated Sludge Processes.

In the activated sludge processes, the bacteria which effect oxidation are not attached to particles of media as in a percolating filter, but adhere to, and combine to form, flocculi suspended in an aerated tank-effluent. The principle of activated sludge treatment involves the oxidation of

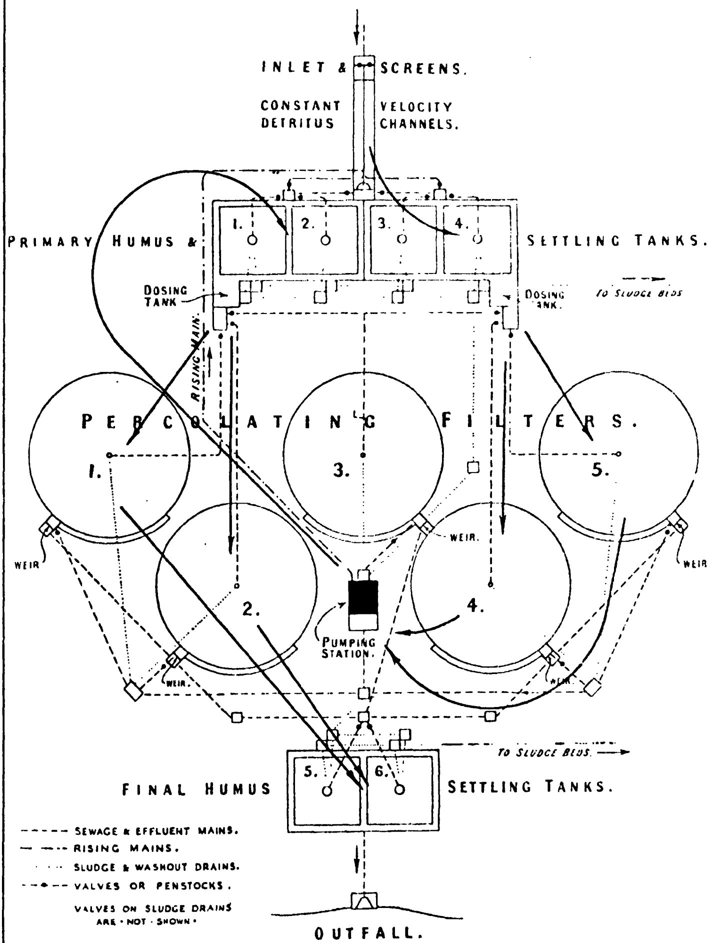


FIG. 7.—Percolating-filter Scheme capable of operation as Normal Scheme or Alternating-double-filtration Scheme.—The arrows show the direction of flow during one phase of double-filtration.

the sewage by bacteria, etc., the continued aeration of the sewage to restore the oxygen so absorbed, and the agitation of the sewage during oxidation so as to prevent the settlement of sludge and resulting local septicity.

There are two main systems of activated sludge treatment: the 'Diffused Air' system, involving the use of compressed air (the proprietary method of Activated Sludge, Ltd.); and the 'mechanical agitation' or 'surface aeration' system, in which aeration takes place at the surface. The latter system includes the 'Simplex' method (proprietary method of Ames Crosta Mills & Co. Ltd.), the Kessener process (developed by Dr. Kessener of Holland), and the Sheffield system or 'Bio-aeration' (the last is not a proprietary process).

Diffused Air System.—After close screening, and sedimentation, sometimes sufficient to remove the heaviest sludge only, the tank effluent is mingled with 'activated sludge' and passed into aeration tanks, which are sub-divided so as to form long channels about 10 to 15 ft. deep and of width equal to depth. One-tenth to one-twelfth of the floor surface of these channels consists of diffuser tiles through which compressed air is blown at a rate sufficient to oxidise the sewage and produce velocities of circulation sufficient to prevent settlement.

There are two main arrangements of diffusers: the 'ridge and furrow' system in which the floors of the channels are formed into longitudinal ridges and furrows, the diffusers being in the bottoms of the furrows; and the 'spiral flow' system in which the diffusers are placed to one side of each of the channels so as to produce a spiral rotation of the 'mixed liquor' as it flows from end to end. The quantity of air injected varies according to the strength of the sewage, the capacities of the tanks, the depths of the tanks and the conditions at the time. For design purposes, allowance should be made for the provision of 2 cu. ft. per gallon of sewage to be treated, or more if there are inhibitory trade wastes.

The capacity of aeration channels for a domestic sewage of medium strength is about 10 hours' dry-weather flow: for stronger or weaker sewages the storage capacity is increased or decreased respectively more or less in proportion to strength.

The aerated 'mixed liquor' is passed to final sedimentation tanks of capacity of not more than 6 hours' dry-weather flow, for removal of the activated sludge. This sludge is continuously withdrawn, and part is returned to the flow of sewage to form 'mixed liquor,' the surplus sludge being passed to the primary sedimentation tanks to be settled out with the crude sludge. It has been found advantageous to store activated sludge in 'reaeration channels' before returning it to the flow of sewage, and a variable number of aeration channels are reserved for this purpose. Activated sludge is recirculated at the rate of 8 to 20 per cent. of the flow of sewage.

'Simplex' System of Surface Aeration.—In the 'Simplex' system, the aeration tank is usually square on plan and has a pyramidal bottom. Aeration and circulation of the mixed liquor are effected by a screw propeller or aerating wheel which is partly submerged in the liquid at the centre of the tank. This draws mixed liquor from the bottom of the tank and sprays it over the surface. In small installations, final sedimentation tanks consist of pockets formed in the corners of the aeration tanks. In large installations the aeration tanks are arranged in series, discharging from one to anothe rand finally to a battery of sedimentation tanks. The aeration capacity is in the region of 16 hours' dry-weather flow for a sewage of average strength.

Kessener Process.—The Kessener process is similar to the 'Simplex' process in main principle, but the aeration takes place in a long channel in which spiral flow is produced by a stainless steel brush which extends the full length of each channel and, by rapid rotation, sprays the mixed liquor across the surface.

Sheffield System.—In this system sewage is made to travel at a velocity of about 1½ ft. per second through a channel about 4 ft. wide and deep, having a capacity of about 16 hours' dry-weather-flow. Part of the sewage is returned to be mingled with crude sewage at the inlet end of the channel, the remainder passes on to final sedimentation tanks for removal of the activated sludge.

The channel is folded backwards and forwards upon itself so as to occupy a small space. The motion is imparted to the mixed liquor by a series of paddle-wheels mounted on a common axle which crosses the channel in the centre of each reach. In an alternative system, paddle wheels are mounted at the bends at the ends of the component channels.

Sewage Treatment for Isolated Buildings.

The works for treating sewage from isolated buildings and from very small communities are somewhat different from other treatment works, because they have to be designed on the assumption that regular maintenance is impracticable. Works for small villages receive attention at least once or twice a week, if not every day; but plants in private ownership most often remain untouched and uninspected for weeks or even months at a time.

For the above reason, efficient continuous flow sedimentation tanks, which must be frequently sludged, cannot be used, and the septic tank, which is obsolete as regards municipal works, is substituted. Septic tanks can be sludged at very infrequent intervals of several months and give good effluents, provided that they are of adequate capacity and well designed. They are sludged by complete emptying, either to trenches or on to ploughed land, or else by cesspool-emptying

vehicles. If the last method is used, the septic tank should have a capacity that is an exact multiple of that of the local tank-vehicle. Tables X and XI give suggested capacities and proportions for septic tanks.

TABLE X.—HEAD OF POPULATION SERVED BY SEPTIC TANKS.

Periods between times of Sludging Months.	Capacity of Septic Tanks in Gallons.			
	750	1,500	2,250	3,000
6	10	20	30	40
4	15	30	45	60
3	20	40	60	80
2	30	60	90	120

TABLE XI.—PROPORTIONS OF SEPTIC TANKS.

Approx. Capacity. Gallons.	Length. Ft. In.	Breadth. Ft. In.	Depth. Ft. In.
750	7 10½	2 7½	5 9
1,500	11 3	3 9	5 9
2,250	13 10½	4 6	5 9
For 3,000 gallons capacity, use two 1,500-gallon tanks in parallel.			

The methods of treating tank effluent at these small works include percolating filter treatment and land treatment by either land filtration or broad irrigation. In addition to these, there is the controversial method of sub-irrigation. The 'Simplex' activated sludge method also has been applied to moderately small works for institutions.

The capacities of percolating filters and areas for land irrigation are generally the same as those recommended for larger works, except where there is reason to believe that the small works will be inefficient owing to some cause such as poor distribution of tank effluent over the surface of a percolating filter, or poor quality tank effluent as a result of inferior septic tank design.

Percolating filters for small works are frequently rectangular and have fixed distributor channels dosed by a tipping tray mechanism. Small rotating sparges are also obtainable. Generally, humus tanks should not be provided because, unless frequently sludged, they do more harm than good. Humus can be separated by irrigation over land.

The method of sub-irrigation is the discharge of settled effluent into a system of shallow under-drains consisting of agricultural pipes surrounded with rubble and covered with earth. Because the degree of treatment is doubtful, this method is not favoured by many authorities. Nevertheless, it is very much used for the disposal of tank effluent and, provided that there are no dangers to water supplies, sub-irrigation is less liable to cause nuisance than broad irrigation or land filtration, because septic tank effluents, exposed to the air, can be very offensive.

No exact recommendation can be given as to the area of land required for sub-irrigation : sub-irrigation is considered to be a means of soaking away effluent without much regard to treatment and a system is considered satisfactory if the subsoil is such that it permits soakage to the desired extent. However, it follows that sub-irrigation is possible only where land filtration would be possible, and consequently it is reasonable to suggest that the area of land that would be required for treatment by land filtration might also serve for sub-irrigation.

SANITATION OF BUILDINGS.

The sanitation of buildings consists of the provision of waterclosets, slop sinks, urinals, sinks, baths, lavatory basins, etc., and the drainage of the wastes discharged therefrom. The sewage from waterclosets, slop sinks and urinals is considered, 'soil-sewage,' as is any mixture of this sewage with any other drainage. The drainage from sinks, baths, lavatory basins, etc., and any surface water mingled therewith is described as 'waste' or 'sullage': and the drainage arrangements for carrying it away can be different from those used for soil sewage. Water from roofs

and paved areas is described as 'surface-water' and the drainage arrangements therefor differ from those required for waste or sullage drainage and for soil drainage.

Where there is a combined sewerage system, soil, waste or sullage, and surface-water are combined together before they are discharged to the public sewer. Where there is a separate system, soil and waste are combined together before they are discharged to the soil sewers : surface-water is kept separate and discharged to the surface-water sewers.

Where surface-water is discharged to a waste or to a soil drain, a gully or disconnecting trap is provided to prevent air from the waste or soil drain escaping to free air via the surface-water connection.

Systems of Soil- and Waste-Pipes.

In the traditional or 'two-pipe' system of sanitation, soil and waste are discharged to separate stack-pipes, and where the waste is discharged to the soil drains a gully or disconnecting trap is provided to prevent foul air from the soil drain passing through the waste drain and stack-pipe.

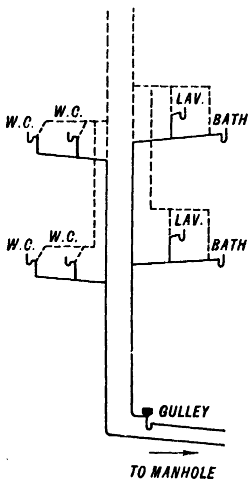


FIG. 8.—Two-pipe System of Sanitation.

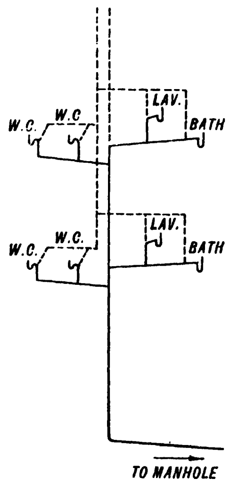


FIG. 9.—One-pipe System of Sanitation.

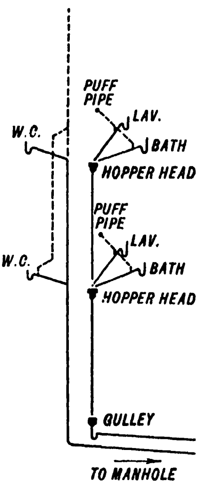


FIG. 10.—Old-fashioned Two-pipe System Involving Hopper-heads and Puff-pipes.

In the 'one-pipe' system, both soil and waste fitments are connected to the soil-stack, there being no separate stacks or drains for waste ; and, as an added protection against the escape of foul vapours into premises, the waste fitments are all provided with 'deep-seal' traps having a water-seal of not less than 3 ins. Both one-pipe and two-pipe systems, different as they are in principle, generally comply with *Ministry of Health Model Byelaws Series IV Buildings* and with London County Council By-laws (at the time of writing under revision).

There is an old form of two-pipe system largely used throughout the country, because it is inexpensive, but not now permitted in London County because it is considered inferior. In this system, the waste pipes from fitments on the ground floor discharge over a gully or a channel leading to a gully (this is permissible everywhere), and the waste pipes from fitments on upper floors discharge into open hopper-heads at the tops of stack-pipes which, in turn, discharge over gullies. (This is not permissible in London County.)

The design of sanitary systems for buildings is based on the need for interception of foul air in order that it shall not flow from the sewers or drains into buildings or elsewhere where a nuisance could be caused. This involves the provision of water-seals or traps, which in turn involve the provision of anti-siphonage pipes, and ventilation pipes (the main purpose of which is to prevent the siphonage of the traps, the second purpose of which is to ventilate the drains).

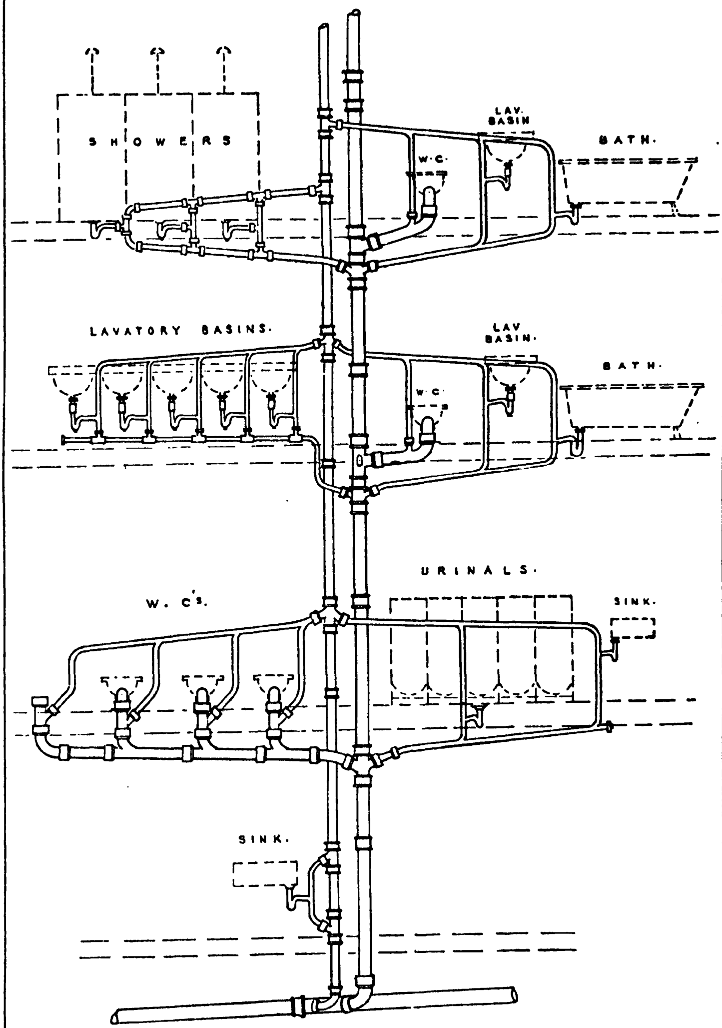


FIG. 11.—Typical example of One-pipe System of Sanitation.

The *Ministry of Health Model Byelaws Series IV Buildings*, which are the basis of local authority by-laws other than in London County, require that:

'every inlet to the drain, other than an inlet provided for the ventilation of the drain, shall be properly trapped.

'A waste pipe from a bath, sink (not being a slop sink), bidet or lavatory basin, and a pipe for carrying off dirty water, shall—

. . . .

'(2) if it discharges to a drain otherwise than by a soil pipe from a watercloset or a waste pipe from a slop sink, be disconnected from the drain by a trapped gully with a suitable grating above the level of the water in the trap;

'(3) if it is more than *six feet* in length, be provided with a suitable trap;

'(4) if it discharges into a soil pipe from a watercloset or a waste pipe from a slop sink, be provided whatever its length with a suitable trap adequately secured against destruction of the water seal.'

'The drains intended for conveying foul water from a building shall be provided with at least *one* ventilating pipe, situated as near as practicable to the building and as far as practicable from the point at which the drain empties into the sewer or other means of disposal:

'Provided that a soil pipe from a watercloset, or a waste pipe from a slop sink, constructed in accordance with these byelaws may serve for the ventilating pipe of the drain, if its situation is in accordance with this byelaw.'

'The soil pipe from a watercloset, and the waste pipe from a slop sink, other than parts of such pipes carried up as ventilating pipes, shall be—

. . . .

'(2) of an internal diameter not less than that of any pipe connecting it with the watercloset or slop sink, and in any case not less than *three inches*.'

'A ventilating pipe to a drain, and the part of a soil pipe from a watercloset or of a waste pipe from a slop sink which is carried up as a ventilating pipe, shall—

. . . .

'(2) be not less than *three inches* in internal diameter;

'(3) be carried upwards to such a height and in such a manner as effectually to prevent the escape of foul air from the drains into any building;

'(4) be covered at its open end, as a protection against obstruction, with a wire cage of copper or galvanized iron or other not less suitable cover admitting the free passage of air.'

'A ventilating pipe to a drain, a soil pipe from a watercloset, and a waste pipe from a slop sink, . . . shall not—

'(1) have a trap at its point of junction with the drain, or (except where necessary as part of the apparatus of any watercloset or slop sink) in any other part of the pipe;

'(2) have any bend or angle, except where unavoidable in which case the bend or angle shall be as obtuse as possible and shall not reduce the internal diameter of the pipe:

'where the watercloset discharges into a soil pipe which also receives the discharge from another watercloset, or from a bath, sink, urinal, bidet or lavatory basin, the trap of the watercloset shall be ventilated by a pipe which shall—

'(*a*) have an internal diameter of not less than *two inches*.

'(*b*) be connected with the arm of the soil pipe at a point not less than *three* and not more than *twelve inches* from the highest part of the trap, on that side of the water seal which is nearer to the soil pipe;

'(*c*) either have an open end as high as the top of the soil pipe or be carried into a soil pipe at a point not less than *three feet* above the highest connection to the soil pipe;'

'if the urinal can be entered from within the building, and is constructed to discharge into a soil pipe which also receives the discharge from another urinal, or from a watercloset, bath, sink, bidet or lavatory basin, the trap of the urinal shall be ventilated by a pipe which shall—

'(*a*) be of an internal diameter not less than that of the trap or *two inches*, whichever is less;

'(*b*) be connected with the waste pipe from the urinal at a point not less than *three* and not more than *twelve inches* from the highest part of the trap, on that side of the water seal which is nearer to the soil pipe; and

'(*c*) either have an open end as high as the top of the soil pipe or be carried into a soil pipe at a point not less than *three feet* above the highest connection to the soil pipe.'

Standards of Sanitary Equipment.

The minimum requirements of sanitary convenience in factories are laid down in *S.R. & O.*, 1938, *No.* 611. In brief, they are as follows :—

'There must be one watercloset for every twenty-five females and one for every twenty-five males, and in calculating the number, any odd number of persons less than twenty-five counts as twenty-five. In premises where the number of males exceeds one hundred, and sufficient urinal accommodation is also provided, there must be four waterclosets for the first hundred persons and one for every forty additional persons, and where the number of males exceeds five hundred, and the district inspector has certified that there is proper supervision and control in regard to the use of the conveniences, it is sufficient for one sanitary convenience to be provided for every sixty males in addition to sufficient urinal accommodation.'

The minimum standards for schools are given in *S.R. & O.*, 1945, *No.* 345.

For a high standard of provision in offices, etc., the figures given in Tables XII and XIII are recommended.

TABLE XII.—ACCOMMODATION FOR MEN.

No. of Staff.	W.C.'s.	Lavatory Basins.	Urinals.
1– 6	1	1	0
7– 15	1	1	1
16– 25	2	2	1
26– 35	2	2	2
36– 45	3	3	2
46– 65	3	3	3
66– 70	4	4	3
71–100	4	4	4
101–133	5	5	5
134–166	6	6	6
167–200	7	7	7

For every additional 40 persons, add 1 closet, basin and urinal.

TABLE XIII.—ACCOMMODATION FOR WOMEN.

No. of Staff.	W.C.'s.	Lavatory Basins.
1– 12	1	1
13– 15	2	1
16– 25	2	2
26– 35	3	2
36– 40	3	3
41– 57	4	3
58– 65	5	3
66– 77	5	4
78–100	6	4
101–120	7	5
121–133	8	5
134–140	8	6
141–160	9	6
161–166	10	6
167–180	10	7
181–200	11	8

For every additional 25 persons, add 1 closet; and for every additional 40 persons, add 1 lavatory basin.

Sizes of Drains.

Domestic soil flows, even from moderately extensive factories, are small generally, and in most circumstances drains of the minimum permissible diameter of 4 ins. are adequate. In very large

establishments, sizes of soil drains may be calculated in the same manner as those of soil sewers, or, in intermediate cases, they should be based on the maximum *simultaneous water demand* of the fitments served.

The peak rate of flow from an individual house can be taken as the discharge of one water-closet, approximately 3 cu. ft. per minute. The discharge from larger premises can be based on the simultaneous water demand, which is estimated as follows :—

The water demands of individual fittings are approximately :

½-in. bib-tap	0·4 cu. ft. per minute.
¾-in. bib-tap	0·9 „ „ „ „
1-in bib-tap	1·6 „ „ „ „
Watercloset W.W.P. tank	0·25 „ „ „ „
Shower roses	0·6 to 1·2 cu. ft. per minute.

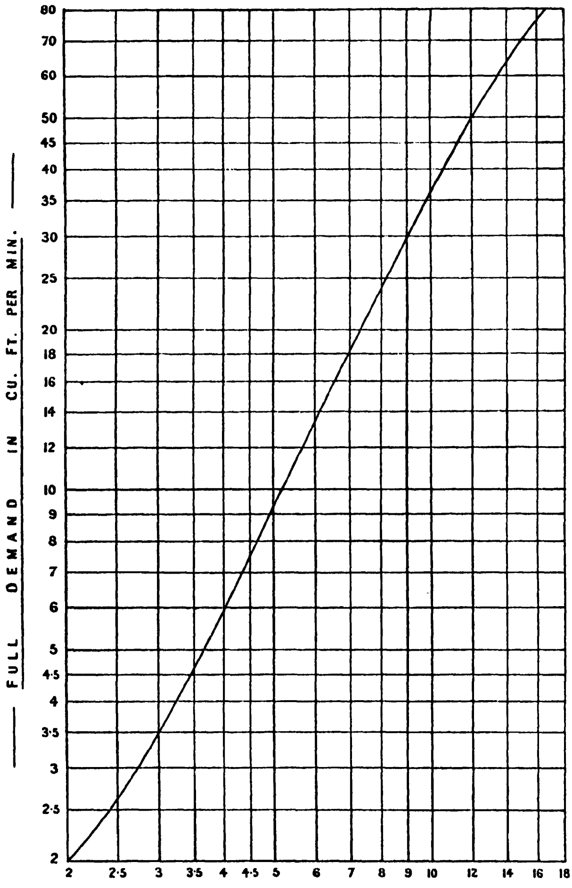


FIG. 12.—Curve of Simultaneous Water Demand of Sanitary Fittings.

The demands of all fittings except showers are totalled, but as all fittings are not likely to be used at one time, the simultaneous demand is found according to Fig. 12. To this is added the total for showers, because these are often used simultaneously. When flow exceeds those shown on the diagram, it should suffice to allow for four times average dry-weather flow.

In most instances individual premises are so small that the time of concentration of the surface water drains is negligible and therefore the Lloyd-Davies method cannot be applied. Theoretically, this would mean that a maximum rainfall of 3 ins. per hour should be allowed, but in practice it is found that, in small systems, surcharge and the storage of water in manholes makes provision for high rates of rainfall unnecessary. Consequently, flat rates have been applied, different authorities recommending very different rates. It appears that present day opinion is centering round an allowance of 1½ ins. of rainfall an hour as being applicable to all sites which are so small that the time of concentration of the drainage system is less than ten minutes.

The gradients of soil drains generally need to be somewhat steeper than those of sewers because they often receive little flow. Table XIV gives the recommended minimum gradients of 4-in. or 6-in. soil drains according to the estimated maximum rate of flow during the day, or flow based on the simultaneous water demand.

TABLE XIV.—MINIMUM GRADIENTS OF DRAINS OF 4-INS. OR 6-INS. DIAMETER.

Peak Daily Flow in Cu. Ft. per minute.	Gradient 1 in:
3	50
4	60
6	80
8	100
10	120

Well filled surface-water drains can be laid at the gradients recommended for sewers, with the addition of a minimum gradient for 4-in. drains of 1 in 90.

The above are recommendations based on both theory and practical experience. It should be stated that difficulties may be experienced with building inspectors, who still adhere to obsolete rules of thumb requiring much steeper gradients, although by-laws seldom include specific requirements as to gradients.

Materials for Sanitation of Buildings.

Waterclosets.—The pan (B.S. 1213), usually of fireclay, has an S-trap with outlet pointing vertically downwards for ground floor use, or a P-trap with inclined outlet for upstairs use. The standard pan in general use is known as the 'wash-down' type. A superior type, known as the 'siphonic' closet, has the advantage of effectively removing floating solids such as tend to remain in the wash-down closet. Earlier forms such as the wash-out closet, valve closet, etc., are obsolescent.

Watercloset suites are 'high level' or 'low level' according to the position of the flushing tank (water waste preventer). Flushing tanks generally fall under one or two heads, the 'bell-type siphon and the 'improved' siphon. The capacity of tank varies from 2 to 3½ gallons according to water undertakers' restrictions. Flushing valves as a substitute for tank-with-siphon are generally not approved in this country, but are used overseas. Special Eastern type closets for people who adopt the squatting position are manufactured for distribution to the Near East, India, etc.

Baths.—Baths (B.S. 1189) are usually cast iron porcelain enamelled, and in two patterns, 66 ins. and 72 ins. long, and are arranged to be with or without side panels. To satisfy the by-laws of many water undertakers, the overflow must discharge into free air and *not* be connected to bottom of trap on waste pipe.

Lavatory basins.—Lavatory basins (B.S. 1188) are usually of earthenware, fireclay, vitreous china, etc.: main sizes 22 ins. by 16 ins., and 25 ins. by 18 ins. They should be fixed with the top 2 ft. 7 ins. above floor level.

Sinks.—Sinks (B.S. 1206) are manufactured in fireclay to standard sizes, 30 ins. by 18 ins. by 10 ins., and 24 ins. by 18 ins. by 10 ins. without shelves: also 30 ins. by 21 ins. by 10 ins., and 24 ins. by 21 ins. by 10 ins. with back shelves.

Fireclay Washtubs and Tub-and-sink Sets.—These (B.S. 1229) consist of a combination of a washtub and a sink in one piece, a washtub and a sink in two pieces to be used together, or an individual washtub.

Metal sinks (B.S. 1244) are constructed of porcelain enamelled cast iron, porcelain enamelled pressed steel, stainless steel, or Monel metal. The units consist of a single sink with or without back ledge, combined sink and draining board with or without back ledge, and combined sink, draining board and work slab with or without back ledge.

Fittings.—Fittings used for plumbing consist of bib-taps with inlet from back, pillar-taps with inlet from below, stop valves, ball valves. The types of ball valves used include the Croydon type, the Portsmouth type (B.S. 1010) and the Equilibrium type.

Soil-and-vent-Pipes.—The majority of soil-and-vent pipes are constructed of cast-iron soil pipe and fittings (B.S. 416) jointed and caulked with lead. Lead pipe is also used for complete installations, for making-up pieces in cast iron installations, and for negotiating difficult bends. Copper pipe has been used recently in pre-fabricated work.

Waste-and-vent Pipes.—Waste-and-vent pipes outside buildings are usually of cast iron or lead; copper is rarely used. Inside buildings either lead or copper pipe is used.

Urinal wastes are preferably of lead and of no other material. Laboratory wastes should be of 'chemical lead.'

The following are suitable joints for good quality work in various materials:—

Lead to Lead.—For most purposes lead pipes are jointed together or junctions inserted by a plumber's wiped joint. When, however, lead piping is required for acids and 'chemical lead' is used, and in other circumstances where a soldered joint is undesirable, a 'lead burned' joint is used. 'Lead burning' is welding of lead work by oxy-acetylene using a pure lead filler rod.

Lead to Cast Iron.—The lead pipe is wiped to a brass caulking sleeve which is inserted in the socket of the cast iron and secured by a caulked lead joint. Where chemical lead is used the lead pipe is carried *through* a caulking sleeve of adequate size and burned to it.

Copper to Copper.—Heavy copper tubes of the kind that are used underground are frequently jointed by screwed joints in the same manner as screwed iron barrel. Light weight copper tube is too thin to take a standard screw thread and for this reason a number of other methods of jointing are employed.

Perhaps the most commonly used in the compression joint. There are many proprietary compression joints on the market, some of which necessitate deforming the tube at the point of junction, but the majority of which secure a watertight joint by squeezing a soft copper ring between the outside of the pipe and the inside of the brass coupling unit. The neatest appearance is obtained by the use of a capillary type fitting. This is a sleeve of brass which closely fits over the copper tube and is secured thereto by solder which is drawn through the annular space by capillary attraction.

Weldable fittings are somewhat similar to capillary fittings in appearance, but are secured in position by bronze welding.

Copper may be welded pipe-to-pipe, without any fittings, with bronze welding material or brazed with silver solder or other hard solder. This last method is at present not much in use for site work, although it is largely used in the assembly of pre-fabricated work such as the sparge of pipes of urinals. It is, however, by no means unsuitable for site erection of pipe-work.

Copper to Lead.—Lead pipe is jointed to copper pipe by plumber's wiped joint.

Copper to Cast Iron.—What is often considered the best arrangement for jointing copper to cast iron is the use of a caulking sleeve having a coupling for copper and a spigot which is inserted into the socket of the cast iron pipe and jointed with a caulked lead joint.

Copper to Stoneware.—A caulking sleeve having a coupling for copper may be used and be jointed to the stoneware pipe with cement mortar. Alternatively, the end of the copper pipe may be flanged out, inserted in the stoneware socket and jointed with cement mortar, in the same manner as stoneware pipes.

Drains.—The majority of drains to premises are of stoneware or fireclay pipe as used for sewers protected with concrete as required by local by-laws. Drains under buildings are preferably of cast iron (B.S. 437): it will be noted that a different specification of cast iron pipe is used for drainage than is used for sewerage.

Fittings.—Standard stoneware fittings are specified in B.S. 539, and cast iron fittings included in B.S. Schedule 1130. These include gullies, access shoes, disconnecting traps, rodding eyes and cast iron bolted accesses.

See also Descriptive Section XXXVII
Pulsometer Engineering Co. Ltd.

SECTION XXXVIII

PART I

CRANES AND CRANE APPLIANCES (pp. 801–815)
(Revised by John H. Huntley, M.I.E.S.)

PART II

LIFTS AND ESCALATORS (pp. 817–830)
(Contributed by Col. E. B. Rook, T.D.)

PART III

ROPEWAYS AND CONVEYORS (pp. 831–844)

SECTION XXXVIII

PART I

CRANES AND CRANE APPLIANCES

CRANES.

(Revised by John H. Huntley, M.I.E.S.)

British Standard Specifications for cranes are now issued as follows :—

Derrick Cranes	No. 327—1934 (under review).
Locomotive Cranes	No. 357—1930.
Electric Overhead Travelling Cranes .	No. 466—1947.

A specification for dockside cranes is in course of preparation and a specification for overhead heavy duty overhead travelling cranes used in iron and steel works will shortly be issued by B.I.S.R.A.

General Notes on Cranes.

The selection of a crane for any particular service is governed by several factors, such as load, space available, power at disposal of user, frequency and speed of operation, height of lift, duration of service and cost.

In power stations, stores, and places where an occasional lift is all that is required, a hand crane is quite suitable, even for loads up to 20 or 30 tons, but for frequent operation, or where the load has to be moved a considerable distance, it is advisable to instal a power-driven crane, even for a load of a few hundredweights, as the saving in time and labour soon pays for the difference in initial cost.

The electric crane has the following points in its favour :—

1. Simplicity of operation.
2. No power consumed whilst crane is standing.
3. Power used varies in proportion to load.
4. Always ready for starting up.

The steam crane has the advantage that it can be used wherever a supply of fuel and water is available, but time is lost in raising steam, and fuel is consumed in keeping up steam between the lifts.

Hydraulic cranes are only economical in operation when constantly working under full load, as the water consumption is the same for light or heavy loads, although by means of double-power rams attempts have been made to overcome this defect. Trouble is caused by frost in exposed situations. Hydraulic cranes are still largely used in shipyards and boiler shops, for handling plates, portable riveters, etc., where a small lift at slow speed is required. The working hydraulic pressure is usually 1,500 lbs. per sq. in., but may be as high as 2 tons per sq. in.

Pneumatic power is applied only to small hoists and pulley blocks, the air pressure being usually about 100 lbs. per sq. in.

The use of steam as the motive power on portable jib cranes has been, to a large extent, superseded by the high compression oil engine, which is either coupled direct to the crane motions by clutches or the engine is used to drive an electrical generator to supply current to motors geared to the different motions. This latter arrangement is preferable as it eliminates reversing clutches and si mplifies the duty of the operator.

The objection to the internal combustion engine for cranes is its non-reversibility necessitating the use of reversing clutches for each motion.

HAND OVERHEAD TRAVELLING CRANES.

For hand cranes up to 40 ft. span and loads up to 5 tons, a single girder crane can be adopted, but for longer spans and heavier loads it is advisable to use the double girder type to obtain lateral stiffness.

Light cranes are usually operated from the floor level by hanging chains, but for cranes above 10 tons it is advisable to work them from the crane platform. The pull on a hand chain should not exceed 40 lbs. per man, and for crank handles not more than 25 to 30 lbs. per man. The handles should have a radius of 16 ins. to 18 ins. and the centre be at a height of from 2 ft. 6 ins. to 3 ft. above platform level.

Travelling wheels for both crab and crane are usually of cast iron, and should be fitted with ball or roller bearings.

The hoisting gear should be efficient, preferably with machine-cut teeth, pinions of mild steel, wheels of cast iron, and should be fitted with a self-sustaining brake which requires to be released by the operator before the load can be lowered, but which is inoperative in the hoisting direction. This condition can be obtained by an internal ratchet or a screw disc brake.

Worm gear should have a coarse pitch and the sustaining power of the crane should not be dependent upon the inefficiency of the gear.

ELECTRIC OVERHEAD TRAVELLING CRANES.

For engineering shops, foundries, steelworks, etc., the overhead electric travelling crane is unequalled for general service.

The motions of hoisting, traversing and travelling are generally arranged, and a separate motor should be fitted for each motion. An auxiliary hoist having a lifting capacity of about one quarter the main hoist is desirable on cranes above 40 tons capacity, and should be operated by an independent motor.

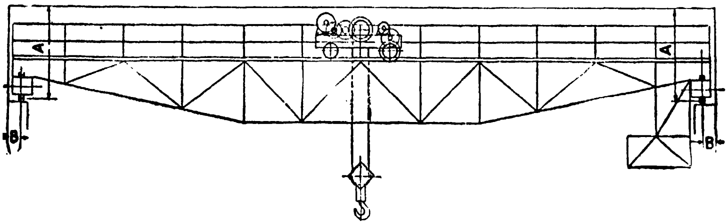


FIG. 1.

The following are particulars of standard 3-motor overhead travelling cranes (fig. 1), 30 ft. lift :—

TABLE OF STANDARD 3-MOTOR OVERHEAD ELECTRIC TRAVELLING CRANES.

Size.	Span.	Head-room. A.		End Space. B.	Wheel Base.		Max. Press. per wheel.	Wt. of Crane.	Speeds in feet per minute.			Falls of Rope.
Tons.	Ft.	Ft.	Ins.	Ins.	Ft.	Ins.	Tons.	Tons.	Hoist.	Traverse.	Travel.	
2	30	5	0	7	6	0	2·7	5·8	40	100	300	2
2	40	5	6	7	8	6	3·0	7·3	40	100	300	2
2	50	5	6	7	8	6	3·3	8·5	40	100	300	2
5	40	6	0	8	8	6	5·3	9·8	30	100	300	4
5	50	6	0	8	8	6	5·8	11·5	30	100	300	4
5	60	6	0	8	9	6	6·1	13·0	30	100	300	4
10	40	6	6	8½	10	0	8·4	12·0	20	80	300	4
10	50	6	9	8½	10	0	9·1	14·5	20	80	300	4
10	60	6	9	8½	10	0	9·7	16·5	20	80	300	4

TABLE OF STANDARD 3-MOTOR OVERHEAD ELECTRIC TRAVELLING CRANES—*contd.*

Size.	Span.	Head-room. A.		End Space. B.	Wheel Base.		Max. Press. per wheel.	Wt. of Crane.	Speeds in feet per minute.			Falls of Rope.
Tons.	Ft.	Ft.	Ins.	Ins.	Ft.	Ins.	Tons.	Tons.	Hoist.	Traverse.	Travel.	
15	40	7	0	9	10	0	11·3	13·8	20	80	275	6
15	50	7	0	9	10	0	12·0	16·2	20	80	275	6
15	60	7	0	9	10	0	12·5	18·0	20	80	275	6
20	40	7	3	9½	10	6	14·5	16·5	15	80	275	8
20	50	7	3	9½	10	6	15·2	19·0	15	80	275	8
20	60	7	3	9½	10	6	15·8	21·0	15	80	275	8
25	40	7	6	10	11	6	17·5	19·0	15	80	250	8
25	50	7	6	10	11	6	18·5	22·0	15	80	250	8
25	60	7	9	10	11	6	19·2	24·2	15	80	250	8
30	40	8	0	10½	12	0	20·3	22·0	15	75	250	8
30	50	8	0	10½	12	0	21·5	25·0	15	75	250	8
30	60	8	3	10½	12	0	22·5	28·0	15	75	250	8
40	40	9	0	11	13	0	26·5	26·0	12	70	250	8
40	50	9	0	11	13	0	27·8	30·0	12	70	250	8
40	60	9	3	11	13	0	29·5	34·5	12	70	250	8
50	40	10	0	11	13	0	32·0	31·0	10	70	230	8
50	50	10	0	11	13	0	34·0	35·0	10	70	230	8
50	60	10	3	11	13	0	35·5	39·5	10	70	230	8
60	40	10	3	12	13	6	38·5	35·0	10	70	250	8
60	50	10	3	12	13	6	40·5	40·0	10	70	230	8
60	60	10	6	12	13	6	42·5	45·0	10	70	230	8

The above table is for cranes on four travelling wheels.

For 70-ton cranes and upwards the usual practice is to mount the crane on eight travelling wheels, and the wheel loads in the following table are based on this. The column headed 'Centres of Wheel Bogies' indicates the distance between the centre of each pair of wheels in each end carriage, and the following column gives the distance between the two wheels of each pair:—

Size.	Span.	Head-room A.		End Space B.		Crs. of Wheel Bogies.		Bogie Wheel Centres.		Max. Press. per Wheel.	Wt. of Crane.	Speeds in F.P.M.			Falls of Rope.
												Hoist	Trav-erse.	Travel	
Tons.	Ft.	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Tons.	Tons.	Ft.	Ft.	Ft.	
70	50	10	9	1	1	10	6	5	0	22·82	46	10	60	200	10
70	60	10	9	1	1	10	6	5	0	23·90	50½	10	60	180	10
70	70	11	0	1	1	10	6	5	0	24·87	56	10	60	180	10
80	50	11	0	1	2	11	0	5	0	25·40	51	8	60	180	12
80	60	11	0	1	2	11	0	5	0	26·75	56	8	60	150	12
80	70	11	3	1	2	11	0	5	0	28·00	62	8	60	150	12
100	50	11	3	1	3	11	6	5	3	32·12	62	8	60	150	12
100	60	11	3	1	3	11	6	5	3	32·62	68	8	60	150	12
100	70	11	6	1	3	11	6	5	3	35·00	75	8	60	150	12

NOTE.—The headroom figures allow for about 3 ins. clearance and the end space for about 1½ in. clearance.

The speeds of working given above are for general engineering shops: for steel works the hoisting speed should be doubled and the other motions increased by 50 per cent.

For short-span cranes of small size, a pair of 'H' beams make an economical set of girders, but the span should not exceed sixty times the width of the flange of one girder. For cranes up to 60 ft. span the most economical construction is the trapezoidal single web girder, with lattice auxiliary connected to the main girder by horizontal and diagonal bracing. This makes a very stiff construction, is easy to manufacture, and all surfaces are accessible for painting. The platform and travelling gear can be well supported by bearers between the main and auxiliary girders. For long spans and for outside service it is desirable to make the girder of the lattice

type to reduce weight and wind surface. The depth of the girder is usually made one-twelfth the span. The maximum bending moment due to the crab, assuming each wheel equally loaded, is given by the formula :—

$$\frac{W}{2S}\left(S-\frac{B}{2}\right)^2 \text{ ft.-tons,}$$

where,

W = load per wheel in tons ; S = span in feet ; B = wheel base of crab in feet.

To this must be added the bending moment due to the deadweight of the girder and travelling gear. The modulus of the girder section should be such as to give a stress not exceeding 6 tons per sq. in. in tension. The stresses in lattice girders are best determined by graphic methods.

See B.S.S. No. 466 for rules on working stresses.

The end carriages can be constructed of channels, and built up of plates and angles, when larger sections are required. They should be connected to the girders by large gusset plates, and fitted bolts to prevent cross winding. The wheel base should preferably be one-fifth to one-sixth of the span.

Rail wheels should be of cast steel, or of cast iron centres with steel tyres shrunk on the outside. The width of the rolling surface of the rail is an important point, and for heavy loads a rounded surface should be avoided. The safe load on a wheel may be arrived at by the following empirical formula :—

For cast steel wheels, safe load in pounds = 800 × W × D,
" steel tyred " " " " = 1,200 × W × D,

where,

W = width of rail face in inches ; D = dia. of wheel in inches.

These figures are suitable for cranes for machine shops, in power stations. For heavy duty cranes in constant use the factors should be reduced to 600 for cast steel wheels and 750 for tyred wheels. The steel castings should in all cases be of hard wearing quality to B.S.S. 24.

To calculate the power required to travel a crane a tractive effort of 60 to 70 lb. per ton is a safe average figure, and if the crane is working in the open, the motor should be powerful enough to move it against a wind pressure of 5 lb. per sq. ft. applied to the exposed area if the girders are of the plate type or if the girders are of the lattice type 1½ times the exposed surface of a single girder may be taken as the area subject to the pressure of the wind. The application of roller bearings to the crane axles will decrease the tractive effort by approximately 50 per cent.

To prevent cross winding the cross shaft should be stiff enough to avoid undue torsional deflection when the driving wheels are unequally loaded due to the crab being at one end of the span. A foot brake should be fitted to the cross shaft on cage control cranes and a solenoid brake fitted to the motor shaft on all cranes working in the open.

The crab frame should be built up of steel sections well stiffened to form a rigid support for the motors and gearing. Seatings should be provided for all bearing brackets consisting of steel strips riveted or fusion welded to the frame and machined in position.

The traversing wheels should be of cast steel and on the larger sizes of cranes have rolled steel tyres shrunk on to cast iron or cast steel centres.

It is not necessary to fit a brake on the traversing motion unless the speed exceeds 100 ft. per minute or the axles are fitted with ball or roller bearings.

The load is preferably lifted on wire ropes winding on to a cast iron drum rotated by gearing, the number of falls of rope depending upon the load. For cranes up to 3 tons capacity, the load may be lifted on two falls winding one fall on to the barrel. Where four or more falls of rope are used, two falls should wind on to the barrel, which should have right and left-hand spiral grooves and thus keep the load central between the girders.

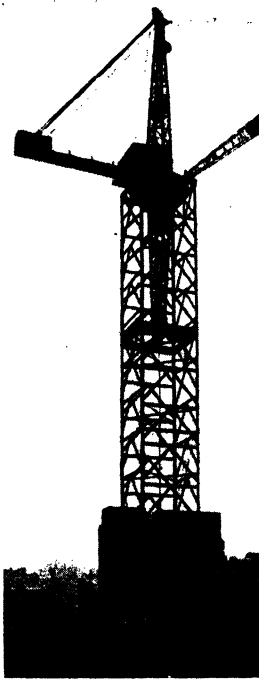
The efficiency of the gearing varies from about 75 per cent. on small cranes to 65 per cent. on large cranes. A convenient rule to determine the horse-power of the hoisting motor is

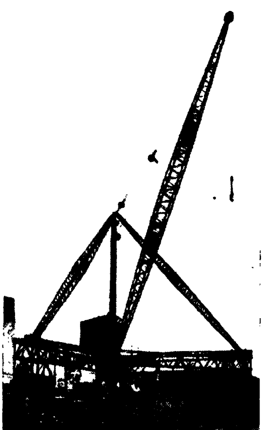
$$\text{H.P.} = \frac{\text{foot-tons per minute}}{10}$$

on a crane with gun-metal bearings. For a crane with ball or roller bearings and gears working in oil baths, foot-tons per minute divided by 12 will provide the motor of ample power.

The speeds of working given in the table, pp. 802, 803, are suitable for general engineering shop work, but for steelworks very much higher speeds are used, and as continuous night and day service is the general practice in these works the design of the crane should be specially considered and made very robust.

The hoisting motion is preferably fitted with two brakes, one of which should be an automatic solenoid brake ; the type of the other brake will depend entirely on the service for which the crane is used.





The Overhead Traveller lends itself to ready adaption for specific duties such as handling molten metal in ladles, serving soaking pits, stripping ingots from moulds, handling rods, pipes and long sections with magnets, charging furnaces, working with grabs, handling heavy forgings under the press and so on.

Ladle cranes have been made up to 250 tons capacity, but the general practice is to instal cranes from 100 to 125 tons capacity. These cranes are provided with two crabs and the larger sizes with four cross girders, the main crab then travelling on the two outer girders and the auxiliary crab on the two inner girders and underneath the main crab. The main crab is fitted with two barrels and the lifting ropes pass between the main and auxiliary crab girders.

In the smaller sizes of ladle cranes, there are usually only two girders, the main crab travelling on the top and the auxiliary crab on a track carried from the lower flange. The main hoist ropes then pass outside the girders.

The auxiliary crab has usually a lifting capacity of about 20 per cent of the main crab.

For handling forgings under the hydraulic press, the crane should be fitted with relieving gear to prevent overload due to the action of the press in pulling down a forging which is suspended not quite level with the anvil. This relieving gear may consist of a spring suspension incorporated in the snatchblock, but it is better to arrange for automatic easing off the brake as the overload is applied.

Cranes of this type have been made up to 300 tons capacity and they are usually fitted with detachable gear for rotating the ingots.

A useful attachment to an Overhead Crane is an underhung jib as this increases the area served by the crane and allows of it handling goods from one bay to another.

For outside service, the Goliath Crane gives the same service as an Overhead Traveller without the cost of erecting a gantry and the cross girders may be cantilevered out at each end so that the crane will serve an area beyond that enclosed by the track rails.

Derrick Cranes.

The Scotch derrick cranes extensively used by civil engineers and building contractors should be made to comply with B.S.S. No. 327 (at present under revision), which has been drawn up with a view to avoiding the numerous accidents which have occurred with this type of crane. Many cranes manufactured before this specification was issued had a rather low factor of safety and great care should be taken when using these cranes at anywhere near their normal load capacity.

All derrick cranes should be fitted with an indicator to give visible and audible warning to **the driver when the full load is exceeded.**

Purchasers and users of cranes should consult Statutory Instruments, 1948, No. 1145, 'The Building (Safety, Health and Welfare) Regulations, 1948.'

Locomotive Cranes.

For general service on railway sidings around works and shipyards the locomotive steam or Diesel crane is unequalled. These cranes are made in standard sizes up to 10 tons capacity and lift the load free on track with the jib in any position.

The following table gives particulars of standard steam cranes :—

Standard Locomotive Steam Cranes, 4 ft. 8½ in. Gauge.

Size.	Radius.	Size of Engines. (Cylinders.)	Size of Boiler.	Steam Press.	Speeds.			Weight.
					Hoist,	Slew,	Travel,	
Tons.	Ft.	Ins. Ins.	Ft. Ins. Ft. Ins.	Lbs.	f.p.m.	r.p.m.	f.p.m.	Tons.
3	16	7 by 10	7 9 by 3 6	100	80	2½	350	14½
5	16	8 by 10	8 3 by 4 0	100	80	2½	350	22½
7	16	8 by 12	8 3 by 4 0	100	50	2	300	32½
10	16	9 by 12	9 0 by 4 0	100	40	2	250	45
15	16	9 by 12	9 0 by 4 0	100	40	1½	200	54

For cranes of longer radius or greater lifting capacity than the above, blocking girders and rail clips are fitted to give the necessary stability, or the cranes may be made heavier.

Cranes that are to travel round sharp curves should have sliding axle gears or differentia gears and the conditions pointed out to the crane makers at the time of ordering.

The gearing **should be of steel throughout.**

The slewing spur-ring forms the roller path and should rest on a machined seating on the truck, the pressure of the slewing rollers being sufficient to hold it in position for normal working, but in the event of sudden loads it will slide sufficiently to absorb the shock and prevent damage to the gearing.

The engines are usually of the double cylinder non-condensing type, cranks at right angles and cut off at about three-quarter stroke. The boiler is of the vertical cross-tube type for a working pressure of 80 to 100 lbs. per sq. in. and should be lagged. Two means of feed should be fitted, and the incoming water should enter above the firebox and not impinge directly on the heated plates.

The mountings should preferably be flanged, and secured by studs to pads riveted to the shell of the boiler, rather than screwed directly to the shell. The safety valve should be separate and not a combined mounting with the stop valve.

A list of standard boiler mountings would consist of

1 steam-pressure gauge with syphon;
1 water gauge with plate glass protection;
1 spring loaded safety valve;
1 steam stop valve;
1 injector with steam cock and feed check valve
2 test cocks;
1 steam jet cock and pipe to funnel;
1 blow-off cock.

The feed pump may be either a steam donkey pump or be driven direct from the engine crosshead.

B.S.S. No. 356, 1930, should be followed as a guide for the construction of these cranes.

Steam Breakdown Cranes

for use on the permanent way, have been made up to 100 tons lifting capacity, but in Great Britain the usual size is from 10 to 35 tons. The truck of these cranes is provided with the usual permanent-way fittings and the axles are spring borne. The maximum load per axle with the crane in running order should not exceed 18 tons. When the crane is 'in train' the travelling gear should be disconnected and the superstructure locked to prevent it turning. Owing to the limited space due to the crane having to clear the loading gauge, a comparatively small size boiler must be fitted, and this should be of the multitubular type as the ordinary cross-tube boiler would not generate sufficient steam.

Mobile Cranes.

Mobile cranes fitted with either solid rubber or pneumatic-tyred wheels or endless chain tracks have proved of great value on the dockside, in warehouses and for general contracting work where there are no rail tracks or gantries. The crane is fitted with internal combustion engine as a source of power, which makes it independent of external sources of supply and the engine may be either coupled direct to the different motions or used to drive a generator on the crane supplying power to electrical motors for driving the different motions. The capacity for this type of crane is limited as it depends upon its dead-weight for stability. A typical crane can lift 2 tons at 18 ft. outreach and 5 tons at 10 ft., but they have been made to lift up to 15 tons.

Wharf Cranes.

For dockside service in loading and unloading ships, the high gantry type of jib crane is principally used. The usual sizes are from 1½ tons to 5 tons and the radius from 40 to 60 ft. The gantry is made wide and high enough to admit of the passage of rolling stock underneath and to keep the jib clear of the sides of large vessels. A separate motor should be fitted for each motion.

Typical speeds of working, in feet per minute, are as follows:—

Description.	Size. 1½ tons.	3 tons.	5 tons.
Hoisting	300	200	120
Slewing	500	500	400
Travelling	100	100	80
Luffing	150	150	100

For high luffing speeds some form of compensating gear is essential to reduce the power consumption due to raising the load and jib each time the radius is reduced. With the ideal compensating gear the path of the load is kept horizontal as the jib is raised or lowered, and with the jib balanced the power required is only that necessary to overcome friction.

Numerous types of compensating gear have been invented during the past few years, some of which are very complicated, and the crane user should select those of the simplest construction with the minimum number of moving parts, and pay special attention to the lead of the hoisting rope, which often suffers considerably through being passed round numerous pulleys in order to give the desired level path to the hook as the jib is raised or lowered.

The luffing motion is frequently operated by a crank, as this eliminates the use of ropes, and the throw of the crank is proportioned so that the jib cannot exceed the prescribed maximum and minimum radii.

For rapid handling of coal, ore, or similar loose materials, the transporter grab crane is unequalled. The load is carried by a trolley which travels on a horizontal track, supported in a suitable manner by the crane structure, which latter may take the form of a tower with cantilevers on each side or a bridge, if it is desired, to convey the load over a considerable distance.

If working at the dockside the outer cantilever may be made to hinge up to clear the masts and funnels of ships when coming alongside.

The operating machinery can be fixed on the structure and the trolley carry only guide sheaves for the hoisting ropes and be hauled along by other ropes winding on to a separate winch, or the hoisting and transporting gear can be carried by the trolley. The latter system is usually adopted when the gross load exceeds 5 tons, as with the fixed machinery system the wear of the ropes becomes excessive in the larger sizes and off-sets the initial saving due to the lighter design of structure.

The advantage of this type of crane over the jib crane is that the load is conveyed in a direct line from point to point instead of through the arc of a circle. The transporting speed can be at the rate of 800 to 1,000 ft. per minute, whilst the maximum safe slewing speed is not more than 400 to 500 ft. per minute. With a 3-ton grab the transporter crane will usually handle 150 tons per hour against 100 to 120 tons on a jib crane.

On the larger types the driver travels with the trolley and is always immediately above the area of operations. In some cases the trolley is fitted with a revolving jib of about 10 ft. radius, from the point of which the grab is suspended. This enables the driver to cover the whole area of the ship's hatchway without moving the main structure, resulting in a speeding up of operations.

For handling coal, ore, and similar materials, much time and labour is saved by using grabs; these may be on the single or double-chain principle.

The single-chain grab may be fitted to any crane lifting on a single fall of rope, but grabs of the double-chain type require an extra drum to coil up the opening or closing rope. This should preferably be driven through a friction drive, although for short lifts a spring drum is quite suitable.

The best practice is to use a grab of the four-rope construction, *i.e.* having two hoisting ropes and two closing ropes, with sheaves of large diameter and properly spaced, the tendency to spin is eliminated, and the grab can be opened or closed in any position. This necessitates two power driven rope drums on the hoisting winch, and within the limit of about 10 tons, a single motor drive can be used with a balancing gear between the drums to ensure that the speeds synchronise and the distribution of the load on each drum is equal. Above this load, the best practice is to have a separate motor and gear for each drum and instal an electrical system for synchronisation and equalising of the load.

Grabbing work is very severe on cranes, and the nominal size of the crane should be 50 per cent. in excess of the combined weight of grab and contents, or in other words, the factor of safety should be 10 instead of 6 to 7. The empty grab may be balanced by sliding counter weights to reduce the power consumption on jib cranes.

Automatic self-dumping single-rope grabs, which open when the weight of the grab is relieved from the hoisting rope, require only a single-drum winch for operation and give a higher duty than the single-rope grab with fixed discharge gear.

On heavy grabs it is desirable to duplicate the rope, *i.e.*, have two hoisting ropes and two holding ropes. The rope sheaves should be at least 24 times the rope diameter and well spread to prevent twisting of the ropes round each other.

CAPACITY AND WEIGHTS OF GRABS.

For Coal.			For Iron Ore.		
Capacity.	Weight of Grab.	Approx. Weight of Coal.	Capacity.	Weight of Grab.	Approx. Weight of Ore.
Cub. ft.	Cwts.	Cwts.	Cub. ft.	Cwts.	Cwts.
45	30	15	21	38	20
56	36	20	31	52	30
63	42	25	42	68	40
82	48	30	65	95	60
112	68	40	150	220	200
180	112	70	—	—	—

Shipbuilding Cranes.

For service over shipbuilding berths, high gantries with overhead travellers and mono-rail cranes, gantries with cantilever cranes, steel derricks and tower cranes, are used, and experience favours the latter type.

Tower cranes may be either of the fixed or travelling type, practice in Great Britain being inclined to the fixed type, as they require less clearance between the berths, and use is made of valuable space which would otherwise have to be kept clear for the track of a travelling crane.

The crane structure consists of a horizontal cantilever supporting the track for the load trolley, and is carried on the top of a braced steel mast which is free to revolve inside a fixed or travelling tower. The cantilever is balanced by a tail carrying the machinery, which gives the crane a hammer-headed appearance.

Due to the development of pre-fabrication in shipbuilding, the capacity of these cranes has increased in recent years and the latest installation includes cranes to lift 40 tons at 70 ft. radius and 20 tons at 140 ft. radius.

These cranes, if desired to lift varying loads at different radii, should be fitted with an automatic anti-overload device which will prevent a load above the safe limit for its radius being lifted, and will also prevent the trolley traversing beyond the safe radius for that particular load.

For fitting out, cranes of the wharf type with derricking jibs are used, and range in size from 10 to 60 tons, with radii up to 80 ft., according to requirements. Two speeds of hoisting are generally arranged, so that light loads may be lifted at a speed of 50 to 80 ft. per minute. A typical crane of this type lifts 30 tons at 55 ft., 20 tons at 72 ft., 10 tons at 80 ft. radius, and is mounted on a gantry to clear rolling stock, the centres of the gantry track being 22 ft.

Cranes of this type are installed for dry dock service, and it is occasionally necessary that they should work from both sides of the dock. This may be arranged by connecting the two sides of the dock by a track at the end and introducing turntables, or better by making the track sweep round on a curve and carrying the rail wheels on special swivelling bogies and inserting an epicyclic gear between the sections of the travelling motion cross shaft which connects the driving rail wheels on each side of the crane.

Floating cranes are occasionally used for fitting out and repair service, and have the advantage of being transferable from dock to dock, but with them it is not easy to deposit a heavy load with precision, as the crane follows the movement of the water and its equilibrium is disturbed with every fluctuation of loading.

These cranes are usually steam operated but several large cranes have been electrically driven and this practice will be extended on account of the increasing use of the diesel engine which can be used to drive an electric generator to supply current to the crane motors as well as to drive the propelling machinery.

The pontoons are usually rectangular, flat-bottomed vessels, but if self-propelled it is advisable to give them a ship shape. The angle of heel under full load conditions and a 5 lb. wind should be not more than 5°. Travelling ballast is sometimes fitted to limit the heel under various loading conditions.

Capstans.

For hauling trucks in goods stations or on the dock side a capstan designed to exert a pull of 1 ton at a speed of 150 to 200 ft. per minute is sufficient. The free bollard type with the rope attached to the bollard is more economical in use than the fixed bollard type which depends upon the friction of the coils of rope round the bollard for hauling power.

The warping of ships into position in the docks is done by capstans of from 10 to 20 tons pull A constant torque at varying speeds is the desideratum for this duty, and the characteristics of the Austin constant-current motor make it eminently suitable for this work.

Overhead Runways.

Overhead runway systems, consisting of a track formed from a single rolled-steel joist which may be curved to suit local conditions, and have switches or turntables, are frequently installed in warehouses, steelworks, gas works, and other situations where an overhead crane is unsuitable or too expensive. Hand-pulley blocks and trolleys are used on the runways for occasional service and light loads, but for speed and loads above two tons an electric trolley hoist should be installed. These may be either floor or cage controlled, and have usually separate motors for hoisting and travelling. The lifting speed may be as much as 100 ft. per minute, and the travelling speed from 200 to 800 ft. per minute, according to the duty required.

Grabs can be fitted to lift coal, ore, or similar material.

List of British Standard Specifications for use in the manufacture of cranes :—

Drums and iron castings	No. 391
Steel castings and forgings	„ 24 (Part 4)
Tyres	„ 24 („ 2)
Hooks	„ 482
Structural steel	„ 15
High tensile steel	„ 548
High tensile steel for welding	„ 968
Spur gearing	„ 435
Worm gearing	„ 721
Ropes	„ 302
Shafts	„ 24
Specification 8, Class 'B,' 'C' or 'D.'	
Rivets	No. 15, Class 'A'
Keys	„ 46
Motors	„ 168
Controllers and resistances	„ 587

Reference should also be made to B.S.S. Handbook No. 4, 'Lifting Tackle,' and to the Electricity Factories Act, 1911. B.S.S. 449 has recently been issued dealing with the maximum stresses on structures, but the figures given therein should only be applied to cranes with a factor depending upon the duty for which the crane is intended.

STRUCTURES.

Crane structures should be made of open-hearth mild steel, 28 to 32 tons tensile strength, extension not less than 20 per cent. in eight inches. In determining the section of members, allowance should be made for the vibration due to the movement of the load and machinery, for the movement of the structure in the case of a travelling crane, and deflection should be considered where bearings for shafts are carried on structural members. The stresses should be such as to give a factor of 5 to 6 on the ultimate strength of the material. The ratio of length to radius of gyration should carefully be considered in struts, and charts plotted from Perry Robertson form a reliable basis on which to determine the stresses. For wind bracings the factor of safety need not be more than three, as this is an exceptional loading infrequently applied.

High tensile steel 37/44 tons has been used where weight is a consideration and is economical on long jibs and cantilevers where the dead weight imposes a considerable proportion of the load on the supporting structure. To obtain full advantage from the use of high tensile steel, the factor of safety should be based on the yield point which is higher in proportion to the breaking stress than ordinary mild steel.

Wind pressure on cranes in exposed positions in Great Britain may be taken as a maximum of 40 lbs. per sq. ft. It is inadvisable to work a crane in a wind pressure above 5 lbs. per sq. ft., and when designing the structure it is therefore unnecessary to allow for a higher pressure under full load conditions. The maximum wind pressure must of course be considered separately.

The revolving superstructure of jib or cantilever cranes may be carried on a ring of live rollers, or on a set of four or eight large diameter rollers designed as travelling wheels, but with conical treads and unflanged. The live rollers may be of either forged or cast steel and should run between two steel guard rings connected to the centre pin casting by radial rods. A gunmetal washer should be fitted between each roller and the outer guard ring, to take the outward thrust of the rollers. For unhardened rollers the pressure in lbs. should not exceed 400 by length in inches by diameter in inches. The roller-paths should be of steel, machined conical on the face to form a full bearing for the rollers.

On jib cranes up to about 5 tons capacity the roller-path frequently consists of a flat-bottom rail bent to a circle, and the superstructure is then mounted on four rollers 15 ins. to 18 ins. diameter. This is a cheap and serviceable construction.

A friction clutch should be incorporated in the slewing gear adjusted to slip and prevent damage in the event of excessive shock due to too sudden application of the brake or the fouling of jib with some stationary object.

GEARING.

On all high-class crane work the gearing should be machine cut. For hand and low-power cranes cast-iron wheels and steel pinions are satisfactory, but for all high-speed cranes the gearing should be of steel, with the possible exception of motor pinions, which, to ensure silent running, may be of rawhide, fabroil, or phosphor bronze.

For heavy loads and high speeds, machine-cut double helical gearing is very satisfactory, efficient and silent.

Modern practice is to enclose all gearing in cast iron, cast steel or welded steel oil baths.

The strength of spur gearing can be determined by the Lewis formula (page 976, Vol. I). For crane work Class 'C,' gears are used.

Gearing with stub teeth and 20° involute is advised for crane work, as the strength is greater than standard 14½° involute, by 30 per cent. with a rack, to 50 per cent. with a 12T pinion. The addendum is 0·27 of the circular pitch and the dedendum 0·32. With diametral pitch the outside diameter of the wheel is determined by adding 1·75 to the number of teeth and dividing by the pitch.

Worm wheels should consist of a phosphor bronze rim fixed on a cast-iron or steel centre, and have machine-cut teeth. The worm should be of high tensile forged steel and need not be hardened unless facilities exist for grinding the threads. In determining the size of worm gearing the heating effect must be considered as well as the strength.

SHAFTS.

In designing shafts the resultant loads should be determined and the effect of combined bending and twisting moments considered. The equivalent twisting moment due to combined bending and twisting is equal to

$$BM + \sqrt{(BM)^2 + (TM)^2},$$

where,

$$BM = \text{bending moment}; \quad TM = \text{twisting moment.}$$

Stresses should be low on account of the frequent reversal and impulsive loads common with crane work. Where keys are fitted, the shafts should be increased in diameter.

Where shafts are subjected to severe reversing stresses tangential keys should be fitted as they are less likely to work loose than the ordinary key and can be more easily adjusted.

BEARINGS.

For cheap hand cranes the bearings are usually of cast-iron and not bushed. All power-driven cranes should have gunmetal bushed bearings. The bearing brackets should be fitted with caps, and the bushes be in halves. Studs for securing the caps are not desirable. Tee-headed bolts should be used, and these should fit in a slot accessible from the outside of the pedestal, so that the bolts can be withdrawn after the cap is removed and without disturbing the steps. The pressure per sq. in. for shafts running at medium speeds should not exceed 600 lbs. For slow-moving pulleys and axles the pressure may be as high as 1,200 lbs. per sq. in. Fast-running shafts, such as motor spindles, should have ring lubrication. Axles are frequently fitted with float lubrication, which consists of a cylinder of cork floating in an oil-well formed in the inverted cap of the bearing and by contact with the axle supplying the necessary lubricant. But these are being gradually superseded by spring-loaded felt pads. Screw-down and spring-feed lubricators with grease are very satisfactory if they are given attention. The best practice is to install a closed high pressure grease lubrication system to all bearings, fed from a central reservoir, which automatically feeds a charge of lubricant to each bearing whenever a pressure pump is operated.

It is desirable to fit ball or roller bearings to cranes in constant operation as the smaller power consumption due to the reduced frictional losses in the bearings will soon pay for the additional cost.

ROPES (*see* SECTION XXI, PART III, *p.* 933, VOL. I).

CHAINS.

Chains should only be used for grabs or situations where the fine strands of a rope would be damaged by abrasion. The working load for a welded iron chain is given by the rule.

$$W = \frac{(\text{dia. of iron in } \tfrac{1}{8} \text{ in.})^2}{9}.$$

Weldless steel chains are about 13 per cent. stronger, and make good slings on account of **their lower weight. Steel plate driving chains are sometimes used for drives instead** of gearing, **and give good results if properly designed. The burden chain for large ingot rotating** gears is **best constructed of one inner and two outer links connected by steel studs.**

(See also page 169, Vol. I.)

BRAKES.

One of the most important features on a crane is the brake, and this should have a large factor of safety. The brakes in common use are electric, automatic solenoid, centrifugal, automatic mechanical, and manual brakes. Hydraulic and pneumatic brakes are used for heavy loads, or where extremely fine control is desired.

A typical arrangement of a solenoid brake is shown in fig. 2. The brake is applied by the weight and released by a solenoid magnet connected in the motor circuit so that it is excited immediately current is switched on to the motor.

If F is the turning force on the rim and μ the coefficient of friction between the brake shoes and sheave, then the weight required to apply the brake equals

$$\frac{C \times M \times F}{H \times L \times 2\mu}$$

where C, M, H, and L refer to the dimensions shown in fig. 2.

The weight of the solenoid plunger and levers must be taken into account, and about 10 per cent. is usually added for friction in joints.

The centrifugal brake is designed to control speed, and does not serve to sustain the load at rest. In one form it consists of a series of weighted slippers which are pressed by centrifugal action on the inner rim of a fixed annulus, and thus have a retarding effect. The brake shoes should be slightly lubricated and ample radiating surface allowed to dissipate the heat generated.

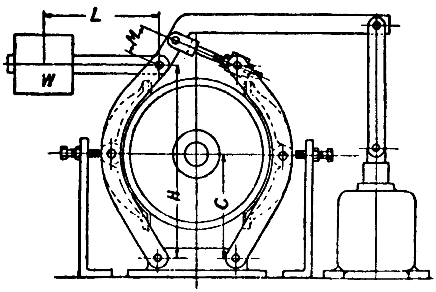


FIG. 2.

The automatic mechanical or Weston disc brake so long used on heavy cranes as a secondary brake for lowering, has been superseded by electrical systems of lowering control. Where two holding brakes are required on the hoist motion, one is arranged to come into action immediately the current is cut-off and the other to have a slightly delayed action, in order to avoid the shock which would result from double braking when stopping the load.

The pitch of the helical **faces is determined by the following formula:**

$$T = \frac{PL}{R \times \mu \times N}, \quad \text{and} \quad \frac{PL}{Tr} - f = \tan A\ ;$$

therefore,

$$\text{lead of helix} = \tan A \times 2\pi r$$

where,

R = average radius of disc ;	**μ = coefficient of friction = ·07 ;**
PL = torque on driving wheel ;	**r = mean radius of helical faces ;**
A = angle of helix ;	**T = thrust or pressure on disc ;**
N = number of rubbing surfaces ;	**f = coefficient of friction of helical faces.**

The brake should be proportioned so that an angle of helix of about 10° will give the requisite pressure. The brake should run at a speed not exceeding 150 revs. per minute, and the pressure on the discs should not be more than 200 lbs. per sq. in. and on the helical faces 500 lbs. per sq. in.

Manual brakes are usually of the band or post type. Band brakes should have a bar fitted round the outside of the strap with adjusting screws to ensure that when the brake is released the strap will come away evenly from the sheave. The connecting rods on manual brakes are being replaced by hydraulic transmission by which pressure on the foot pedal is transmitted by a column of liquid in a tube to the actual brake lever. This allows for the foot pedal being placed in any position convenient to the operator and at any distance from the brake.

Band brakes are generally only suitable for a torque in one direction, i.e. away from the fixed end of the strap.

To find the pull to be applied to the loose end of the strap, the following formula may be used:

$$P = \frac{F}{e^{\mu\theta} - 1},$$

where,

P = pull; μ = coefficient of friction;
F = turning force on rim of brake sheave; θ = angle of sheave embraced by strap in radians.
e = base of Napierian logs.

Brake sheaves are made of cast iron and cast steel, the latter material whilst being less likely to fracture than cast iron wears more quickly and where metallic bonded friction surfaces are used scores badly.

Special alloy cast irons which have a higher tensile strength than ordinary cast iron and harder surface are recommended for cranes and transporters working continually at high speeds.

The brake shoes or strap may be lined with wood, leather, or one of the asbestos fabrics such as 'Chekko' or Ferodo lining; wood and leather are inclined to char and glaze and the coefficient of friction is reduced. Bonded asbestos is not subject to these objections and maintains a constant coefficient of friction, which in practice may be taken as 0·3.

Electric cranes may be controlled by dynamic or rheostatic braking if direct current is used. Dynamic braking with potentiometer control is to be preferred to plain rheostatic braking, as practically automatic speed control is obtained in the lowering direction irrespective of the weight of the load. Heavy rushes of current and mechanical shock can be produced on the motor by either system of electric braking, and some form of current limit control should be provided.

Speeds up to twice full load hoisting speed can now be obtained without difficulty in lowering the light hook under potentiometer control.

A series solenoid should be used with this system of braking, as the current may fall too low to excite a series coil sufficiently to hold off the brake.

With alternating current where the duty is severe reverse current braking can be used to slow down the load before applying the solenoid brake. This reduces the wear on the brake.

To control the lowering of heavy loads at slow speeds, a hydraulic brake is the most efficient. In principle this consists of either a reciprocating or rotary pump with closed circuit and throttle valve.

The pump is in the train of gearing between the motor and the rope drum, and by means of a free wheel drive is inoperative in the hoisting direction. On lowering, the solenoid or other brake is released, and the descending load drives the gearing and pump. The speed is controlled by the throttle valve, and very fine regulation at constant speed can be obtained. The system is especially suitable for placing guns, turbines, and engines on board ship.

For gun tempering, armour treatment, and processes where immersion in a cooling bath is necessary, it is essential that the load should be rapidly lowered at speeds varying from 60 ft. to 150 ft. per min.

The lifting speed is about one-tenth of these speeds, and the usual practice is to disconnect the rope drum from the gearing and motor when lowering.

A coil clutch is used for the purpose, and a powerful brake must be fitted to control the lowering speed and sustain the load. Manual brakes have proved satisfactory for this purpose, particularly in conjunction with a centrifugal brake for speed control. On cranes of over 100 tons capacity elaborate electro-hydraulic systems have been used with success, but they are very costly.

Electrical Equipment.

Cranes may be worked by either direct or alternating current, but the former is to be preferred for the following reasons:

1. Series-wound motors may be used with high starting torque and speed variation in proportion to load.

2. Very fine control can be obtained and dynamic braking used.

3. Less wiring than with slip-ring motors.

Alternating-current motors usually require less attention than direct-current motors, but the starting torque is limited and the running speed is practically the same under light and full load conditions. The starting torque for alternating-current motors should be not less than two-and-a-half times the full load torque, and in fixing the horse-power it is advisable to make it about 10 per cent. higher than for direct-current motors for the same duty.

With reference to paragraph 1, it should be noted that whilst with ordinary series motors the speed increases with a reduction in the load. the variation is not very great between full and approximately half load, and as in many cases light loads are more frequently lifted, mechanical change gear is used to increase the speed and take advantage of the full power of the motor. This can, however, be eliminated by means of a special two-speed motor, designed to develop its full horse-power at two definite speeds besides having the usual series characteristic. The alteration in speed is obtained by diverting part of the field current, and with the Scott Bentley discriminator control this can be done automatically, thus relieving the operator from the trouble of changing gear and ensuring the correct speed for the weight of the load. The efficiency of the crane is increased without any added complications.

Squirrel-cage motors are not in general suitable for crane work, although a number of cranes have been equipped with special squirrel-cage motors having high-resistance rotors and giving twice full load torque at starting. The starting current is about four to five times the full load current, and the switchgear should be of robust design. These motors are not suitable for a heavy duty or for where more than about 20 h.p. is required.

The alternating current commutator motor is now a practical proposition for crane service and will give a speed range of 10 to 1 and controls the speed when lowering as well as when hoisting.

All motors should preferably be of the totally enclosed type. For ordinary engineering shop service crane motors rated to develop their full power for a period of half-an-hour with a temperature rise of 90° F. are quite suitable. For steel works and other situations where the duty is severe, the rating should be for one hour. It is desirable to have the yoke and bearings made in halves so that easy access may be had to the interior of the motor.

A special design of direct-current motor known as the Mill type is largely used on steelworks cranes. It has a steel carcase made to hinge open along the centre line to permit of easy removal of the armature. The armature is of small diameter to reduce inertia stresses on reversal, and the insulation and winding are such as to permit of stalling the motor without injury. It was originally developed for service in rolling mills, hence the name. These motors have 'Class B' insulation and are rated for a temperature rise of 75° C.

See B.S.S. No. 168, 1936.

CONTROLLERS.

For ordinary crane service and motors up to about 60 h.p. at not less than 440 volts the drum controller gives very good service. There should be ample room inside the controller for connections, all parts accessible, finger tips and drum contacts easily renewable, insulation of mica or micanite, and a definite step should be felt as the barrel is moved round to each contact. Magnetic blow-out should be fitted on direct-current controllers.

For frequent operation and large currents a solenoid operated or contactor controller should be fitted. The contactor switches handle the main current to the motor and are controlled by a small drum, or master controller, which deals only with the shunt current required to excite the contactor coils. A development of this system is the Holme Portable Remote Control, in which the master controller can be carried about by the operator and connection is made to the contactor panel by a multicore flexible cable.

Creeping control for foundry or riveting work is obtained by inserting resistance in parallel with the motor armature circuit, thus magnifying the strength of the field and enabling full-load torque to be obtained at a very slow speed.

Cam operated controllers have outwardly the appearance of drum controllers but the fingers consist of a series of simple contractors actuated by a number of cams mounted on a vertical spindle rotated by the controller handle. They are used where the current is considered rather heavy for an ordinary drum controller and yet not sufficiently heavy to justify an automatic controller.

On alternating current circuits speed regulation is obtained by resistance in the rotor circuit, but the same range of speeds cannot be obtained as with direct current, particularly on light loads.

When creeping speeds are required with alternating current the most practical method is to load up the motor with a manual brake, and then use the controller and resistance to adjust the speed.

A recent development is to use two motors connected through a differential gear. When both motors lift in the same direction the crane works at full speed but on reversing one motor the speed is reduced to about one-tenth of normal. The operation is entirely from the controller and the speed may be kept constant for an indefinite time.

Up to about five h.p. resistances are usually made of wire-wound porcelain bobbins, and above this power consist either of stamped steel grids or a drawn continuous strip bent into grid form. The grids should be well stiffened to eliminate vibration and sherardised to prevent rusting. In corrosive atmospheres, special cast-iron grids give better service than stamped or coiled steel grids.

Resistances are usually rated to be in circuit two or five minutes out of every fifteen minutes, with a temperature rise not exceeding 350° to 400° F. For creeping control ten minutes or even continuous rating is desirable. See B.S.S. No. 587–1940.

SWITCHBOARDS.

Many cranes are fitted with a main switch and fuses for the protection of each motor circuit, but it is desirable that these should be replaced by a main automatic circuit breaker and relay switches for each branch circuit. The relay switches do away entirely with fuses and obviate the loss of valuable time at critical moments due to replacing them. Each switchboard should have a pilot lamp and socket for portable lamp connection. It is advisable, although not essential, to fit an ammeter and a voltmeter.

LIMIT SWITCHES.

Limit switches to prevent overwinding should be fitted on all cranes, and if the speed of working is moderate, a shunt limit switch is suitable, connected in the no-volt coil circuit of the main breaker on the switchboard, so that in the event of the switch coming into operation the main circuit is opened at the breaker.

Special contacts can be fitted on the controller to short-circuit the limit switch on bringing the controller on to the first notch in the reverse direction. The breaker can then be reset and the motor reversed. The limit switch should be of the totally enclosed self-resetting type. On high speed cranes, a main current limit switch is desirable, as it can be designed to create a dynamic braking circuit whenever it is operated, and thus assist the solenoid brake in promptly stopping the motion.

SOLENOIDS.

Solenoids for brake operation are preferably of the series type on direct-current circuits, but if of the shunt type, a discharge resistance should be fitted to take up the induced current on breaking circuit. For alternating circuits solenoids are preferably connected in all phases. The rating of a solenoid should be the same as the motor with which it is in circuit. The coils should be enclosed as a protection against damage, dirt, and damp.

Large solenoids are being superseded by Thrustors which consist in principle of a small motor operating a pump producing a hydraulic pressure on a plunger connected to the brake lever. The apparatus is self contained in a casing similar to a direct current solenoid and is made in a range of sizes up to a maximum push of 800 pounds through eight inches. Thrustors operate either on direct or alternating current, and are preferable to solenoids on alternating current circuits.

The Lewis brake solenoid made by Ransomes & Rapier, Ltd., has two plungers, one sliding inside the other. The inner one is connected to the brake lever and is free to move independently of the outer plunger or armature. When the magnet is excited the two plungers are automatically and mechanically coupled and act as one in releasing the brake, but when the current is switched off they are disconnected and the inner plunger is free to follow the movement of the brake lever independently of the stroke of the magnet. As the brake surfaces wear the inner plunger is partially withdrawn from the outer, but owing to the automatic clutching arrangement, the full stroke of the magnet is always effective and releases the brake freely, obviating the rubbing and wear which take place with the ordinary brake if the shoes are not occasionally adjusted.

WIRING.

For crane work the current density may be in excess of ordinary practice for shop motors owing to the intermittent service, and the I.E.E. has drawn up a set of rules for this duty. See also B.S.S. No. 327, appendix iv.

Cables, unless lead-covered and armoured, should be enclosed in metal conduits. Current is supplied to travelling cranes from bare conductors stretched along the track by means of slipper or trolley collectors. The bare conductors may be of copper wire, or, as is now common practice in steel works, steel tee or angle bars ground on the sliding face.

For jib cranes the conductors may be carried overhead or in a conduit. An alternative system is to run the cables in a closed conduit with junction boxes at intervals from which connection can be made by plug and flexible cable. A drum should be fitted on the crane to coil up the slack cable.

The drum may be hand operated if the crane is infrequently moved or if the length of travel is small, not exceeding 50 ft., and the cable light a spring can be used to automatically revolve the drum as the cable becomes slack and prevent loose coils lying on the ground. Where space will permit, it is better to have a weight-operated overhauling gear to exert a torque on the drum, and lengths of heavy cable up to 150 ft. long can be handled in this manner. Cable drums geared to the travelling motion are inadvisable owing to wheel slip on the rails, which may cause a considerable tension on the cable, with resulting damage.

Electro-Lifting Magnets.

Electro-lifting magnets are largely used in connection with cranes for handling all classes of iron and steel materials, and their application shows a considerable saving in time and cost compared with manual labour or even ordinary crane service with slings.

For lifting pig iron, small and heavy scrap, and large solid masses, a single circular magnet is adopted, and for lifting long and wide plates two or more of these magnets are used spaced along a beam which can be slung from the crane hook.

As regards the *lifting of hot material*, circular magnets are unsuitable, because they offer such large heating surfaces, and the windings are so close to the heat that there is bound to be overheating. The Phœnix bi-polar magnet offers the minimum surface to the heat, and when constructed for lifting hot material the pole pieces are extended about 8 ins. at the bottom to lift the coils farther away from the heat; the pole faces are then at a distance of 12 ins. from the coils.

Steel may be handled by magnets up to a temperature of about 700° C.

The following table gives the approximate weights and lifting capacities with ingot and pig-iron for various sizes of circular lifting magnets :—

Diameter of Lifting Magnet.	Weight of Magnet.	Safe Load of Solid Steel Ingot.	Approximate Load of Pig Iron per Lift.	Current Consumption.
24 in.	5½ cwt.	5 tons	—	1·75 k.w.
36 ,,	16 ,,	7 ,,	9 cwt.	3·5 ,,
42 ,,	25 ,,	10 ,,	14 ,,	5 ,,
54 ,,	39 ,,	18 ,,	23 ,,	7·5 ,,
66 ,,	60 ,,	30 ,,	40 ,,	12 ,,

SECTION XXXVIII

PART II

LIFTS AND ESCALATORS

(Contributed by Col. E. B. Rook, T.D.)

ELECTRIC LIFTS.

These notes are confined to electric lifts, which for reasons of safety, cheaper operating costs, and availability of power supply have superseded the hydraulic types for all but a few special applications. The electric lift is a complex machine involving a number of individual parts, which in the main conform with normal engineering practice in their design and production. It is not the intention in these notes, therefore, to discuss such components, but rather to treat the lift as an entity and to consider those matters which affect its application and use as such.

Much useful work has been done by various bodies in recent years aimed at laying down standards and codes of practice for the design of lift installations, loads and speeds to suit various applications. These are welcome both as a means of raising the general standard of lift practice and for their tendency to do so without raising costs seriously by virtue of savings in production costs which should result from the standardisation of loads, speeds and ancillary lift equipment they advocate. Information on such standards and codes have been published from time to time, notably by the Building Industry National Council, The British Standards Institution and in Post-war Building Studies, No. 9, by H.M. Ministry of Works. These notes comply with the recommendations made in such publications and where extracts from them have been used are now gratefully acknowledged.

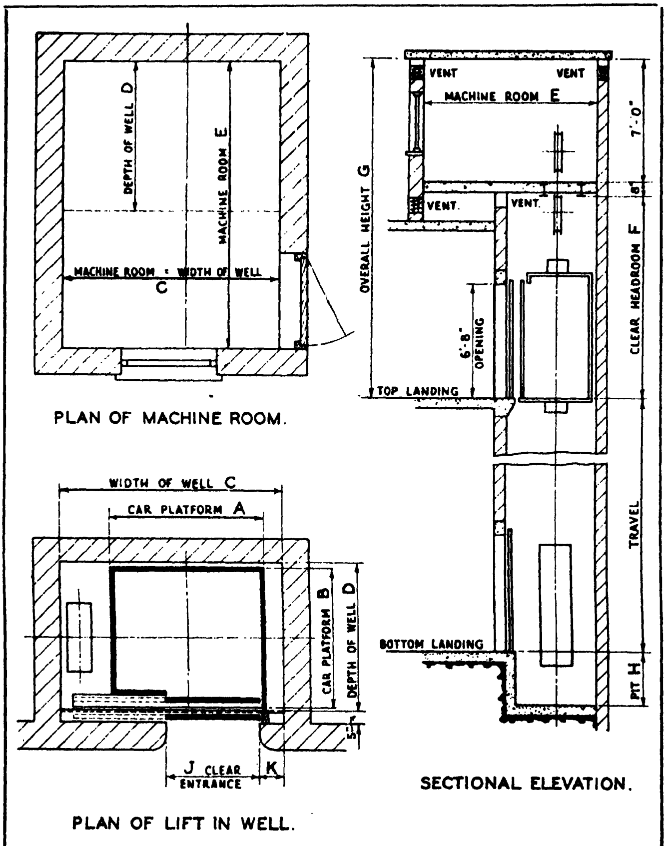
Recommended Layouts and Sizes for Lift-Wells and Machine Rooms.—The information given in the diagrams and tables contained in figs. 1–5 cover the use of electric lifts for those applications which lend themselves to a measure of standardisation and are, therefore, recommended for use wherever possible. Applications outside those thus covered, *e.g.* heavy or special purpose goods lifts and high-speed (above 300-ft. per minute) passenger lifts, need considering on their merits in conjunction with the lift maker. Nevertheless, they will be found to have many features in common with the details and sizes given in these figures and the amplification of them which is given below.

The Machine Room.—This accommodates the driving machine, of either the geared or gearless traction type, and its control panel and accessories. It may be placed above or below the lift-well, the former being the better position resulting in a reduced load on the building, lower capital cost, a smaller lift-well for a given size of car, reduced power consumption and cost of rope renewals. Machine rooms for high speed lifts with gearless traction should always be above the lift-well. Wherever it is placed it must be dry and weatherproof, well lighted both naturally and artificially, properly ventilated and of fire-proof construction.

The Lift-Well.—It is essential that lift doors and machinery operate as quietly as possible, but all noise cannot be eliminated. The lift-well should, therefore, be fully enclosed, even when occupying the space available in a staircase well, and so positioned as to result in the minimum of transference of audio frequencies through the main structure. Even if this be done it will remain advisable to noise insulate the lift machine by mounting it on sound insulators or a concrete block insulated from the machine-room floor by suitable anti-vibration material.

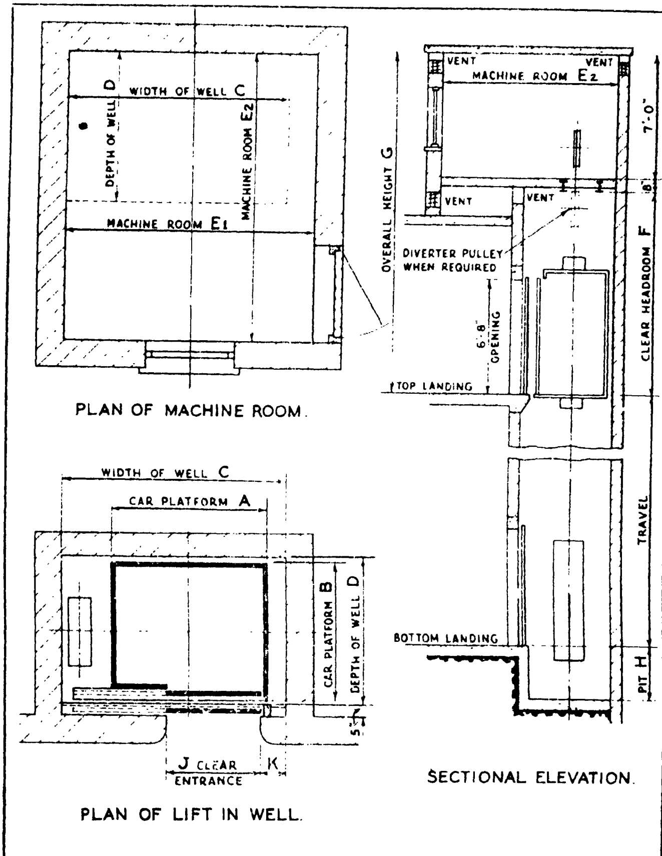
The lift-well must be finished flush internally and its use restricted to that of the lift installation, no other services being allowed in it despite the attractive position it often offers for them. Use of a totally enclosed lift-well has the added advantage of providing easy facilities for car and counterweight guide rail fixings at frequent intervals, which will add materially to the rigidity of the installation and so improve running conditions and may reduce the size of rail required.

To guard against the possibility of a totally enclosed lift-well being the means of spreading fire by acting as a flue or chimney, all landing entrances need to be fitted with fire-resisting doors or metal shield gates, and the well enclosed with solid fire-proof walls. Nevertheless, such lift-wells need to be well ventilated.



LOAD		NETT CAR AREA	OUTSIDE PLATFORM		LIFT WELL		MACHINE ROOM	CLEAR HEAD-ROOM	TOTAL HEAD-ROOM	PIT DEPTH	WIDTH OF ENTRANCE	RETURN
PERSONS	LBS	SQ. FT.	WIDTH A	DEPTH B	WIDTH C	DEPTH D	LENGTH E	F	G	H	J	K
4	600	8½	3'-8"	3'-1"	5'-8"	3'-4"	7'-3"	12'-1"	19'-9"	3'-0"	2'-3"	10"
6	900	12¼	4'-2"	3'-8"	6'-2"	3'-11"	7'-6"	12'-1"	19'-9"	3'-0"	2'-6"	10"
8	1200	16	4'-8"	4'-1"	6'-8"	4'-4"	10'-0"	12'-10"	20'-6"	3'-0"	2'-9"	10"
10	1500	13½	5'-1"	4'-6"	7'-3"	4'-9"	10'-2"	12'-10"	20'-6"	3'-3"	3'-0"	11"

FIG. 1.—Lifts for Industrial Workers' Flats—with Solid Sliding Doors—Speed 100 F.P.M.



							SPEED 100 FT PER MIN					SPEED 150/200 FT PER MIN.						
LOAD		NETT CAR AREA	OUTSIDE PLATFORM		LIFT WELL		MACHINE ROOM		CLEAR HEAD ROOM	OVERALL HEIGHT	PIT DEPTH	MACHINE ROOM		CLEAR HEAD ROOM	OVERALL HEIGHT	PIT DEPTH	WIDTH OF ENTR'CE	RETURN
PERSONS	LBS	SQ FT	WIDTH A	DEPTH B	WIDTH C	DEPTH D	WIDTH E1	LENGTH E2	F	G	H	WIDTH E1	LENGTH E2	F	G	H	J	K
4	600	$8\frac{1}{2}$	3'-8"	3'-1"	5'-8"	3'-4"	5'-8"	7'-3"	12'-1"	19'-9"	3'-0"	-	-	-	-	-	2'-3"	10"
6	900	$12\frac{1}{4}$	4'-2"	3'-8"	6'-2"	3'-11"	6'-2"	7'-6"	12'-1	19'-9"	3'-0"	-	-	-	-	-	2'-6"	10"
8	1200	16	4'-8"	4'-1"	6'-8"	4'-4"	6'-8"	10'-0"	12'-10	20'-6"	3'-0"	7'-6"	11'-5"	13'-4"	21'-0"	4'-0"	2'-9"	10"
10	1500	$19\frac{1}{2}$	5'-1"	4'-6"	7'-3"	4'-9"	7'-3"	10'-2"	12'-10	20'-6"	3'-3"	7'-6"	11'-8"	13'-4"	21'-0"	4'-0"	3'-0"	11"

FIG. 2.—Lifts for Small Office Buildings with Solid Sliding Doors. Speeds up to 200 F.P.M.

Positioning of Lift-Wells.—Decision as to the best position for the lift-well will be dictated to some extent by the interior planning and organisation of the building concerned. This will be especially so in the case of goods lifts, but some general rules are applicable in the case of passenger lifts for office buildings and departmental stores. In the former, the lift or lifts are best sited in a

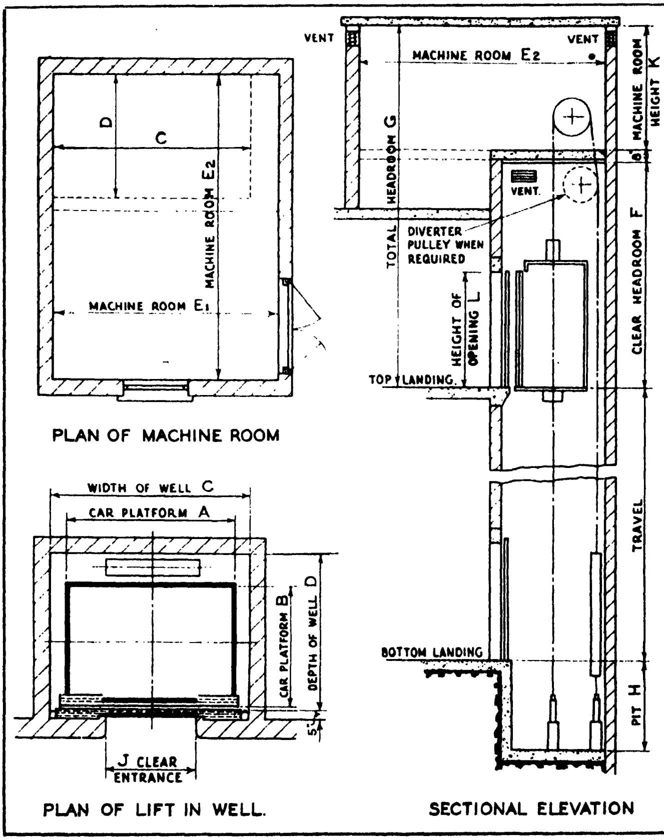


FIG. 3.—Lifts for General Office Buildings.
(To be read in conjunction with Tables 1 and 2—Page 821.)

separate lift-well adjacent to the main staircase. If the building be of sufficient height to warrant high-speed lifts it will be found generally that more than one lift will be required, in which case it will be best to arrange the two or more lifts side by side in a common well. So doing involves the advantages of providing a common machine room for all lifts, that the control of the lifts can be interconnected more conveniently, and of greater convenience in service as the landing entrances are adjacent on each floor.

In the shop or small store the lift or lifts are best sited adjacent to the main traffic aisle through the premises and need not be adjacent to the main staircase. Approach to the lifts should be

well defined and clear of obstructions. Where two or more lifts are concerned use of a common lift-well will carry similar advantages to those given above for lifts in office blocks. In the larger stores where escalators are employed the lifts will be the secondary form of vertical transport and must give pride of place in consequence. The former will lie in the main flow of traffic to and from the principal entrances, whilst the lift installation will occupy a position in the rear of the premises.

TABLE 1.

LOAD		NETT CAR AREA SQ. FT.	OUTSIDE PLATFORM		LIFT WELL		WIDTH OF ENTRANCE J
PERSONS	LBS.		WIDTH A	DEPTH B	WIDTH C	DEPTH D	
10	1500	19	6'-0"	4'-0"	7'-4"	5'-0"	3'-0"
15	2250	26	6'-9"	4'-7"	8'-1"	5'-7"	3'-6"
20	3000	33	7'-6"	5'-3"	8'-10"	6'-4"	4'-0"

These dimensions apply only to travels not exceeding 100 ft. For dimensions E, F, G, H, K, L and M, see Table 2.

TABLE 2.

LOAD												
LOAD	PERSONS		10				15			20		
	LBS		1500				2250			3000		
SPEED FEET PER MINUTE			100	150	200	300	150	200	300	150	200	300
MACHINE ROOM	WIDTH	E1	7'-6"	7'-6"	7'-6"	9'-0"	8'-1"	8'-1"	9'-2"	8'-10"	8'-10"	10'-0"
	LENGTH	E2	11'-3"	12'-6"	12'-6"	14'-3"	13'-0"	13'-9"	15'-3"	15'-6"	16'-6"	16'-9"
CLEAR HEADROOM		F	11'-7"	12'-1"	12'-1"	12'-10"	12'-4"	12'-4"	13'-7"	12'-4"	12'-4"	14'-7"
TOTAL HEADROOM		G	19'-3"	19'-9"	19'-9"	21'-0"	20'-0"	20'-0"	22'-3"	21'-0"	21'-0"	23'-9"
PIT DEPTH		H	3'-3"	4'-0"	4'-0"	5'-0"	4'-0"	4'-0"	5'-0"	4'-0"	4'-0"	5'-0"
MACHINE ROOM HEIGHT (MIN.)		K	7'-0"	7'-0"	7'-0"	7'-6"	7'-0"	7'-0"	8'-0"	8'-0"	8'-0"	8'-6"
LANDING ENTRANCE HEIGHT		L	6'-0"	6'-8"	6'-8"	6'-5"	6'-8"	6'-8"	7'-0"	7'-0"	7'-0"	7'-0"

Where speed exceeds 300 ft. per min. or the travel exceeds 100 ft. the lift engineer should be consulted.

Where two lifts are adjacent the width of the lift-well will be twice dimension ' O ' (Table I) + 4 in.

(Tables to be read in conjunction with Fig. 3—Page 820.)

Number of Lifts Required.—Decision regarding the number of lifts necessary to satisfy the traffic requirements of any particular building is not easy and there are no hard and fast rules available to help. The number of goods lifts required is less difficult to establish than passenger lifts for the reason that the probable weight and bulk, total and individual, of freight items to be carried and their cycle and frequency of movement can be estimated with fair ease. The probable passenger lift traffic is less easy to estimate since it involves a number of factors which cannot be known with any degree of accuracy until the building is in use.

In the case of industrial workers' flats the number of passenger lifts required in all probability settles itself by the very grouping of the flats into their individual blocks and access services. For office blocks, the number of lifts required, and their capacity and speed, since these aspects will affect the number, is better based upon the estimated population per floor of the building than by using any fixed relation of lifts to floor areas. The effects of interfloor traffic resulting from the siting of staff canteens, lavatories, etc., on particular floors must be taken into consideration. In general it may be taken that if lift capacity is provided capable of handling the normal traffic peaks, occurring morning, mid-day and night, it will be sufficient for the other periods of the day.

In shops or small stores not provided with escalators, the number and capacity of lifts should be determined from the number of customers in the building during peak 'sale' periods. The maximum peak is likely to occur between noon and 3 P.M., and the traffic will be in both directions. A good guide to this population is obtained by allowing one person per 30 sq. ft. of net rentable floor area. The influence on traffic requirements of restaurants, demonstrations and exhibitions on upper floors must be taken into account. For the larger stores equipped with banks of escalators giving up and down service, passenger lifts become a secondary feature of the traffic problem and are needed only for those who dislike or cannot use escalators due to infirmity, or wanting to travel several floors in one journey.

Loads and Car Sizes.—No rules are possible in the case of goods lifts since both the load to be carried, and the car size and width of car opening required to accommodate it will be determined by the physical characteristics of the freight to be handled. The car size corresponding to the loads given in the Tables of figs. 4 and 5 must, therefore, be treated as suggestions requiring a check on the basis of the type of goods to be carried.

The basis for determining the rated load for passenger lifts is the average person whose weight is taken to be 150 lb. It must be taken as axiomatic that the floor area provided for a passenger lift car must not be greater than that required to accommodate the number of persons corresponding to the rated load. To do otherwise would involve the risk of overloading the equipment. It can be achieved by working to the following allowances—

For lifts rated to carry up to 10 persons (1,500 lbs.) allow 2 sq. ft. of floor area per person.

For loads in excess of this reduce this allowance to 1·75 sq. ft. per person.

Recommended Speeds.—Selection of lifting speed involves factors difficult to express as hard and fast rules. Speed, by affecting travelling time has a considerable effect upon the traffic handling capacity of a lift. Owing to the time lost loading and unloading, however, this capacity does not increase in direct proportion with increases of speed. For this reason, with a passenger lift serving a travel of 80 ft. a 100 per cent. increase in speed will have the following effect upon capacity :—

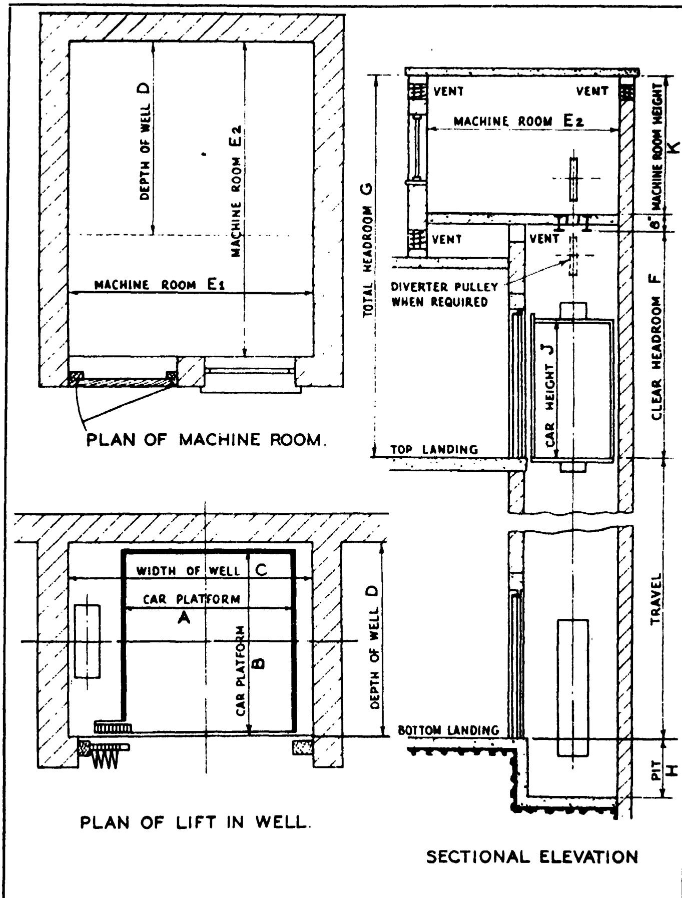
Speed increased from 150 to 300 ft. per minute—approximately 50 per cent. increase in traffic capacity.

Speed increased from 200 to 400 ft. per minute—approximately 33 per cent. increase in traffic capacity.

This disproportionate effect of speed upon capacity is even more marked in the case of goods lifts where the time occupied in loading and unloading forms a greater part of the average circular trip time than with passenger lifts. It is seldom in consequence that there is sufficient capacity advantage to be gained by operating goods lifts at more than 100 ft. per minute, except perhaps where bulk freight is trucked into and out of the lift car, and where there is sufficient traffic to warrant the use of an attendant-operated machine. In such cases speeds of 150 or 200 ft. per minute may be justified.

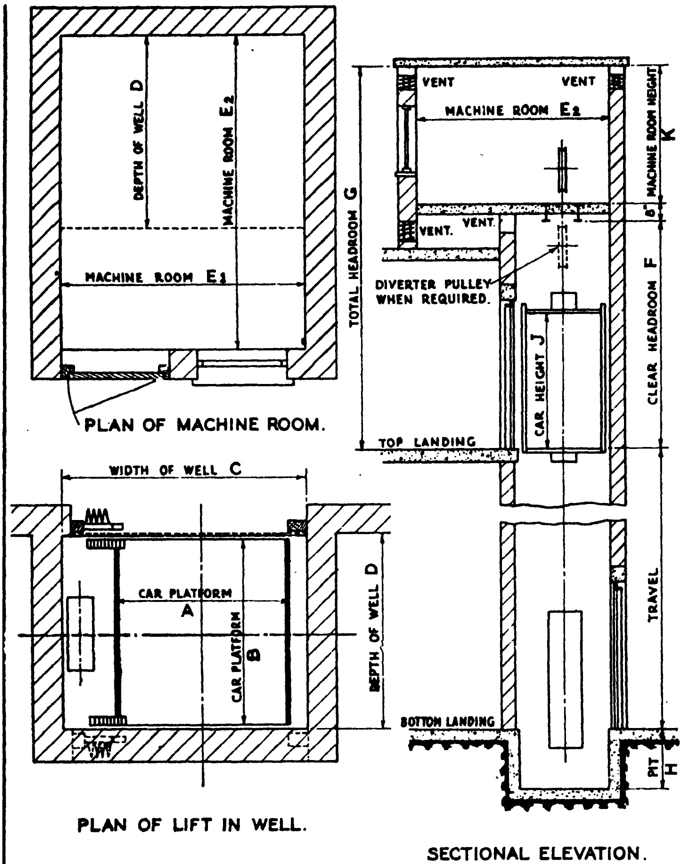
For passenger lifts the selection of speed must be guided by the psychological effect upon the user, who will become restive if called on to travel long distances at slow speeds, but this aspect must be related to the type of service required of the lift. Obviously, lifts normally giving floor to floor service may be operated at lower speeds than those giving terminal service.

The effect of speed upon capital cost must not be overlooked. A speed of 100 ft. per minute represents the maximum for which single-speed motor and control equipment will be suitable and ensure satisfactory levelling for passenger lifts. Even this speed is often too high for goods lifts to be similarly equipped since a higher degree of floor levelling accuracy is demanded, especially where goods are trucked into and out of the lift. For passenger lift speeds in excess of 100 ft. per minute, and goods lifts at any speed in excess of about 40–50 ft. per minute where any accuracy of floor levelling is required, some form of slowing and levelling equipment will be essential and will involve additional capital cost.



LOAD IN CWTS	OUTSIDE PLATFORM		LIFT WELL		MACHINE ROOM		CLEAR HEAD-ROOM	TOTAL HEAD-ROOM	DEPTH OF PIT	CAR HEIGHT	MACHINE ROOM HEIGHT
	WIDTH A	DEPTH B	WIDTH C	DEPTH D	WIDTH E_1	DEPTH E_2	F	G	H	J	K
10	4'-0"	4'-6"	5'-8"	5'-0"	5'-8"	7'-6"	11'-1"	18'-9"	3'-0"	6'-6"	7'-0"
15	4'-6"	5'-3"	6'-4"	5'-9"	6'-4"	10'-6"	11-10"	19'-6"	3'-3"	7'-0"	7'-0"
20	5'-3"	6'-3"	7' 2"	6'-9"	7'-8"	11'-5"	12'-6"	20' 2"	3'-6"	7' 0"	7'-0"
30	6'-0"	8' 0"	8'-1"	8-6"	8'-4"	11' 7"	13' 8"	22-4"	3' 9"	7'-6"	8'-0"
40	6'-9"	10'-0"	9'-1"	10' 6"	11' 1"	10-6"	13 4"	22' 6"	4'-0"	7'-6"	8'-6"

FIG. 5.—Goods Lifts for Office Buildings.—Speeds up to 100 F.P.M.



LOAD IN CWTS	OUTSIDE PLATFORM		LIFT WELL		MACHINE ROOM		CLEAR HEAD-ROOM	TOTAL HEAD-ROOM	DEPTH OF PIT.	CAR HEIGHT	MACHINE ROOM HEIGHT.
	WIDTH A	DEPTH B	WIDTH C	DEPTH D	WIDTH E_1	DEPTH E_2	F	G	H	J	K
10	4'-0"	4'-6"	5'-8"	4'-10"	5'-8"	7'-6"	11'-1"	18'-9"	3'-0"	6'-6"	7'-0"
15	4'-6"	5'-3"	6'-4"	5'-7"	6'-4"	10'-6"	11'-10"	19'-6"	3'-3"	7'-0"	7'-0"
20	5'-3"	6'-3"	7'-2"	6'-7"	7'-8"	11'-5"	12'-6"	20'-2"	3'-6"	7'-0"	7'-0"
30	6'-0"	8'-0"	8'-1"	8'-4"	8'-4"	11'-6"	13'-8"	22'-4"	3'-9"	7'-6"	8'-0"
40	6'-9"	10'-0"	9'-1"	10'-4"	11'-1"	10'-4"	13'-4"	22'-6"	4'-0"	7'-6"	8'-8"

FIG. 4.—Goods Lifts for Office Buildings.—Speeds up to 100 F.P.M.

The following general recommendations on speeds for passenger lifts may be used for guidance—

(i) Selection should be made from the following standard range :—
100, 150, 200, 300, 400, 500, 600 and 700 ft. per minute.

(ii) For industrial workers' flat lifts a speed of 100 ft. per minute is recommended to ensure the use of economical equipment.

(iii) For office and similar buildings—
No of floors served, five or less—200–300 ft. per minute.
" " " six to ten—300–400 " " "
" " " more than ten—not less than 400 ft. per minute.

(iv) For shops and departmental stores—

Small establishments	150 ft. per minute.
Larger "	200–300 ft. per minute.
Large establishments provided with escalators	300–400 ft. per minute with the higher speed used for lifts giving primarily terminal service.

Types and Rating of Lift Motors.—Three-phase alternating current power now being almost universal, motors suitable for such a supply only are considered below. Lift motors are required to start against load and must, therefore, be designed to give at least twice full load torque at starting which may limit the employment of squirrel cage motors where permissible starting currents are fixed at a low figure. It is considered poor practice to use lift motors operating at more than 1,000 r.p.m. and the majority of geared traction machines operate around this figure. Unless site conditions demand otherwise lift motors will be of the screen protected type.

Until comparatively recently lift motors have been rated on the basis of the normal intermittent half or one hour ratings, the former being considered suitable for the type of lift service required in small office buildings, residential flats, etc., whilst the latter was adopted for the more intense service required in busy office buildings, departmental stores and the like. Increased use of two-speed motor equipment involving slowing and levelling conditions in addition to starting has shown previous ideas of rating to be inadequate if over-heating of motors is to be avoided. The modern conception of rating of lift motors is to adopt as a basis the number of starts per hour and to call for motors capable of handling such numbers without exceeding the limits of permissible temperature rise. The following ratings in terms of starts per hour can be taken as a representative guide—

Light duty —Below 60 starts per hour.	Intermittent service, *e.g.* goods lifts, industrial workers' flats, small office buildings.
Normal duty—Up to 120 " " "	Normal passenger lift service.
Heavy duty — " 240 " " "	Departmental stores, busy office buildings, 'under-lifted' buildings.

The types of motors in general use for lifts are :—

Single Speed Slip-ring and High Torque Squirrel Cage Induction Motors.—Suitable for use on passenger lifts operating at speeds not greater than 100 ft. per minute, and for goods lifts at similar speeds where great accuracy of floor level is not required. Use of the squirrel cage type will be limited to cases where starting currents in the region of 3½ times full load current are permitted and where the refinement of rheostatically controlled acceleration is not required.

Two-Speed High Torque Squirrel Cage Motors.—Used extensively in the three to one speed range for passenger lifts up to 200 ft. per minute and goods lifts up to 100 ft. per minute. Capable of use with direct approach forms of levelling equipment, but will give levelling accuracy equivalent only to the lower of its two speeds. Incapable of giving refined conditions of acceleration, slowing and stopping. Requires starting current of 4½–5 times full load if switched direct on high-speed winding. Where this is not possible it can be started through the slow-speed winding, but with less smooth acceleration characteristics.

Two-Speed Tandem Motors.—Its greater speed range of six to one renders this type suitable for passenger lifts up to 250 ft. per minute with both direct approach and corrective levelling to give a high degree of levelling accuracy. Starts as an ordinary slip-ring machine requiring 2½ times full load current and rheostatically controlled acceleration ensures smooth starting. Deceleration and stopping conditions are less refined than with variable voltage equipment.

Variable Voltage Equipment.—Operates on the well-known Ward Leonard principle and is suitable with a geared traction machine for passenger lifts up to 350 ft. per minute and above this speed in its gearless traction form. Useful for goods lifts in the 150–200 ft. per minute speed range where great floor levelling accuracy combined with corrective levelling is required. Can be designed for speed ranges of 25 to 1, in the geared traction types and greater ranges in the gearless. Capable of great refinements in accelerating, slowing, levelling and stopping conditions, and, therefore, employed extensively in high-class installations.

Motor Horse-powers and Consumptions.—The horse-power required for any lift will be not only a function of the load and lifting speed, but also of the overall efficiency of the machine and will,

therefore, vary a little between different makers. It will also be affected by the degree of balance provided. Theoretically the greatest economy in running cost is achieved by balancing the weight of the lift car and its sling and safety gear plus the average load carried over a period. The latter will rarely be a constant figure and it is obviously impossible to adjust the counterweight to suit it. Lifts are, therefore, normally balanced for 40 per cent. or 50 per cent. of their rated load plus the weight of the car, etc., such balancing having proved to result in the most economical operating conditions on the average installation. The following formula will give a good approximation of the b.h.p. required for a lift.

B.H.P. $= W \times S \times C$, where

W = rated load in cwts.

S = speed in ft. per minute.

C = A factor, being 0·0045 if the driving machine is above the lift-well and 0·0048 if below.

This formula assumes a 50 per cent. balance of the rated load and an overall efficiency of 36 per cent. and, therefore, is applicable to geared traction machines only.

The power consumption of electric lifts is surprisingly small and a close approximation to what it will be is given by the following formula :—

$$U = \frac{P \times k \times T}{3{,}600}$$

where U = Energy consumption in b.o.t. units for one complete circular trip, *i.e.* up and down, of the lift.

P = b.h.p. of motor.

k = A factor being 0·65 for a fully loaded trip.
0·52 „ half load trip.
0·6 „ no load trip.

T = the time in seconds for one complete circular trip is given by $\frac{L \times 60}{S}$ seconds

where L = twice the lift travel in ft.

S = speed in ft. per minute.

Control and Signal Systems.—A number of control systems designed to meet varying conditions of lift service are available. In the main they are variations or combinations of the following :—

(i) *Car-switch Control.*

A method of control whereby the movement of the car is directly under the control of the attendant by means of a switch in the lift-car.

(ii) *Automatic Push-Button Control.*

A method of control by means of buttons at the landings and in the lift-car, the momentary pressure of which will cause the lift-car to start and automatically stop at the landing corresponding to the button pressed.

(iii) *Collective Control.*

An improved form of automatic control in which calls from the lift-car and lift-landings are registered, and are answered by the lift-car stopping in floor sequence at each lift-landing for which a call has been registered, until all calls have had attention.

(iv) *Signal Control.*

A method of control in which, although the lift-car is started by means of a car-switch or button situated therein, the stops are determined and registered by the pressure of signal buttons on the various lift-landings, and by pressure of other buttons in the lift-car. With a group of lifts, the signals made by the pressure of 'up-down' landing-buttons are answered by the first available lift-car travelling in the corresponding direction.

The use of the simple forms of automatic push-button control should be limited to where the required lift service is light and intermittent, and the number of floors served small, *e.g.* small blocks of flats and offices. In all large office buildings attendant operated control is necessary, preferably of the signal control type and interconnected if batteries of two or more lifts in centralised lift-wells are involved. It is an advantage, nevertheless, for one lift in each battery to have in addition passenger control, usually of the collective type, to provide for lift service outside normal working hours. Such dual forms of control can be applied usefully wherever it is necessary to provide 24-hour lift service throughout the day, but attendant operation is needed only during certain well-defined peak periods.

The requirements of departmental store passenger lift service are usually met by some form of attendant operated control which can be one of the simpler forms for lifts operating in 150–200 ft. per minute range. Above this range floor finding and levelling should be automatic and outside the control of the attendant. This can be done by using a form of signal control simplified to suit the floor to floor service normally required in departmental stores.

The signal systems required for the intelligent and efficient operation of a lift service will vary with the type of control used. Thus they will range between the inexpensive forms of call bell annunciators or landing 'lift coming' or 'lift here' indicators associated with simple car-switch

and automatic push-button controls respectively, and the comprehensive position indicators and interconnected signal systems associated with collective and signal controls. It must be accepted that properly selected and applied a signal system is not an unnecessary luxury but an essential towards securing the maximum and most efficient service from any lift installation.

Protection of Car and Landing Entrances.—The modern tendency towards solidly enclosed lift-wells requiring fire-resisting doors to landing openings has resulted in the increasing use of solid sliding doors for both car and landing entrances. There can be no doubt these offer the best form of protection and their use is recommended wherever possible and certainly exclusively for passenger lifts. They are available in various types :—

(i) Single leaf sliding to one side.
(ii) Two leaf centre opening.
(iii) Two leaf two-speed, or three leaf three-speed opening to one side.
(iv) Four leaf comprising a pair of two leaf two-speed arranged to give opening to opposite hands from the centre.

The first two of these four types should be used wherever lift-well and car opening dimensions render them suitable. All such doors should be provided with vision panels and tracks in bronze or some other non-corrosive material.

Power Operation of Car and Landing Doors.—The serious accident hazard introduced by the practice of using moving car floors as a means of permitting unoccupied passenger operated lift-cars to be moved with gates or doors open has led to the abandoning of this device. So doing has introduced the intolerable nuisance of such lifts being put out of action by the car door being left open. It is, therefore, essential some form of closer be fitted to the car door which can be either of the spring-operated or motor-driven type, the latter being the better of the two.

Complete power operation of both car and landing doors is now becoming accepted practice and its use is justified by the improvement in lift service given.

It not only eliminates stoppages due to doors on passenger operated lifts being left open, but expedites the service given by attendant operated machines by reducing loading and unloading times. It should be noted that power operation when applied to a centre opening form of solid sliding doors, *i.e.* types (ii) and (iv) mentioned above, will be twice as efficient in respect to opening and closing times, assuming the same door speed, as if it is applied to the other two types. Power operated doors of the former types are recommended, therefore, for lifts required to give a high traffic capacity performance, *e.g.* in large office buildings and departmental stores.

Safety Equipment.—As befits a machine placed at the service of the ordinary public, a lift must be fully equipped with safety devices to prevent accidents and guard against the hazards of mechanical or electrical failures in the equipment. Such devices will include :

(i) *Safety Gears.*

Accidents due to rope failure are prevented by fitting a rope safety gear, which operates on failure and suspends the car upon its guides. For passenger lifts this safety gear must be fitted beneath the car and be operated by an overspeed governor for lifts having a travel exceeding 20 ft. It may be instantaneous in its operation for lifting speeds up to 200 ft. per minute, but above this speed must be designed to be applied gradually.

(ii) *Protection against Over-travel.*

Every lift must be provided with upper and lower normal and final stopping device. It is usual for the normal terminal stopping device to interrupt the control circuit for the direction in which the lift is moving when the device operates, and to automatically re-set itself when the lift is moved away in the opposite direction. The final terminal stopping device on the other hand once operated disconnects the machine from the supply thus preventing further movement in either direction, and is arranged for re-setting by hand.

Buffers for both the car and counterweight are provided in the pit as a final protection against the effects of over-travel. They will take the following forms :—

For lifting speeds not exceeding 75 ft. per minute . .	Buffers of timber or rubber.
" " " " " 300 " " . .	Spring buffers.
For lifting speeds exceeding 300 ft. per minute . .	Oil buffers or their equivalent.

(iii) *Locking Devices for Landing Gates or Doors.*

Every landing gate or door must be fitted with a locking device which will make it impossible for such gate or door to be opened unless the lift is in that particular landing zone and impossible under normal operation to start the lift car in motion unless all landing gates or doors are in the closed and locked position. The design of such locking devices should be such that they fail to safety and that they be tamper-proof. It is recommended that for lifts serving more than two floors at speeds in excess of 100 ft. per minute the operating ramp for landing lock be of the retiring or retractable type to prevent the opening of any landing gate or door while the car is passing through a landing zone to another. It is further recommended that such locking devices be of the pre-locking type in which the landing gate or door must be in the closed and mechanically locked position before the electrical interlock is made and the lift-car capable of being moved.

Interlocking of the car gate or door is provided for electrically only.

ESCALATORS.

Where large crowds have to be moved from one level to another, or where there is advantage in encouraging them to do so, escalators provide a better answer than do lifts. They have a passenger-carrying capacity equalled only by an excessive number of lifts. By eliminating passenger waiting time and the need for lift attendants, escalators give continuous, and therefore better, service at a lower operating cost. Their field of application has widened during recent years from that associated with railway systems into use in departmental stores, subway tunnels, factories—for bulk movement of operatives, office buildings, public exhibitions and amusement parks.

Escalators are built as self-contained units giving service in either the up or down direction between any two levels, *i.e.* simultaneous up and down service requires two units. Control equipment is provided to make them reversible at will and vertical heights of up to 100 ft. have been served by single units, although it is advisable to break down heights in excess of 50 ft. 0 in. into shorter flights. Each unit consists essentially of :—

(i) A fabricated steel truss. This truss is supported from the building fabric :—

For vertical rises up to 20 ft. 0 in.	Two or three point suspension.
" " " " " 30 ft. 0 in.	Three point suspension.
" " " above 30 ft. 0 in.	Four or more point suspension.

(ii) The continuous band of steps contained within this truss plus its associated equipment—step chains, driving end sprocket, reversing carriage, the tracks on which the steps run and by which they are constrained to take up their stepped formation and to level off at either end of the machine.

(iii) The driving unit consisting of motor, worm gear and controller. This and its associated supporting steelwork is situated normally at the head of the escalator immediately beneath the upper level served. A duplex chain drive forms the link between the driving unit and the escalator head shaft.

(iv) The moving rubber handrails and their associated chain and sprocket drive from the escalator head shaft.

(v) The balustrade casing usually decoratively styled.

Escalators are standardised as to angle of inclination and widths, as follows :—

Angle of Inclination.—This may be either 30° or 35° to the horizontal. Table 3 gives the leading dimensions for escalators of either angle and demonstrates the reduced overall length of the 35° type. The 30° angle results in a better ratio of step tread to riser and is, therefore, to be preferred whenever site conditions permit. It is suitable for any vertical rise. The 35° angle, due to its steeper slope, is not recommended for use with rises in excess of 20 ft. 0 in., but within this limit can be used with advantage in many cases owing to its reduced overall length.

TABLE 3.

α	30°	35°
A	8 ft. 4 in.	7 ft. 5 in.
B	H × 1·732	H × 1·428
C	5 ft. 6 in.	9 ft. 7 in. (C and D)
D	8 ft. 0 in.	
E	4 ft. 10 in.	5 ft. 3 in.
F	5 ft. 9 in.	5 ft. 1 in.
G	4 ft. 6 in.	3 ft. 11 in.
H	Vertical Rise	Vertical Rise
J	3 ft. 0 in.	3 ft. 0 in.
K	3 ft. 6 in.	2 ft. 9 in.
L	4 ft. 6 in.	3 ft. 11 in.
M	2 ft. 0 in., 3 ft. 0 in., 4 ft. 0 in.	2 ft. 0 in., 3 ft. 0 in.
N	4 ft. 7 in., 5 ft. 4 in., 6 ft. 2 in.	4 ft. 3 in., 5 ft. 1 in.

Note.—These dimensions will vary with different makes of escalators, but those given can be taken as generally representative of all makes.

Nominal Widths.—These are 2 ft. 0 in., 3 ft. 0 in., and 4 ft. 0 in. and refer to the width between the balustrade casings measured 24 in. above the step nose line. The clear width of step is approximately 6 in. less than these nominal widths. The 4 ft. 0 in. width is not available in the 35° angle machine. Table 3 gives the overall widths of machine corresponding to these nominal

widths. Selection of the most suitable width involves a number of factors, including that of traffic capacity required, but in practice tends to become standardised into the following :—

2 ft. 0 in. Passing into disuse except where overall width of the escalator is a deciding factor due to space limitations. Escalator will accommodate one person per step only.

3 ft. 0 in. Considered adequate for most departmental store applications. Escalator will accommodate two medium size persons per step and will permit passing except between heavily laden persons.

4 ft. 0 in. Use limited in main to railway and subway tunnel applications, but may be adopted for service between lower floors of departmental stores. Escalator will accommodate and permit passing between two heavily laden persons.

The capacity of an escalator in terms of the number of passengers it can transport in a given time is a function of both its width and the step speed. Experience has shown a speed of 90–100 ft. per minute on the incline to be suitable for most applications other than where either vertical rises in excess of 30 ft. 0 in. are involved or exceptionally large peak crowds have to be moved. The 2 ft. 0 in., 3 ft. 0 in., and 4 ft. 0 in. wide escalators will handle 4,000, 6,000 and 8,000 persons per hour respectively when operating at 90 ft. per minute.

Increasing the speed above 90 ft. per minute will increase these capacities in almost direct proportion up to an optimum of 150 ft. per minute, after which further increases have little effect owing to the inability of users to feed on to the escalator sufficiently fast for each step to be occupied as it becomes available. When considering capacity it must be borne in mind escalator service must be provided to handle peak loads over relatively short periods rather than the average over long intervals. Selection of speed must take into account the ability of users to get on and off the escalator rather than its effect upon capacity bearing in mind the purpose for which the escalator is provided—whether as a form of vertical transport people must use to save time and avoid physical exertion as with railway escalators, or, as in departmental stores, as an inducement to move from one level to another.

The horse-power required to operate escalators will vary with the vertical rise involved and the type and make of escalator employed. A rough guide can be obtained from the following empirical formula, which refers to 30° escalators operating at 90 ft. per minute.

$$\frac{0{\cdot}0109\ WH}{E}$$

where E = overall efficiency of the escalator and equal to 56–60 per cent.

W = Nominal width of escalator in inches.

H = Vertical rise in feet.

This formula gives the horse-power required to operate the escalator when fully loaded. For half and no load the power required will be about 64 per cent. and 27 per cent. respectively of this figure. Since there will be considerable periods in the normal daily life of an escalator when the passenger traffic is below the maximum capacity of the machine, the power consumed will be less than that corresponding to the horse power rating and is found in practice to be one-third to one-half of this latter figure. Motors for escalator duty are, of course, continuously rated.

Being a form of public transport it is essential that proper safeguards be provided in escalators to ensure safety to the user. In addition to designing to a high factor of safety and for minimum running clearances throughout the exposed portions of the machine, the safeguards required by various codes of practice for escalators are :—

(i) Prohibition of glass panels in balustrade casings, which are to be smooth and substantially flush throughout their length.

(ii) The track arrangements must prevent displacement of the steps and running gear if a step chain breaks.

(iii) A conspicuously marked 'Stop' push or switch accessible to the public must be fixed at the top and bottom of each escalator.

(iv) Any escalator operating in the ascending direction must be equipped with an anti-reverse device, which will interrupt the power supply and apply the main drive brake in the event of accidental reversal of direction of travel.

(v) Each escalator must be provided with an overspeed governor, which will operate and bring the escalator to rest in the event of the speed exceeding normal by 25 per cent.

(vi) Each escalator must be provided with a broken chain device, which will interrupt the power supply and apply the main drive brake in the event of one or both of the step chains breaking.

(vii) Each escalator must be provided with a main drive brake, which will automatically stop and hold the escalator when travelling or tending to travel in the descending direction should any of the above safety devices function.

Where several escalator units have to be combined into a system giving service to a series of floor levels—*e.g.* in departmental stores—a variety of layouts are possible of which a representative selection is given at fig. 6 and discussed below :—

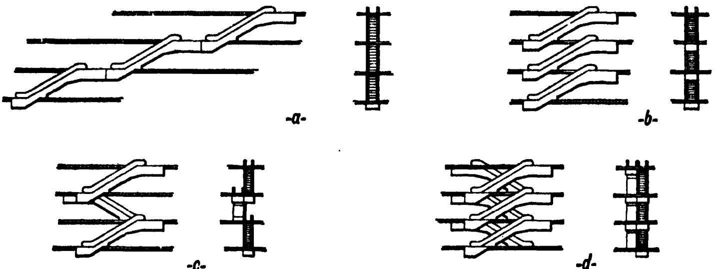


FIG. 6.

Fig. 6*a*.—Provides up or down service suitable for both interfloor service or continuous travel between lowest and highest floors without passengers needing to change direction from one flight to the next. Simultaneous service in the reverse direction will involve a similar bank of machines.

Fig. 6*b*.—Provides similar service to that given by fig. 6*a* layout. It is a more compact arrangement but has disadvantage that passengers travelling between more than two floors must reverse their direction and walk the length of one escalator between flights. Simultaneous up and down service will involve a similar bank of machines, which if placed parallel and close to the first bank will result in up and down traffic becoming confused at the upper and lower landings of each pair of escalators.

Figs. 6*c* and 6*d*.—Examples of compact arrangements to provide both interfloor and continuous travel service. Passengers pass from flight to flight with a minimum of walking. In the double corkscrew bank (fig. 6*d*) the simultaneous streams of up and down traffic are kept distinct from each other at the tops and bottoms of the escalators. These arrangements are without doubt the most suitable for departmental stores.

Fire and escape regulations imposed by Local Government authorities affect the layout and siting of escalators and need to be studied in the initial stages of any project. There is a growing tendency on the part of such authorities to demand special precautions which limit complete freedom to the planner where escalators pass through the floors of a multi-storey building and in so doing involve openings in the fire break between such floors. The expedients required to provide these precautions will vary between the extremes of either siting the escalator system in its own distinct zone, divided from the rest of the building by fire-proof partitions and automatically-controlled shutters, or relatively simple arrangements of sprinkler systems with smoke break curtain walls suspended below the floor openings through which the escalators pass.

SECTION XXXVIII

PART III

AERIAL ROPEWAYS AND BELT CONVEYORS.

AERIAL ROPEWAYS.

(*See also* SECTION XXI, PART III, VOL. I.)

Aerial Ropeways may be divided into two fundamental types:—The SINGLE ROPE or MONOCABLE, and the DOUBLE ROPE or BI-CABLE. In the first, a single endless rope is used both to support and move the loads. In the second, there are one or more supporting ropes and an endless hauling rope. Adapted to these types are the 'Fixed or Non-detachable Clip System,' on the Single Rope Type, and the 'Jig-back' or 'To-and-fro System,' which is applicable to either type.

There is considerable divergence of opinion upon the merits and demerits of the two systems, which is due largely to the fact that a number of plants have been installed without giving due consideration to which type is most applicable to the nature of ground to be traversed, and the duty to be performed.

THE SINGLE ROPE OR MONO-CABLE SYSTEM.

Ropes.

By experience, it has been found the most suitable form of rope on an average length line is a 6·7 Lang's lay, with a specially hard hemp core. On short lines when the rope is subject to a great number of bends per hour around the terminal sheaves, compound lays are sometimes used. Wire cored ropes are also occasionally employed, owing to the fact that they tend to stretch less than those with hemp cores, but they have been found by experience not to have such a long life.

On a well-constructed line, depreciation on rope is covered by ·125 pence to ·3 pence per ton mile carried, on the basis of a line not less than 1,000 yards long, and a capacity of over 25 tons per hour; as the length and capacity increases, the depreciation per ton mile carried decreases. The fact that on a single rope system the entire length of the rope is constantly passing under observation at the terminals renders it very safe against failure, and it can be worked until a comparatively low factor of safety is reached.

Loads.

When possible, individual loads should be kept to about 1 per cent. of the hourly capacity of a line, for the purpose of distributing the weight on spans. Heavy individual loads tend to increase prime costs.

Carriers and Clips.

Among the best known are Roe's Patent Toggle and Saddle Clips.

The former, fig. 1, is capable of sustaining a load at an angle of over 45°, and in theory is the most efficient of all clips actuated by the load, as the whole of the tare weight also is used

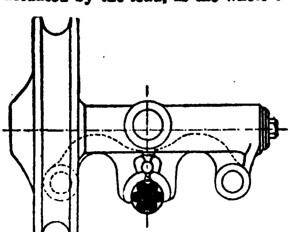


FIG. 1.

for gripping. In practice, however, the saddle clip, fig. 2, is found preferable and is usually adopted. The gripping in this type is performed by two cast chilled saddles, the sides of which

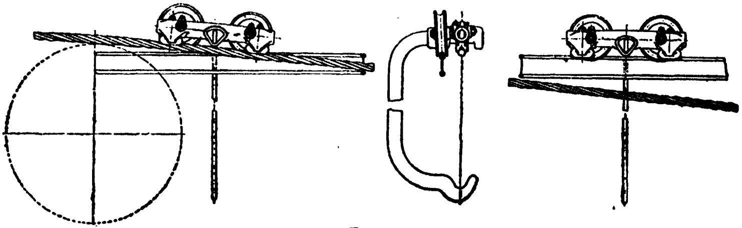


Fig. 2.

are inclined, one side having a feather projection which engages into the lay of the rope, this projection being so formed that it disengages itself without damaging the strands when the carrier enters a station. The ruling grade on which this type of clip can be safely worked is 1 in 2½. The tare weight of clip, carrier, and bucket is usually about one-third of the net load.

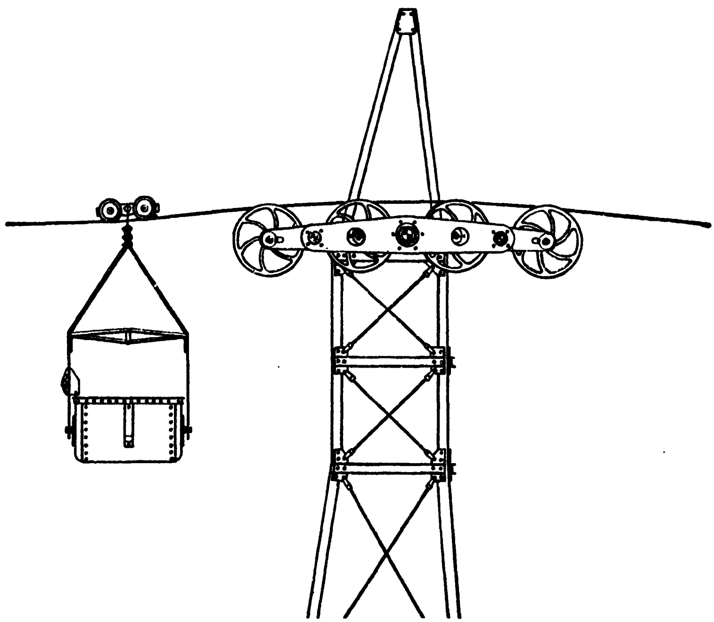


Fig. 3.

Fig. 3 shows the 'Roe Balance Beam.' The arrangement is used for distributing the pressure equally on a series of sheaves; these are arranged in groups of 2, 4, and 8. The introduction of a number of balanced sheaves in series greatly facilitates the working of long spans, and reduces wear and tear of rope where heavy pressure on any trestle has to be sustained.

A fair speed for the rope is 400 ft. per minute, though under certain conditions it can be run up to 550 ft. per minute, when no automatic return or dumping is required. The factor that rules the speed is the velocity at which the loads can be brought up to for engaging

with the rope, bearing in mind that the load and rope velocity should synchronise at the point of attachment.

Power.

Any available power can be used as long as it is steady. Trouble, however, may be caused by using intermittent power, as this gives rise to oscillation of the rope in long spans. On automatic lines, the surplus power is preferably absorbed by means of a fan or turbine. When possible, the power should be placed at the upper end of a line, and the tension at the lower terminal. In long lines that have to be sectionalised, the driving or brake station, as the case may be, is sometimes placed between two sections and the tension arrangements at the ends.

Tension

To ensure a constant and even tension, it is essential that the counterweights should be always free and floating, and owing to the amount of stretch in hemp cored ropes, taking-up gear must also be allowed for. Fig. 4 shows an arrangement which combines a multiplying and taking-up device. The two sheaves at A are attached to the tension trolley, the winch and differential drum being mounted on an independent frame placed over the tension

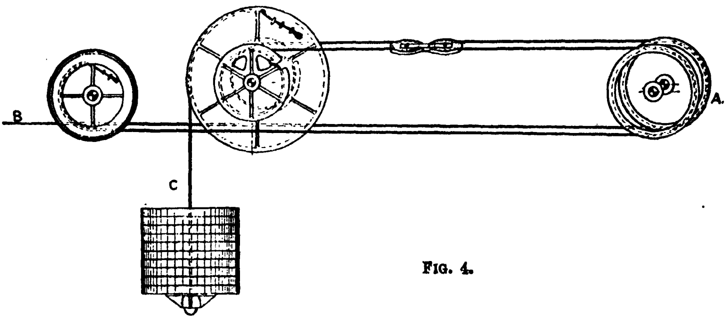


FIG. 4.

pit; one end of the rope B is anchored, the other being made fast to the tension winch drum. By this arrangement the resultant pull at A is four times the dead weight applied at C, and the weights can always be kept in suspension without the necessity of a deep pit, by taking in the slack with the winch.

Spans.

The distance between standards is dependent on the configuration of the ground, but over undulating country the average distance is approximately 330 ft. for lines of up to an hourly capacity of 60 tons. On a ropeway over the Andes, having a total length of 46 miles, several spans are being worked of 2,800 ft., though the longest recorded span on the single rope system is 3,600 ft.

DOUBLE ROPE OR BI-CABLE SYSTEM.

Carrying Ropes.

Carrying ropes may be divided into two main classes, 'spiral,' fig. 5, and 'locked coil,' fig. 6. The main advantage of the spiral type is its cheapness, and in lines transporting a small tonnag

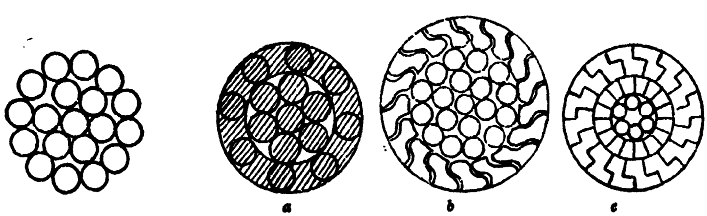


FIG. 5. FIG. 6.

per hour it will generally answer its purpose. Ropes of this type should consist of 19 wires in ropes up to 1 in. diameter and of 37 wires in ropes up to 1½ in. diameter. With this latter the

individual wires have a diameter of about ·205 in., which is about the limit to ensure perfect homogeneity of the material. Above 1¼ in. diameter, locked coil ropes should invariably be used.

Locked coil ropes, fig. 6, are of two types, the special locked, *a*, and the full locked, *b* and *c*. The former consists of a spiral core surrounded by a layer of round wires kept apart by wires drawn to a special section. This rope is sold at a medium price between plain spiral rope and full locked coil. The surface of the special locked rope is not perfectly smooth, small interstices occurring between the wires. The full locked coil, on the other hand, presents a perfectly cylindrical surface to the passage of the cars. The second type of full locked construction—shown at *c*—is particularly useful for very heavy carriers, as, owing to the rope having an inner layer of tapered section wires, the crushing and distorting effect of the load when passing over the saddles on which the rope is supported is minimised. In addition to the great advantage of offering a smooth surface to the wheels of the cars (with the result that the wires may be of a lower breaking stress than with spiral ropes), the locked coil ropes possess the great advantage that, should by any chance one of the individual wires break, it is held in place by the neighbouring wires, and does not spring from the texture of the rope, as is the case with the spiral construction. This springing of the wires is often the cause of accidents, as the cars run against the outstanding wire, and are thrown off the line.

Fixture of Ropes.

The ropes are anchored at one end of their length, and some form of tensioning device applied at the other end. This generally takes the form of a hanging weight, which may generally be from one-quarter to one-fifth of the total breaking strength of the rope. A factor of safety of 4·5 is a good medium. It has been proved that for spiral ropes a breaking stress of 95 tons per square inch, and for the locked coil ropes 65 tons per square inch, give the best results. When calculating the tension weight required, the fact must be taken into account, that should the weight be situated on a lower level than the anchorage, the weight of the rope itself acts as a tensioning factor.

For fixing the ends of the rope a satisfactory arrangement is to employ the ring wedge coupling. In this anchorage—shown in section in fig. 7—the single wires are kept apart by small nail-like

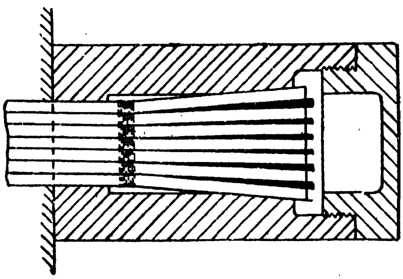


FIG. 7.

wedges, and the layers by means of ring wedges made in halves. These wedges are driven in and then a cap is screwed home, firmly locking them in place and entirely preventing any chance of the wires pulling out.

The same form of coupling, only made tapered at both ends, fig. 8, may be used for joining the various lengths of rope along the line.

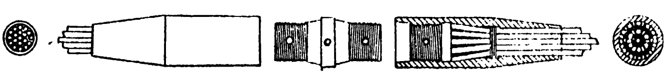


FIG. 8.

The distance apart of the tensioning weights varies according to the carrying capacity of the line, weight of carriers, number of long spans occurring, etc. A good rule is not to exceed a length of 1¼ mile between anchorage and tensioning device, although on level lines, with a small capacity, distances up to 2 miles may be taken.

Another method of tensioning is by means of buffer springs at either end of the rope, but this system is only applicable to short lines with light loads.

Diameter of Carrying Ropes.

It has been proved by experience that most trouble is caused by breaking due to bending near the saddles, and for this reason in order to obtain the diameter of the carrying rope required, many firms employ formulæ based on stresses due to bending. A formula often used is :—

$$\text{Weight per metre of ropes} = \frac{0{\cdot}85}{100}\left\{(\text{weight of carrier}) + \Big((\text{distance between carriers in metres}) \times (\text{weight of traction rope per metre}).\Big)\right\}$$

This is for spiral construction ropes. With locked coil ropes the factor 0·85 becomes 1·05. This works well for small lines up to about 50 tons capacity hourly. For each additional 5 tons, 0·05 should be added to the factor 0·85. The following table shows the various diameters of a special locked rope for various loads and traction ropes :—

(Two-Wheeled Carriages.)

Dia. of Hauling Rope.	Weight of Loads in Cwts. 3	4	5	6	8	10	16	20	24	28
In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	Ins.
$\frac{1}{2}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1	$1\frac{1}{8}$	$1\frac{3}{16}$	—	—	—	—
$\frac{9}{16}$	$\frac{13}{16}$	$\frac{15}{16}$	$\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{1}{4}$	—	—	—	—
$\frac{5}{8}$	$\frac{7}{8}$	$\frac{15}{16}$	1	$1\frac{1}{16}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{1}{2}$	—	—	—
$\frac{11}{16}$	$\frac{15}{16}$	$\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{1}{16}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{9}{16}$	—	—	—
$\frac{3}{4}$	$\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{5}{16}$	$1\frac{9}{16}$	$1\frac{11}{16}$	$1\frac{13}{16}$	—
$\frac{13}{16}$	—	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{9}{16}$	$1\frac{3}{4}$	$1\frac{7}{8}$	2
$\frac{7}{8}$	—	—	$1\frac{3}{16}$	$1\frac{3}{16}$	$1\frac{5}{16}$	$1\frac{3}{8}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{7}{8}$	2

Design of Ropeway.

To lay out the profile for a ropeway, when the line is laid out on paper, the curve of the rope may be obtained and set out from the formula—

$$\text{Dip} = L^2W/8T,$$

where,

L = span; W = weight per unit length; 8 = constant; T = tension.

and then the supporting trestles placed accordingly. It sometimes happens, however, in practice that when the rope is fully tensioned it 'floats' or rises up from the supporting tower, owing perhaps to a minute error in the drawn profile. To avoid this, some Continental makers employ a so-called factor of safety for the dip—that is, the denominator 8T is multiplied by a constant —the formula giving, as a result, a dip smaller than the actual. It will be obvious that if the curve with the dip so obtained is set out and the trestles drawn up to it, they will be slightly higher than need actually be the case, thus avoiding any possibility of 'floating' when the line is erected. This factor of safety may be 1·5 for spiral ropes and 1·1 for locked coil ropes.

There has been of late years an increasing tendency to employ saddles (upon which the rope is supported at the trestles) which are pivoted at the centre, and can consequently adapt themselves to the varying gradients of the rope, and there is little doubt that this type when properly designed, and well greased to allow of play in the rope, gives full satisfaction. In designing the shoes a good medium value for the pressure of the rope is 250 lbs. pressure per square inch of projected area of the rope. This is for locked coil ropes; for spiral ropes a little less should be taken.

It sometimes happens that at a trestle, on either side of which are long free spans, there is a heavy downward reaction of the hauling rope, which, when the carrier passes over the saddle, produces a heavier pressure on the rope than that for which it was calculated. In this case the rope is protected by means of a steel plate cap over which the carrier runs, thereby removing the pressure from the rope. There is a difference of opinion as to when these caps should be employed, but it may safely be said that their use is to be recommended where the hauling rope reaction exceeds 450 lbs.

Hauling Ropes.

The *traction* or *hauling rope*, by means of which the carriers are propelled, is invariably a stranded rope with a hemp core, the flexibility depending on the size of the individual wires in the strands and the breaking stress of the material. This rope is endless and in constant motion. In the general case it passes round a driving sheave at one end of the line and round a surging or tension sheave at the other end. In some cases this plan may be varied, as, for instance, when the carriers are required to pass automatically round a large sheave at one end

station. In this case no manual attention is required at this station and the driving and tensioning sheaves are combined in the station at the opposite end of the line. Whenever possible the driving should take place at the upper and the tensioning at the lower end of the line. In this way the accumulative tension in the rope due to its own weight is utilised as a tensioning agent, whereas, were the tensioning to take place at the upper end, an additional amount of weight would be required to counterbalance the weight of the rope. The amount of the tensioning weight should be such as to give sufficient grip round the driving sheave to ensure perfect driving when the full load is on the line.

Diameter of Hauling Ropes.

The diameter of the hauling rope depends upon the highest tension occurring in it

$$\text{Tension of rope in lbs.} = \left\{\text{Initial tension} + \left(\frac{W}{S} \times H\right) + (w \times H)\right)\right\} + \left\{(W \times \mu_1 \times N) + (L \times \mu_2 \times w) + (R \times 2)\right\},$$

where,

W = weight of full carrier; S = spacing of carriers in feet; H = difference in level between stations in feet; w = weight per foot of traction rope; μ_1 = coefficient of friction of cars; μ_2 = coefficient of friction of traction rope; L = length of line in feet; N = number of carriers on one side of line; R = number of supporting rollers on one side of line.

This tension if multiplied by the factor of safety—6 to 8—and divided by the breaking stress, gives the area of the rope necessary. This is for the general case with the drive at the upper terminal and the loads ascending. When the loads are descending, the plus sign in the centre of the formula becomes minus. To obtain the tension when the ropeway is just put into work, the second half of the formula should be doubled.

The usual breaking stresses of traction ropes are from 80 to 115 tons per square inch. The surge of the hauling rope tension sheave depends upon the distance apart of the carriers, whether long spans occur in the line, etc. As an average, 6 ft. play may be allowed for to commence with, and then an additional 12 ft. for each mile of length. In some cases in long lines it is necessary to split up the line into sections to avoid excessive tensions occurring.

Driving Power.

$$\text{Power in ft. lbs.} = \left\{\left(W_1 \times W_2 \times \mu_1 \times \frac{N}{2} \times V\right) + (w \times 2L \times \mu_2 \times V) + (2R \times V) + (F)\right\} + \text{ or } - \text{ theoretical power,}$$

where,

W_1 = weight of full carrier; W_2 = weight of empty carrier; μ_1 = coefficient of friction for cars; N = total number of cars on line; V = velocity in feet per minute of cars; w = weight per foot of traction rope; L = length of line in feet; μ_2 = coefficient of friction for traction rope; R = number of supporting pulleys; F = power required for station friction.

The first four factors give the power necessary to overcome friction, and the theoretical power required or developed depends upon whether the loads are ascending or descending. The factor for station friction varies greatly for different lines, and is solely a matter of experience, although 0·02 of the total weight of the moving parts on the stations multiplied by the speed may be taken as a good mean.

Carriers.

There are two kinds of carriers in general use, those employing screw grips and those with lever grips. These may again be divided into sections, the overtype and the undertype grips. In the former, the hauling rope is attached to the carrier or running head at a point above, and in the latter at a point below the carrying rope.

The main advantage of the overtype grip is that the length of the stations is shorter than with the undertype grip, as the hauling rope may be led in and out of the jaws without interfering with the shunt rails. In the case of lever grips the overtype is generally the cheaper, and on lines containing no steep gradients can often be employed with advantage. Further, it may be more easily adapted for taking curves in both directions than the undertype. The overtype grip has, however, several great disadvantages. Chief amongst these is the fact that when ascending steep gradients the traction rope reaction falls outside the wheel base, tending to tip the whole running head, a tendency which may not be counterbalanced by the weight of the carrier, with the consequence that the whole apparatus leaves the line. Again, owing to the fact that the centre line of the hauling rope must fall outside the centre line of the carrying rope to avoid any entangling, there is a decided tendency for the carrier to tip sideways as soon as a heavy hauling rope reaction occurs, a tendency which is very detrimental to the perfect working of the line when it is required to discharge the contents of the carrier automatically whilst the carriage is on the open track. These defects may be remedied by having the hauling rope directly below the carrying rope, *i.e.*, using an undertype grip. Practically the only defect

of this type is that the stations are generally longer than with the overtype grip, thus involving extra structural work. Even this may be overcome by employing an undertype grip in which the jaws open downwards instead of upwards, as is usual.

The speed at which the carriers should be run depends upon whether it is necessary for them to be hauled automatically round curve or return sheaves by the traction rope. The following figures represent good practice :—

Straight lines	400–450 ft. p.m.
Curves up to about 45° change of direction	Undertype up to 400 ft. p.m. Overtype „ 300 ft. p.m.
With automatic haul-round—	
Sheave, 20 ft. dia.	Undertype, 350 ft. p.m. Overtype, 300 ft. p.m.
Sheave, 16 ft. dia.	Undertype, 300 ft. p.m. Overtype, 250 ft. p.m.
Sheave, 12 ft. dia.	Undertype, 200 ft. p.m. Overtype, 150 ft. p.m.

The running wheels should be of high quality cast steel, the diameter depending on the load and type of rope, larger diameter wheels being employed with spiral ropes than with locked coil. The runner pins should be self-oiling. The following table gives some diameters of wheels for varying loads, carriers with two wheels being employed :—

Weight of load and carrier, cwts.	Diameter of wheel, ins.
14	8
30	10
40	12

Screw Grips.

There are two distinct main classes of screw grip, in the first of which the grip is operated by a lever and strikers in the stations. In the second the grip is operated by the weight of the carrier and load. The best example of the former is shown in fig. 9. There are two jaws—(*a*) the movable and (*b*) the fixed. The movable jaw is actuated by a shaft carrying two screws, one a rapid thread and the other a fine thread. This shaft is turned by the lever C. The action of the grip is as follows :—On the lever C being turned over, the rapid thread operates and the movable jaw quickly approaches the fixed jaw until the rope is just gripped. The fine thread now comes into action and applies the necessary force to grip the rope firmly. To unlock the

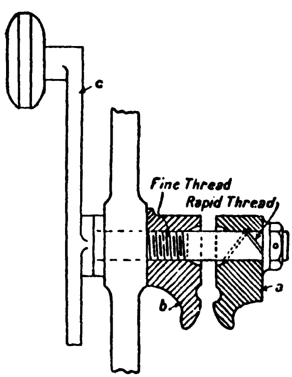


FIG. 9.

grip the reverse action takes place. The lever C is actuated at the stations by strikers constructed of angle irons.

In the other form of screw grip, in which the pressure is obtained from the weight of the carrier and load, the action is the same, the necessary turning moment being applied to the screw by hanging the carrier at the end of the lever arm. The grip is opened to allow of the rope entering or leaving, by means of rails, generally formed of angle irons, which depress small pulleys fixed at the end of the lever arm opposite to that on which the carrier is hung, thus raising the latter and thereby opening the jaws. It has been found in practice that the position of this lever arm should not be less than 35° to 40° to the vertical, to allow of easy depression of the rollers.

Lever Grips.

One of the best known Continental types is that shown in fig. 10. In this grip the pressure is applied by the weight of the carrier and load acting through a simple system of levers to give a multiplied pressure on the jaws. The grip is released by lifting the load, which is accomplished by means of small rollers running on to rails in the stations. Another type of grip—the

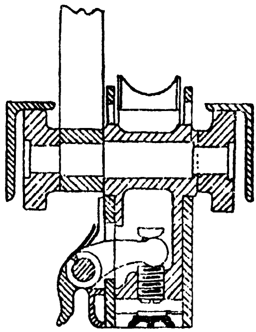


FIG. 10.

inclined plane type—is illustrated in fig. 11. The carrier and load are attached to a hanger having at its upper end a curved slot. This slot engages with a circular pin attached to the movable jaw. The downward pull of the hanger draws in this pin and with it the movable jaw, thus closing it upon the rope. When the pin is in the upper part of the slot the grip is absolutely locked, as no amount of upward pull of the rope can force the jaws apart. The grip is released

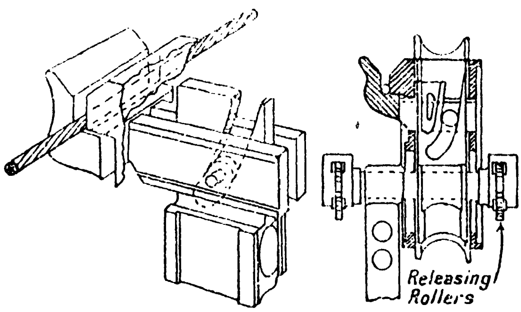


FIG. 11.

in the stations in the same manner as the type shown in fig. 10. In practice this form of grip has been found extremely simple and efficient.

In the matter of taking curves automatically in both directions, the advantage lies with the lever type of grip. In cost the latter generally has a great advantage. A ropeway, with lever grips, carrying two-ton loads on an incline of 86 per cent., is being used for carrying timber in South-West Africa.

FIXED CLIP SINGLE ROPE SYSTEM.

This type of ropeway is only applicable for light capacities, and is used largely on plantations and for the transport of valuable ores, where comparatively small quantities are handled.

The carrier head, fig. 12, is permanently attached to the rope by means of a spring steel strap, and is designed in such a way that it can pass freely over trestle sheaves, under depression wheels and round the terminals, without being detached from the rope. As both rope and carriers are in constant motion, loading has to be done by hand, or by means of a travelling hopper, which is usually hung from overhead rails at the terminal. The terminals are extremely simple and

compact, as all shunt rails are done away with. The tension terminal is usually mounted on a trolley or on skids, the whole being movable by means of tension gear, to allow for stretch of rope, etc. Angles can be worked with the fixed clip system, but they should be preferably avoided, as they tend to produce excessive rope and gear wear at such places.

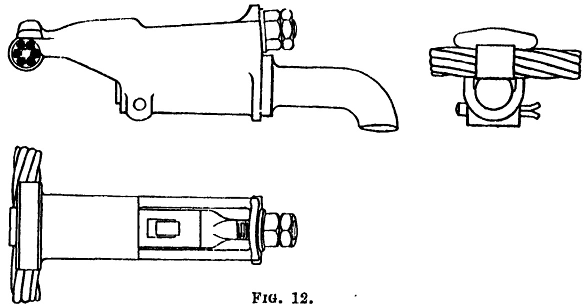


FIG. 12.

This type of line is run at a slow speed, namely, about 200 ft. per minute, and the only advantages it can claim are its simplicity, adaptability to steep grades, and its low prime cost when light capacities and moderate lengths have to be dealt with.

JIG-BACK OR TO-AND-FRO.

This type of ropeway is useful for short distances, and is operated with a pair of carriers that travel in reverse directions and are loaded or brought to rest alternatively at the opposite stations, but which do not pass round the terminals; it is, therefore, intermittent in its action, and the direction of travel is reversed after each journey. Where loads are to be carried which are heavy individually in relation to the capacity, and where the distance is short, the 'To-and-Fro System' is particularly applicable. A high carrier and rope velocity may be adopted during the run, especially where a clear span occurs between terminals, and where there are no intermediate trestles, These installations may be on the 'Monocable' or 'Bi-cable' principle, and in either case the moving rope is clamped to the carriers, and is not usually disconnected from them at the terminals.

It should be mentioned that, up to the present, the 'To-and-Fro' ropeway is the only system that is used for regular passenger traffic, as the carriers lend themselves to the adoption of a safety grip used in conjunction with an auxiliary standing rope, but with such appliances these installations are very costly and present mechanical difficulties on lines of any length.

The following table, giving a rough idea of the capabilities of the various systems mentioned, shows the maximum lengths, capacities, and loads carried on various lines at present working or in course of construction:

System.	Present Maximum Lengths.	Present Maximum Hourly Capacity.	Present Maximum Individual Load.
	Yards.	Tons.	Tons.
Monocable .	80,196	200	$1\frac{1}{2}$
Bi-cable .	38,800	250	2
Fixed Clip .	2,330	8	4 cwts.
To-and-Fro .	1,850	Passengers' weight unknown.	17 tons.

LIMITATIONS OF TYPES OF AERIAL ROPEWAYS.

The *jig-back* type is the cheapest for short distances and a capacity of 8 to 12 tons per hour. Where the distance is considerable the loss of time in making the double journey between each load is too great, and where individual loads are heavy and the speed high too much time and energy is lost in starting and stopping.

The *monocable* type is suitable for a capacity up to 80 or 100 tons per hour where the grade of the rope is not steeper than 1 in $2\frac{1}{2}$ or 1 in $2\frac{1}{4}$. Where the rope grade would be steeper than this the boxhead clips would be liable to slip on the rope, and a bi-cable would have to be considered.

The *bi-cable* type is required for steep grades and large hourly capacity. It is evident from general principles that if the number and weight of the individual loads carried by a rope be gradually increased the tension and weight of the rope will have to be increased also, and at last

a point will be reached when it is cheaper to fix the heavy main rope and use it for carrying purposes only and to provide an extra and lighter rope for hauling the loads. This point marks the natural transition from monocable to bi-cable.

CONVEYORS.

Belt Conveyors.*

For conveying materials continuously over long distances in a horizontal or inclined direction band conveyors are usually employed.

This type of conveyor consists of an endless belt running over terminal pulleys and supported at intervals both on the loaded and the return strands by intermediate idler rollers and with suitable arrangements to maintain the belt at its correct tension.

The belts are sometimes made to run flat, but more generally they are troughed on the loaded side, the effect of the troughing being to greatly increase the capacity.

The material is discharged over the head pulley or it may be discharged at any point throughout the length of the conveyor by means of throw-off carriages, either fixed or travelling, and in the case of flat belts the material may be thrown off by means of travelling or fixed ploughs.

Design Data.

Before designing any belt conveyor the necessary basic data consists of a complete knowledge of the nature of the material, the quantity to be handled, and the relative positions of the points from and to which the material is to be conveyed. The source of power available, space, etc., must also be known. A narrow belt at high speed, or a wide belt at a slower speed may be selected, either of which will give the required capacity. The initial cost of the belt, idlers, and driving unit will usually favour the narrow belt running at high speed, but there are limits, both as to speed and width, precluding the general use of high-speed narrow belts.

Factors governing the size and speed of belt are as follows: the influence of lump size on belt width; the influence of belt tension on belt width; the influence of belt speed on loading; the influence of belt speed on discharge; and the influence of belt speed on the belt life.

Capacity of Belt Conveyors.

Knowing the weight per cu. ft. and size and percentage of lumps in the material to be handled, the width and speed of belt to give any desired capacity may be calculated. It has been found from experience that the most satisfactory loading for troughed belts is obtained when the cross sectional area of the load on the belt is equal to $0 \cdot 084 \times (\text{width of belt in inches})^2$, *i.e.* $0 \cdot 084\ W^2$ = sq. ins. Assuming the belt be loaded to a cross sectional area equal to the above, the capacity in cu. ft. per hour carried at a belt speed of 100 ft. per min., will be equal to

$$0 \cdot 084 \frac{(\text{width of belt in inches})^2}{144} \times 100 \times 60 = 3 \cdot 5\ W^2 \text{ for troughed belts.}$$

For flat belts the capacity may be taken as $1 \cdot 5\ W^2$ in cu. ft. per hour at a belt speed of 100 ft. per min.

The capacity in tons per hour for any material weighing M lbs. per cu. ft. may be calculated for troughed and flat belts as follows:—

Capacity of troughed belt. $T = \dfrac{3 \cdot 5\ W^2 \times M \times S}{2{,}240 \times 100}$

Capacity of flat belt. $T = \dfrac{1 \cdot 5\ W^2 \times M \times S}{2{,}240 \times 100}$

where T = Capacity in tons per hour.
W = Width of belt in inches.
S = Speed of belt in feet per minute.
M = Weight of material in lbs. per cubic foot.

Table I gives the carrying capacity for troughed belt conveyors of various belt widths at a belt speed of 100 ft./min., calculated from the formula given. To find the width and speed of belt for a troughed belt conveyor to handle 400 tons per hour of unsized coal containing lumps of 12 in. maximum size: select from the table the minimum size of belt that will carry the maximum size of lumps. This will be found to be 30 in. for unsized material.

From the table also it will be seen that this size of belt will have a capacity of 70 tons per hour for each 100 ft. per min. of the belt speed. The speed of the belt to give the required output would therefore be $\dfrac{400 \times 100}{70} = 570$ ft. per min.

Although this speed comes within the safe maximum speed given in Table I, it is considered too high for the class of material being handled. The next size of belt available is 36 in. wide

* For much of this information we are indebted to the New Conveyor Co. Ltd.

TABLE I.

CAPACITY CHART FOR TROUGHED BELT CONVEYORS.

Size of Belt.	Maximum Size of Lumps.		Max. Recommended Belt Speed.	Capacity in Cu. Ft./Hour.		Capacity in Tons per Hour.							
						Material at 50 lb./cu. ft.		Material at 75 lb./cu. ft.		Material at 100 lb./cu. ft.		Material at 125 lb./cu. ft.	
	Sized.	Unsized.		Belt Speed of 100 Ft./Min.	Max. Belt Speed.	Belt Speed of 100 Ft./Min.	Max. Belt Speed.	Belt Speed of 100 Ft./Min.	Max. Belt Speed.	Belt Speed of 100 Ft./Min.	Max. Belt Speed.	Belt Speed of 100 Ft./Min.	Max. Belt Speed.
Ins.	Ins.	Ins.	Ft./Min.										
12	2	2	350	505	1,765	11	40	17	60	22	78	28	98
14	2	3	350	686	2,400	15	53	22	80	30	105	37	132
16	2½	4	350	900	3,140	20	70	30	105	40	140	50	175
18	3	5	450	1,130	5,100	25	113	37	170	50	228	63	285
20	3½	6	450	1,400	6,300	31	140	46	210	62	280	78	350
22	4	7	450	1,700	7,620	38	170	57	255	76	340	95	425
24	4½	8	550	2,016	11,100	45	250	67	370	90	495	112	620
28	6	10	550	2,750	15,100	60	340	90	500	120	672	150	840
30	7	12	650	3,150	20,500	70	460	105	685	140	915	175	1,140
36	8	15	650	4,500	29,500	100	660	150	990	200	1,320	250	1,650
42	10	18	750	6,200	46,300	135	1,035	205	1,550	275	2,070	340	2,600
48	12	20	750	8,100	60,500	180	1,350	270	2,025	360	2,700	450	3,375

and to handle 400 tons per hour would have to run at $\frac{400 \times 100}{100} = 400$ ft. per min. This is considered a reasonable speed, and could safely be adopted. If, however, the coal being handled should be of a particularly friable nature, and it is important for the installation that a minimum of breakage should be obtained, it would be necessary to consider the use of a 42-in. belt running at $\frac{400 \times 100}{135} = 295$ ft. per min.

The h.p. required to drive a conveyor will depend on capacity, size, speed of belt, length, efficiency of driving gear, and type of idler rollers used. If the conveyor is inclined the h.p. will also depend on the height to which the material has to be elevated.

The length of a conveyor has a big influence on the h.p. irrespective of the load carried, because a percentage of the total power is absorbed by friction in the head and tail pulleys. A table of constants for various lengths of conveyors is given in Table II below.

TABLE II.

Speed Factors.		Length Factors.		Length Factors.	
Width of Belt. W (Ins.).	Speed Factor. C.	Length of Conveyor. L (Ft.).	Length Factor. F.	Length of Conveyor. L (Ft.).	Length Factor. F.
12	0·0133	20	2	150	1·133
14	0·0206	25	1·8	200	1·1
16	0·0277	30	1·67	250	1·08
18	0·0350	35	1·57	300	1·067
20	0·0423	40	1·5	350	1·057
22	0·0495	45	1·445	400	1·05
24	0·0570	50	1·4	500	1·04
28	0·0713	60	1·33	600	1·033
30	0·0785	70	1·286	700	1·029
36	0·1000	80	1·25	800	1·025
42	0·1220	90	1·22	900	1·022
48	0·1440	100	1·2	1,000	1·02

These values are equal to $1 + \frac{20}{\text{length of conveyor in feet}}$ and represent the length factor F in the belt conveyor h.p. formula given below.

The Influence of Speed of Belt on H.P.

The h.p. is also influenced by the speed of the belt. This is due to the friction of the idler bearings which varies in proportion to speed and type of bearings. A Table of Constants is given in Table II.

These values have been found by experience, and represent the speed factor C in the h.p. formula. The h.p. of a horizontal belt conveyor may be calculated by the following formula:

$$\text{H.P.} = \frac{(CS + DT)\ L\ F\ B}{1{,}000\ G} + 10 \text{ per cent. for contingencies}$$

where C = Speed factor (given in Table II).
D = Load factor = 0·0757.
L = Length of conveyor in feet.
F = Length factor = $1 + \frac{20}{\text{length of conveyor in feet}}$ or from table.
B = Idler co-efficient = 0·7 for ball bearings.
= 1 for journal bearings.
G = Efficiency of driving gear.
S = Speed of belt in feet per minute.
T = Capacity in tons per hour.
H = Height of lift in feet.

If the conveyor is inclined the power required to elevate the material to the desired height must be added to the power that would be necessary if the conveyor were horizontal instead of inclined.

The h.p. for an inclined belt conveyor may, therefore, be calculated by the following formula:—

$$\text{H.P.} = \left[\frac{(CS + DT) \times LB}{1{,}000} + \frac{HT}{884}\right] \frac{F}{G} + 10 \text{ per cent. for contingencies.}$$

Angle of Inclination.

The maximum gradient on which material can be carried on a belt without rolling or slipping back is governed by the shape, size, and assortment of the material, the speed, the feed on to the belt, and whether loading is continuous, or intermittent. Definite figures cannot be given to meet all requirements, but the figures given in Table III may be taken as a rough guide. The figures given are for a continuous feed and a well filled belt.

TABLE III.—MAXIMUM RECOMMENDED ANGLE OF INCLINE OF BELT CONVEYORS FOR VARIOUS MATERIALS.

Material.	Angle of Gradient in Degrees.	Material.	Angle of Gradient in Degrees.
COAL.		SAND.	
Screened, nuts, peas	17 to 18	Dry	15
Fine sized	18	Damp	22 to 24
Run of mine	18	Foundry moulding	24
Slack, fine with lumps	20 to 22		
Damp slack	22	STONE.	
Lumps	22	Limestone and ground clinker	18
		Road metal	16
		Iron ore and crushed stone	18
COKE.			
Oven run. Large gasworks	18	GRAIN	18 to 23
Small breeze	18 to 20		
Crushed	22	PACKAGES.	
		Paper wrapped with flat surfaces on rubber cover belt	16
GRAVEL.		WOOD CHIPS	27
With plenty of sand	18 to 20		
Screened	15 to 18	*Note.*—Definite figures for various materials cannot be given as the nature of the material and conditions are so varied, but the above may be taken as a guide.	
PEBBLES.			
Or similar round material	12		

Belt Tension.

In order that the driving pulley may obtain a grip on the belt sufficient to drive the conveyor, a certain amount of initial tension must be applied to the belt by means of a tensioning gear. To drive a conveyor additional tension must be applied to the carrying side of the belt through the medium of the driving pulley. This additional tension may be termed the effective tension or h.p. pull representing the actual force required to move the load.

With the conveyor in operation, the tension in the carrying or tight side of the belt is the tension T_1, which is equal to the initial tension T_2, plus the effective tension of h.p. pull, while the tension in the return or slack side of the belt is equal to the initial tension T_2. The effective tension is, therefore, equal to $(T_1 - T_2)$, and is dependent upon the maximum tension of the belt, the angle of belt wrap and the co-efficient of friction between the belt and the pulley. Ratio factors for $\frac{T_1}{T_1 - T_2}$ for various type of drives are given in Table IV below.

TABLE IV.

Type of Drive.	Angle of Belt Wrap in Degrees.	Ratio of Belt Tension. Ratio $\frac{T_1}{T_1 - T_2}$. Bare Pulley.	Ratio of Belt Tension. Ratio $\frac{T_1}{T_1 - T_2}$. Lagged Pulley.
Open	180	1·84	1·5
Snub	210	1·67	1·38
Tandem	420	1·19	1·08

Number of Plies required for Belt.

If the power width and speed of belt is known, the number of plies required for the belt may be calculated as follows :—

$$N = \frac{H.P. \times 33{,}000 \times R}{S \times W \times fs}$$

where N = Number of plies of belt.
$H.P.$ = H.P. to drive conveyor.
R = Ratio of $\frac{T_1}{T_1 - T_2}$ see table above.
S = Speed of belt in feet per minute.
W = Width of belt in inches.
fs = Safe stress allowable per ply per inch of belt, according to quality of duck.

This equals :

25 lbs. per inch of ply for 28 oz. duck	British Standard
27 " " " " 32 " "	Specification
30 " " " " 36 " "	No. 490.

The Belt and Cover.

Conveyor belts are constructed to fulfil two requirements, namely strength to transmit the drive, and durability against abrasion due to the material, and also the friction of the driving pulleys. Strength is governed by the number of plies or layers, and the weight of the duck used. In order that the belt may trough properly, the number of plies must be kept within certain limits, which vary with the width of belt.

In order to give the belt more durability against wear due to the friction of the driving pulley and against the wear caused by the friction and abrasion due to the material carried, the belt is protected by a covering of rubber. This covering is $\frac{1}{32}$ in. thick minimum in the driving side, but varies in thickness on the carrying side to suit the nature of the material to be carried, and belts are available with the following thickness of top cover—$\frac{1}{32}$ in., $\frac{1}{16}$ in., $\frac{3}{32}$ in., $\frac{1}{8}$ in. and $\frac{3}{16}$ in. thick, all of which may be obtained in either Grade 'A,' 'B' or 'C' rubber in accordance with British Standard Specification No. 490.

Special heat-resisting conveyor belts are available for carrying hot materials, such as coke, cement clinkers and hot ash, the temperature, however, should not exceed 230° F.

Selection of Pulley Size.

The diameter of the terminal pulleys for good service should be as large as possible. General practice is to make the diameter in inches of all driving pulleys five times the number of plies, and tail pulleys four times the number of plies. The diameter in inches for snub and bend pulleys should be three times the number of plies. It is recommended that the faces of the pulley should also be at least 2 in. wider than the belt for belts up to 24 in. and 3 in. to 4 in. wider for larger belts. In order to increase the tractive effort of the driving pulley on the belt, the pulley may be covered or lagged with rubber. The driving pulley for the belt should be arranged wherever possible at the delivery end of the conveyor.

Several methods are available for obtaining the maximum driving efficiency from the driving pulley. In short and medium length conveyors, the drive pulleys get sufficient traction by using a plain open drive. Longer conveyors require greater effective belt tension which may be obtained by increasing the initial or slack side tension ; lagging the plain driving pulley with a covering of rubber or other suitable material ; using a snub pulley to increase the arc of driving contact ; or by using a tandem or two-pulley drive which increases the arc of contact still further.

Loading the Belt.

In order to ensure efficient working of any belt conveyor it is essential that the belt be uniformly loaded and that a minimum of wear is caused to the belt by abrasion. This is a feature which is governed by many factors, and the design of feed chutes and the provision of feeders will depend on the class of the material to be handled, and also the conditions of service required. The design and slope of the chute is of the greatest importance in order that the speed or flow of the material be given a velocity approximate to that of the belt, thereby reducing the abrasion and friction on the belt cover.

SECTION XXXIX

PART I

MINING (pp. 847–939)

(Revised by Sir Richard A. S. Redmayne, K.C.B., M.Sc., M.I.C.E., M.I.M.M., F.G.S., etc., and W. H. Evans, M.Sc. (Eng.), Ph.D., A.M.I.Mech.E.)

PART II

MINERAL VALUATION (pp. 941–945)

(Revised by Sir Richard A. S. Redmayne, K.C.B., M.Sc., M.I.C.E., M.I.M.M., F.G.S., etc.)

SECTION XXXIX

PART I

MINING

(Revised and brought up to date by Sir Richard A. S. Redmayne, K.C.B., M.Sc., M.I.C.E., M.I.M.M., F.G.S., etc., and W. H. Evans, M.Sc. (Eng.), Ph.D., A.M.I.Mech.E.)

LEGAL—CLASSIFICATION OF MINERAL DEPOSITS—COAL—METALLIFEROUS MINERALS — BORING — BLASTING — DRIFTS — SHAFTS — COAL CUTTERS — SHAFT PILLARS—MINING IN VEINS AND MASSES—DRILLING—HAULAGE—WINDING — WIRE ROPES — PUMPS AND PUMPING — VENTILATION—SAFETY LAMPS—RESCUE APPARATUS—DRESSING OF ORES AND COAL CLEANING — GOLD MILLING—ALLUVIAL MINING.

Legal.

DEFINITIONS OF MINERALS AND MINES.

By the term 'mineral,' as used in legal documents, is meant a constituent of the earth's crust which possesses in its severed condition an intrinsic value by virtue of its composition, that is to say, 'any substance that can be got from within the surface of the earth which possesses a value in use apart from its mere possession of bulk and weight' (Lord Fletcher Moulton, Court of Appeal, Nov. 23, 1908). The meaning of the word 'mineral' may be affected by the context; according to Lord Halsbury, in 'a grant of *mines* and *minerals*,' it 'is a question of fact what these words meant in the vernacular of the mining world, the commercial world and landowners at the time when they were used in the instrument' (Lord Provost of Glasgow *v.* Fairie, 13 App. Cas. 1888).

In this country a *mine* is defined as a working for mineral carried on by artificial light without intentionally disturbing the surface, whilst a *quarry* is an excavation for the extraction of mineral open to the day; the distinction is important from a legal point of view. It must be noted that a quarry less than 20 feet deep is by law classed as a factory. In many other countries, notably in the United States of America, the distinction between a 'mine' and a 'quarry' depends not upon the method of working but upon the mineral wrought, any working for e.g. building stone, whether underground or open, being classed as a quarry. In legal instruments 'mine' sometimes has the wider meaning of the space containing the minerals, or even from which the minerals have been extracted.

Ownership of Minerals.

The tenant in fee simple or 'owner' of land is deemed to own everything over, on, or under the surface down to the centre of the earth between verticals drawn upwards and downwards from the boundaries of his surface, and until recently, the only exceptions that existed to this *prima facie* right (apart from special custom) were those of gold and silver which are excepted in favour of the Crown. Now, however, since July 1, 1942, all coal, with certain associated minerals (*e.g.* interstratified ironstones and fire clays) has passed from private ownership, and under the provisions of the Coal Act of 1938 become the property of the State; a Coal Commission having been appointed by Government to exercise the functions of owners of the fee simple in coal and mines

of coal, but they shall not themselves engage in the business of coal mining or carry on any operations for coal-mining purposes, other than searching for and boring for coal, but shall grant leases for those purposes.

The department of State under which the Coal Commission exercises authority is the Board of Trade.

Compensation awarded to late owners of Coal and Associated Minerals.—A 'global' sum of £66,450,000 was decided upon as the amount of the compensation to be awarded by Government to the owners of coal in Great Britain, which sum was divided by the Central Valuation Board so as to be allocated as follows:—

The Scottish Valuation Region	£8,512,245
„ Northern „ „	13,515,930
„ Western „ „	5,409,030
„ Yorkshire „ „	10,306,395
„ North Midland Valuation Region	6,645,000
„ „ Staffordshire „ „	1,315,710
„ East Midland Valuation Region	1,475,190
„ West „ „ „	1,628,025
„ South Wales „ „	16,446,375
„ Southern „ „	1,196,100
	£66,450,000

The work of valuing the various holdings was entrusted to bodies of experts constituting the several region valuation boards named as above.

The '*valuation*' *date* was the first day of January, 1939, and the work of valuation had to be completed by July 1, 1942, the *acquistion date*, but as a matter of fact, owing to the intervention of the war, was not concluded until a year later.

As it now falls to the Coal Commissioners to exercise the functions previously possessed by the agents of many coal owners, it is to be presumed that not only will the colliery leases be simplified in structure but will tend more or less to follow a standardized form. Certainly the number of leases will be greatly reduced.

Certain requirements which existed in the past will, however, continue to operate in the future, some of these may be briefly referred to.

Certain Rent.—The right to work the coal is conceded in return for what is termed a *certain fixed* or *dead* rent of so much per annum, which merges into the *royalty* rent. The amount of the certain rent being commonly determined by the capability of working an agreed minimum number of tons of coal per annum at an agreed tonnage rate. The rent usually averages in normal cases from £1 to £1 15*s*. per acre of surface. Of course, the thickness of the seam worked is an important factor in the case.

'*Overs*' *and* '*Shorts*.'—The full amount of the certain rent is payable whatever the quantity of coal worked in any given year may be. When in any year the number of tons worked at this rent exceeds in value the certain rent, the surplus is paid as '*over workings*,' that is, chargable with *royalty rent*; when it falls short, the deficiency is carried forward as '*short workings*' to next year's account. Power is usually—nearly always—granted to the lessee to make up 'shorts' —that is, set them against subsequent 'overs'—within, it may be, a given period of years.

Royalty Rent.—The royalty in which the certain rent 'merges' is usually payable:—

(*a*) as a fixed sum per ton raised;
(*b*) as a fixed sum per foot per acre worked;
(*c*) as a fixed proportion of the value of the mineral raised;
(*d*) by way of a sliding scale;
(*e*) as a varying proportion of the value of the mineral sold, regulated as a sliding scale.

It is usual to make a final deduction from the saleable output of coal for coal used for *colliery consumption*. The amount varied in regard to past leases, but usually was 4 per cent.

The average of the 'royalties' inclusive of '*way-leaves*' payable per ton of coal raised over the United Kingdom, by whatever method payable, was estimated by the Royal Commission on Mining Royalties in 1889 to be 5·44*d*. per ton of coal worked inclusive of way-leaves, and as the way-leaves may be taken as averaging about ¼*d*. per ton, the average of the actual royalty rent would be 5¼*d*. per ton.

Way-leaves.—Way-leaves are divisible into:

(*a*) surface; (*b*) underground,

and imports the leave or permission of the owner of the property through or over which passage is required. The expression is applied usually, if not exclusively, to mining roads or ways, or shafts (*shaft rent*) for the passage of minerals, or for other mining purposes.

It is to be presumed that in so far as underground way-leaves and shaft rents (in the few cases where the latter have been charged by former lessors) now that the ownership of the coal is vested in a single lessor—the State—that these will cease to be chargeable against the lessee.

Coal Contents of a Seam of Coal.—In estimating the coal contents of a given area of a seam it is customary to take an acre as containing 1,510 tons per foot of thickness, and making deduction therefrom for loss of coal in working, the occurrence of faults and other geological disturbances such as thinning of the seam, 'nip' or 'wash outs,' and so forth.

The National Coal Board.

The National Coal Board was constituted by the Minister of Fuel and Power (to whom it is responsible) in July 15, 1946, under the Coal Industry Nationalisation Act, 1946. Its duties were :—

(*a*) to work and get coal in Great Britain ;
(*b*) to secure the efficient development of the coal mining industry ;
(*c*) to supply coal without showing 'undue preference' to anyone, and to regulate qualities and sizes, quantities and prices so as best to further the public interest in all respects.

Its policy had also to promote health, safety and welfare of all employees, and to utilise their practical knowledge. Subject to certain directions from the Minister, the Board was solely responsible for managing the industry and running it on sound business lines.

The Board comprised a Chairman and a Deputy Chairman, and seven other members, some of whom were charged with responsibility for certain aspects of the Board's work ; *e.g.* production, labour, scientific, marketing, and finance. The N.C.B. officially assumed its duties on January 1, 1947, with headquarters in London.

For the new organisation the N.C.B. decided to group the pits into planning units of the size envisaged in the Reid Report. These units are known as Areas, and they now form the nucleus of management. There are 48 of them ; each is of a size not too large for effective supervision by their senior officials. In some cases they followed the group pattern created under private ownership, in others new grouping was required.

To avoid over centralisation the 48 areas are not directly responsible to headquarters. Instead 8 Divisions were formed, corresponding largely to the geographical distribution of the coal fields. In charge of each Division is a Board, collectively responsible to the N.C.B. The main function of these Boards is to frame a policy for their particular coalfield, to advise the N.C.B. on national policy, and to guide and help their own Area officials.

The names of the Divisions and their several component districts of the coalfields are as follows :—

Division.	*Districts.*
Scottish	East, West, Ayrshire.
Northern	Northumberland, North Durham, South Durham, Cumberland.
North Eastern	West Yorks, South East Yorks, South West Yorks.
North Western	North Lancs., mid-Lancs., North Wales.
East Midland	North Derby, Notts. and Lincs., Leicester with South Derby.
West Midlands	North Staffs., South Staffs.
South Western	West Wales, Cardiff, Newport, Gloucester with Somerset.
South Eastern	Kent.

Each Divisional Board resembles the N.C.B. in having a Chairman, Deputy Chairman, and other members (known as Directors), each in charge of a department of the Division's activities. The Chairmen were chosen from among men of wide administrative experience outside the industry; the other members had usually been in the industry previously. Each of these Boards has its own Divisional Headquarters.

The Divisional Boards carry heavy responsibility in the direction of the industry, but, having no legal entities, they function in the name of the N.C.B. In particular, the Divisional Finance Directors are specially responsible to the N.C.B. for sound finance.

The working of each Area is carried out in departments, corresponding approximately to those at N.C.B. Headquarters. The officials in charge of these departments constitute the Area Committee, which sits under the Chairmanship of an Area General Manager. He is responsible for Area management in all its aspects and answerable to the Divisional Board for its efficient working. His capacity for leadership is all-important to the success of the new organisation. But the

colliery manager remains statutorily responsible for the safe working of his pit, and its efficiency is still largely in his hands.

In most cases the Area Committee comprises about two dozen members, each covering one field of engineering, labour relations, marketing, coal preparation, estates, or accounts, etc. The Production Manager may deputise for the Area General Manager. In some Areas, usually those containing a large number of small collieries, control is by a small committee comprising a General Manager, Administrative Officer and Labour Officer.

Because of the urgent need for immediate improvement in output at the vesting date the first steps in reorganisation were short-term measures to that end. For example: concentration of men and equipment in the more productive seams and districts, and at the more productive pits; putting all available machinery to work, particularly conveyers; sending more men on to the coal face; recruiting and training more workers, including foreign workers.

In effect, during 1947 there was a nett increase of 26,000 in number of workers, making a tota of 711,400; the number of trunk conveyors increased by 1,500 or 16 per cent.; of 62 new locomotives delivered 32 were put to work underground. There was also an increase in the number of coal cutters and face conveyors, as well as in the more modern types of cutters and loaders.

Unfortunately these measures did not procure the desired output, largely because of the continued high rate of absenteeism and other labour troubles, and that in spite of the introduction of the five-day week on May 5, 1947. The output of deep-mined coal in 1947 was 187 m. tons, against a target of 190 m. tons. The Government's opencast workings produced an additional 10 m. tons.

The extension of working hours agreed to in November, 1947 (usually in the form of Saturday shifts) gave poor results, and the difficult labour situation early in 1948 led to the formation of a Joint Committee composed of representatives of the N.C.B. and N.U.M. which issued a Joint Report on Production in November 1948. The Report stated that the primary object, to produce maximum output of coal, depended largely on the number of men at the coal face, and recommended several measures to that end. In particular, to reduce the figure for absenteeism, which averaged about 14 per cent. in 1948, it was proposed that joint attendance committees of management and men be set up, with power to impose small fines and other disciplinary action, such as court proceedings or dismissal. But by a majority of 7 out of 10 the District Branches of the N.U.M. refused to accept these recommendations which representatives of their own National Executive had agreed, on account of the measures to stop absenteeism. Except in slump periods absenteeism has always been prevalent in coal mining in this country. But its ill-effect on production is vastly greater with mechanised methods than with hand getting. A proper understanding of the phenomenon is yet to seek, but a start has been made.*

The long-term policy of the Board in respect of mining operations proper is based largely on the recommendations of the Reid Report. It includes the development of 'horizon mining,' in which straight level roads are driven through the coal measures, of a kind suitable for locomotive haulage; the introduction wherever possible of 'room and pillar' systems, and of longwall retreating in preference to the more usual system of longwall advancing; the maximum use of coal-cutters and loaders and face conveyors; the extended use of electricity underground; the substitution of locomotive haulage (diesel and electric) for existing methods of rope-haulage and conveyors in roadways; and as a corollary, the introduction of mine cars of 1·5 tons to 3·5 tons capacity, in place of present tubs for cage winding; cars of capacity up to 6 tons are to be used with skip winding, which is to be extended where practicable; the Koepe system of winding is to receive further consideration. Large scale schemes for reorganisations of surface equipment and layouts are envisaged; plans for 37 major schemes and 21 of lesser size, and two new collieries and four new drift mines were either preparing or in course of execution during 1947.

To the greatest possible extent a variety of fully mechanised methods of cutting and loading are to be developed and applied, with due regard to their appropriateness to local conditions. The machines already established, and to be applied more widely and intensively, are mostly of British origin and designed for longwall work, *e.g.* the Meco-Moore cutter-loader; 30 of these machines are at work—mostly in the excellent conditions obtaining in the East Midlands Division.

RENTS AND ROYALTIES PAYABLE IN RESPECT OF MINERALS OTHER THAN COAL.

Common Surface Clay.—**The amount payable in royalty depends on the suitability of the clay for the purpose to which it is to be put and the rates are usually as follows :—**

Worked in the open, 3*d.* per ton but commonly fixed as so much per thousand bricks made and varies from 9*d.* to 2*s.* 6*d.* per thousand, according to the quality of the clay.

***Fireclay* as worked in collieries usually 2*d.* to 4*d.* per ton but special qualities command a higher royalty of from 8*d.* to 10*d.* per ton.**

***Ganister* frcm 5*d.* to 1*s.* per ton.**

*** 'Men in the Pits,' by Dr. F. Zweig. Gollancz. 1948.**

China clay and its derivatives, usually as follows :—

Blue and white China clay	3*s*. to 4*s*. 6*d*. per ton
Cream white	2*s*. „ 2*s*. 6*d*. „ „
Mica	9*d*. „ „
Pot granite	4*d*. „ „
Bricks	4*d*. „ „
Sand	4*d*. „ „

Oil Shale.—The dead rent merges into royalties and short workings are recoverable over the whole period of the lease. The royalty is 6*d*. to 1*s*. per ton, or when based on selling price, between $\frac{1}{18}$ and $\frac{1}{16}$ thereof.

Stratified Ironstone.—The certain rent varies from 8*s*. to £1 per acre. The royalty in Scotland from 6*d*. to 2*s*. 6*d*. per ton according to purity, or if on a sliding scale basis, from $\frac{1}{12}$ to $\frac{1}{4}$ of selling price. In South Walés about 8*d*. per ton, and in Cleveland about 5*d*. per ton.

Brown Hæmatite.—

Northampton	2½*d*. to 6*d*. per ton ; average 4*d*. per ton.
North Lincolnshire	9*d*. to 1*s*. 6*d*. ; average 1*s*. per ton (of 21 cwts.).
Leicestershire	6*d*. to 9*d*. ; average 7½*d*. per ton.
Oxfordshire	2*d*. to 3½*d*. per ton (of 21 cwts.).

Unstratified Ironstone.—Royalty is usually on a sliding scale basis, being a fraction of the selling price of the ore at or near the mine. These agreements vary slightly, but an example may be given of one which has been operative in the Furness district of Cumberland.

$\frac{1}{7}$th where the price of the ore is under 12*s*. per ton.
$\frac{1}{6}$th „ „ „ is between 12*s*. and 16*s*. per ton.
$\frac{1}{5}$th „ „ „ is above 16*s*. per ton.

Hæmatite.—In the case of hæmatite iron ore the certain rents vary from about £2 to £10 per acre.

BUILDING STONES.

Slate.—Royalty rent varies in amount in accordance with the character of the slate, etc., between $\frac{1}{20}$ and $\frac{1}{12}$ of the value of the slates at the quarry.

Other Stones.—Not possible to lay down hard and fast rules respecting rental value of all quarries. To quote two possible extremes, a *high class marble* near to good transport facilities would fetch a high rental, whereas a quarry of ordinary building *sandstone*, far distant from railway or canal, would probably be difficult to let, and then only at say 2*d*. per cubic foot. In the case of *Igneous Stone*, used for macadam, royalties ranging from 1*d*. to 4*d*. per ton are paid and for 'setts' from 3*d*. to 8*d*. per ton according to size. *Chalk* used with clay for making cement.—In a recent instance the royalty payable under the lease for the chalk and clay together was 6*d*. per ton of cement manufactured, and this was considered very favourable. *Sand and gravel* in the same instance were charged 2*d*. per cubic yard. These royalties were believed to be the lowest in the United Kingdom for these cement-making materials. The amount of the royalty charged in respect of *sand* and *gravel* is governed very largely by the position of the deposit and the demand, and ranged from 6*d*. to 1*s*. 6*d*. per cubic yard.

Gypsum.—The royalty varies from 6*d*. to 1*s*. per ton according to the quality of the mineral.

Salt.—The royalty for *rock* salt (Cheshire) amounts to about 2*d*. per ton. In the case of the sale from brine wells the royalty may be taken at 3*d*. per ton, to which should be added another 3*d*. per ton which the producer pays in tax for surface damage due to subsidence.

Fluorspar.—The rate varies from 1*s*. to 1*s*. 9*d*. per ton of dressed mineral sold.

METALLIFEROUS ORES.

Gold.—From $\frac{1}{15}$ to $\frac{1}{18}$ of the value of the gold raised.

Tin and Copper.—An average dead or certain rent is about 2*s*. to 7*s*. per acre. The royalty averages $\frac{1}{20}$ of the value of the ore at the mine.

Lead.—Variable and assessed on a sliding scale, the fraction varying with the selling price of the ore. The Great Laxey Mine in the Isle of Man, the ore of which contains silver, used to pay $\frac{1}{10}$ of the value of the ore ; other mines in the island $\frac{1}{12}$ to $\frac{1}{20}$.

Zinc.—The royalty in respect of zinc ore or blende is sometimes payable under a sliding scale, sometimes merely a fixed proportion of the dressed ore—usually about $\frac{1}{20}$th.

Barium.—The royalty on 'heavy spar,' as it is commonly termed, varies as from 9*d*. to 1*s*. 3*d*. per ton, but as high a figure a 2*s*. per ton has been charged. Usually, however, the royalty is from 3 to 5 per cent. of the selling price of the dressed spar at the mine.

Industrial legislation contained in the Mines and Quarries Acts, and the Orders, Rules and Regulations made under them, determine the conditions of extracting minerals. The whole body of legislation in recent years has been the subject of a comprehensive review by three Royal Commissions ; the first of these, appointed in 1906, was exclusively on coal mines and allied mines, and presented its reports in 1907 to 1910, and led to legislation in 1911. The second, on metalliferous mines and quarries, was appointed in 1910, reported in 1914, and has not yet been made the subject of legislative enactment. The third, appointed in December 1935, reported in December 1938, but, owing to the intervention of the war, its recommendations have not yet

been incorporated in an Act of Parliament. Certain portions of previous Acts were not repealed and will be found in the 1911 Act, and for the first time the spheres of demarcation between the Coal Mines Acts and the Factories and Workshops Acts were made clear. A large number of these Acts refer to the hours of work; thus in the 1908 Act deputies were allowed to work 9½ hours. The 1919 Act reduced the hours from 9½ to 8; the 1926 Act increased the hours by one hour longer; the 1930 Act reduced the hour again. The 1931 Act made deputies' hours 8½ hours, and the 1932 Act, which is in force at present (1945), has kept the deputies at 8½ hours. Boys may be employed according to the 1937 proposal only from 6 in the morning until 10 at night; this has not as yet been passed into legislation. The Government itself seems to accept the confusion by stating that the Coal Mines Acts may be cited as the Coal Mines Acts from 1887 to 1937, a space of some half century.

List of Principal Legislative Enactments in Force affecting Mines and Minerals.

1. An Act putting an end to life 'bondage' of miners, 1778.
2. The Land Tax Act, 1797 (sect. 4), and subsequent Acts.
3. Truck Acts, 1831, 1887, 1896.
4. Prohibition of Employment of Women and Girls in Mines; Regulation of Employment of Boys, and making other provisions relative to persons working in Mines, 1842.
5. Railway Clauses Consolidation Act, 1845 (sects. 77–85). See also *Howley Park Coal and Canal Company* v. *L. & N.W. Railway* (Law Reports, 1913).
6. Waterworks Clauses Act, 1847 (sects. 18–27). See also *New Moss Colliery* v. *Manchester Corporation* (Law Reports, 1908).
7. Acts for inspection of Coal Mines, 1850, 1855.
8. Crown Lands Act, 1866 (sects. 21–25).
9. Metalliferous Mines Regulations Acts, 1872, 1875.
10. Rating Act, 1874 (sects. 7–8).
11. Explosives Act, 1875 (sect. 59).
12. Slate Mines (Gunpowder) Act, 1882.
13. Public Health Act, 1875 (Support of Sewers) Amendment Act, 1883.
14. Coal Mines Regulation Acts, 1860, 1862, 1872, 1886, 1887–1908.
15. Quarries (Fencing) Act, 1887.
16. Quarries Act, 1894.
17. Mines (Prohibition of Child Labour Underground) Act, 1900.
18. Factory and Workshop Act, 1901.
19. Coal Mines (Weighing of Minerals) Act, 1905.
20. Notice of Accidents Act, 1906.
21. Mines Accidents (Rescue and Aid) Act, 1910.
22. Finance Acts: 1910 (sects. 20–27); 1912 (sect. 11); No. 2, 1915 (sect. 43); 1916 (sect. 46); 1917 (sect. 21); 1919 (sect. 33); 1920 (sect. 49).
23. Coal Mines Acts, 1911, 1914, 1919, 1930–1932, 1938, 1942, 1943.
24. Coal Mines (Minimum Wage) Act, 1912.
25. Coal Mines Regulation (Amendment) Act, 1917.
26. Education Act, 1918 (sect. 14).
27. Education (Scotland) Act, 1918 (sect. 17).
28. Petroleum (Production) Acts, 1918, 1934.
29. Income Tax Act, 1918 (First Schedule, Schedule A, No. III).
30. Coal Mines (Emergency) Acts, 1920, 1921.
31. Coal Mines (Emergency) Acts, 1920, 1921.
32. Mining Industry Acts, 1920–1926.
33. Employment of Women, Young Persons and Children Act, 1903, 1907, 1920 (sect. 1).
34. Mines (Working Facilities and Support) Acts, 1923, 1925. See Report of Mining Cases decided by Railway and Canal Commission Court under the Mines (Working Facilities and Support) Act, 1923, by P. J. Bamber.
35. Rating and Valuation Act, 1925.
36. Workmen's Compensation Acts, 1887, 1906, 1923–1934.
37. Rating and Valuation (Apportionment) Act, 1928.
38. Petroleum (Consolidation) Act, 1928.
39. Local Government Act, 1929 (sects. 2, 77 and Schedule I).
40. Coal Mines Act, 1940.
41. Mines (Working Facilities) Act, 1934.
42. Workmen's Compensation (Coal Mines) Act, 1934.
43. Public Health Act, 1936 (sects. 91, 93, 109).
44. Coal Act, 1942.
45. Coal Act, 1943.
46. Coal Industry Nationalisation Act, 1946.

NOTE.—Regulations made under the Coal Mines Act, 1911, are, with a few exceptions, published annually by the Stationery Office. See also Reports of Board of Trade under sect. 12 of Mining Industry Act, 1926 (Cmd. 3214, etc.) and under sect. 7 of the Coal Mines Act, 1930 (Cmd. 3905, etc.).

PRINCIPAL CLAUSES IN COLLIERY LEASE.*

Description of contracting parties (lessor and lessee) and of property demised.

Powers granted to lessee:—
- To break surface, make shafts, adits, etc.
- To win and work coal, bring to bank, make merchantable, dispose of.
- To work by instroke or outstroke or both.
- To make and dispose of coke and other products.
- To have necessary surface accommodation.
- *To have necessary wayleaves.
- To construct roads, tramways, railways, etc.
- To erect buildings, machinery, plant, etc.
- To work quarries of lime, clay, etc. (sometimes for own use only).
- To remove coals, dead and live stock, plant, etc., after expiration of lease.
- To relinquish at stated times before termination of lease.
- To renew lease for a further period.
- To generally do all things necessary for getting coal, etc.

Reservations to lessor:—
- Of mines and minerals not demised.
- *Of certain rights and wayleaves.
- Of use of ways, railways, etc.

'Habendum':—
- Lessee to have and hold property for a fixed term of years.

'Redendum':—
- Annual certain rent or deadrent, usually merging into royalty.
- Royalty per ton or otherwise.
- Rent for surface occupied.
- *Rent for wayleaves, waterleaves, or other facilities.

Provisoes:—
- No royalty payable in respect of colliery consumption.
- Power to make up short workings of previous years.
- Power to carry over excess workings to subsequent years.
- For distress in case of default in payments.
- For re-entry if rents, etc., remain unpaid for certain period.

Covenants on part of lessee:—
- For due payment of rents and royalties.
- For due payment of taxes.
- For compensating for damage done to surface or to neighbours.
- For erecting (or against erecting) buildings, houses, coke-ovens, etc.
- For keeping proper accounts accessible to lessor, and rendering the same.
- For keeping proper mine plans, and permitting inspection of mine.
- For leaving proper barriers.
- For working in approved and miner-like manner.
- Against sub-letting without authority.
- For giving up possession in good condition at determination of lease.
- For keeping mine, plant, buildings, etc., in good repair.
- For properly fencing or securing shafts, adit mouths, etc.
- For giving lessor power to purchase plant, etc., at a valuation on determination of lease.

Covenants on part of lessor:—
- For proper title.
- For quiet enjoyment.

Arbitration clause.

Leases of ironstone mines as also of metalliferous mines follow practically the same lines; as regards the latter, it is usually set forth that the royalty shall be a certain part of the value of the minerals, cleaned, dressed, calcined, or otherwise rendered merchantable, as the case may demand, at the expense of the lessee. There are often clauses specifying that a certain amount of development work, driving, sinking, etc., shall be performed.

A tacknote or license to prospect is often granted; this differs from a lease, inasmuch as it transfers no interest in the property. Its usual form is a license for one year to explore for all or for certain specified minerals within a definite area, usually for a nominal consideration, the licensee making compensation for all damages, and paying a royalty on all minerals gotten and sold. The license is generally renewable at the option of the licensee for a term of years, usually three, and the licensee is entitled to a lease upon conditions specified; this specification often takes the form of a schedule embodying the conditions of a draft lease.

*** This has reference to the principal clauses contained in colliery leases prior to the acquisition of the coal and adjacent minerals by the State. It is presumed that the Coal Commission will in granting new and consolidated leases retain many of these clauses but not so in respect of those relating to way-leaves.**

Classification of Mineral Deposits.

A mineral deposit may be defined as a portion of the earth's crust containing mineral matter of intrinsic value under favourable conditions.

The mineral substances that form the subject of mining operations may be classified as under :—

Group I., Fuels ; II., Ores ; III., Salts ; IV., Gems ; V., Rocks.

Group I., contains the different varieties of coal, asphalts, bitumen, and other solid hydrocarbons, mineral oils, etc., while sulphur may conveniently be included.

Group II. contains the various heavy metals either in the native state or in combination.

Group III. contains the haloids, sulphates, carbonates, silicates, phosphates, borates, etc., of the alkaline earths and alkalies.

Group IV. contains those minerals that are prized as gems on account of their hardness, brilliancy and transparency, as well as the semi-gems, such as opal, turquoise, etc., which though opaque are prized as precious stones on account of the beauty of their colouring and their rarity.

Group V. comprises the various rock masses, such as granite, slate, limestone, etc., the exploitation of which is more properly considered under quarrying.

With respect to their mode of occurrence, mineral deposits may be classified morphologically as follows :—

Tabular = Beds, veins.
Irregular = Masses, lenses, pipes, etc.

A more complete classification, based on genetic as well as morphological characters, is as follows :—

Class I.—Symphytic; **Syngenetic, contemporaneous or stratified**		**Group *a*, clastic.** **„ *b*, formed by chemical agencies.** **„ *c*, formed by organic agencies.** **„ *d*, disseminated through beds.**
Class II.—Epactic; **Epigenetic, subsequent or non-stratified.**	**Sub-Class A— Veins.**	**Group *a*, fissure** **„ *b*, bedded.** **„ *c*, contact.** **„ *d*, gash.**
	Sub-Class B— Masses.	**Group *a*, stockworks.** **„ *b*, disseminated in igneous rocks.** **„ *c*, connected with igneous rocks.** **„ *d*, connected with soluble rocks.**

Class I.—Group *a* includes auriferous alluvials, tin-bearing alluvials, ironsands, etc. Group *b* contains deposits due to chemical precipitation, to evaporation or to metasomatic changes. Practically all the stratified ironstones belong to this group. Of group *c* the most important representative is coal. Group *d* includes the Knottensandstein of Mechernich, the Kupferschiefer of Mansfeld, the Banket beds of Johannesburg, and similar deposits.

Class II.—Sub-Class A. Group *a* includes most of the important vein systems of the world, such as those of Cornwall, etc.

Group *b* includes veins parallel to the stratification.

Group *c* embraces certain veins following the contact of dissimilar rocks. (Not important.)

Group *d* contains veins carrying galena and zinc blende that characterise certain limestone districts, e.g., that of Missouri. (Sometimes classed in sub-class B.)

Sub-Class B. Group *a* consists of masses traversed by minute veinlets carrying ore, such as the tin deposits of Saxony.

Group *b* comprises such deposits as the Kimberley diamond-bearing pipes, the copper deposits of Lake Superior, etc.

Group *c* are very largely lenticular in form ; the pyrite deposits of the Huelva district are a typical example.

Group *d* may be represented by the red hæmatite deposits of Cumberland and Lancashire.

Branches of Mining.

I. Prospecting, the most important part of which is 'boring.'

II. Exploiting, including the various operations of opening up mineral deposits by shafts or adits, extracting and transporting the mineral, securing, pumping and ventilating the workings, and the proper care of the men employed.

III. Dressing or preparing the mineral extracted for the market.

COAL.*

CLASSIFICATION.

Various classifications of coal have been attempted from time to time. Perhaps the best is that adopted by the World Geological Congress, which met at Toronto in 1913, which took as the basis of their classification the fuel ratio

$$\frac{\textbf{Fixed carbon}}{\textbf{Volatile matter}}$$

*** See article 'Coal' by R. A. S. Redmayne in the 13th Edition of the *Encyclopaedia Britannica*.**

and in the case of sub-bituminous coals (inclusive of brown coal and lignite) the split volatile ratio

$$\frac{\text{Fixed carbon} + \text{Volatile combustibles}}{\text{Hygroscopic moisture} + \frac{1}{2}\text{ Volatile combustible}}$$

Coal being divided into four main classes and three sub-classes, letters instead of names being used, namely A, B, C and D. A corresponding to anthracite, B to bituminous coal, C to cannel and D to sub-bituminous coal. Between anthracite and bituminous coals occur the *semi-anthracites* or dry steam coals of South Wales and elsewhere, and the smokeless steam coal; whilst in the sub-bituminous class occur such different coals as black lignite, little distinguishable in appearance from ordinary bituminous coal, and brown coal.

Commercially speaking bituminous coal is divisible into gas, coking, house, manufacturing and steam coal.

The following are some characteristics and analyses of various coals:—

Professor Elwood A. Moore, the American authority, gives the following as an average composition of *brown coal* compiled from analyses of these coals from all parts of the world:

Moisture	14·42 per cent.
Volatile matter	40·78 „ „
Fixed carbon	36·00 „ „
Ash	9·32 „ „
Sulphur	1·14 „ „

Lignite differs from *brown coal* in that it has retained its woody structure.

The calorific value of brown coal and lignites varies greatly, *e.g.* 3·270 to 5·760 B.T.U. (brown coals of Germany), 8,000 to 9,600 B.T.U. (Canadian lignites).

Bituminous coal comprises many sub-classes, *e.g.* (1) *gas coals* or those bituminous coals which have the highest volatile hydrocarbon content, the most remarkable of which is 'Cannel' coal. Some cannels are very rich in volatile matter, that at Newbattle yielding 13,720 cubic ft. of gas (of 35 candle power) per ton of coal. It is a requirement of a good gas coal that it should yield about 10,500 cub. ft. of gas per ton with an illuminating power of 16 candle power.

(2) *Coking Coals.*—Coals which are possessed of the quality of intumescence or swelling upon the application of heat. A high class coking coal would have some such analyses as the following (*e.g.* of a well-known Durham Colliery):—

Fixed carbon	67·31 per cent.
Volatile hydrocarbon	25·61 „ „
Sulphur	0·78 „ „
Ash	5·10 „ „
Moisture	1·20 „ „
Phosphorus	0·005 „ „

The purest *coke* in the world is Ramsay's Garesfield (Durham) with

Carbon	97·6 „ „
Sulphur	0·85 „ „
Ash	1·55 „ „

Anthracite, which has a high fixed carbon content, and a low oxygen content, and *lignites* and *brown* coals, which have a low fixed carbon content and a high oxygen content, do not coke.

The sub-bituminous coal of Waikato, New Zealand, which is a glossy black immature coal, has a calorific value of 11,000 B.Th.U.

(3) Nearly all coals can be applied to domestic heating. It is largely a question of convenience and price as to what class of coal is employed as a *house coal.*

(4) *Gas coals* are usually coals which give off at least 10,000 cub. ft. of gas per ton of coal. A characteristic analysis of a good Durham gas coal shows a content.

Fixed carbon	62·32 per cent.
Volatile hydrocarbons	31·32 „ „
Sulphur	0·41 „ „
Ash	4·35 „ „
Moisture	1·60 „ „

Yield of gas 10,800 cub. ft. per ton with an illuminating power of 18·10 candles.

Cannel Coal contains the highest volatile hydrocarbon content of any coal. The following analysis is that of a Scotch cannel.

Fixed carbon	42·48 per cent.
Volatile hydrocarbons	52·88 „ „
Sulphur	0·22 „ „
Ash	3·10 „ „
Moisture	1·32 „ „

This coal gives 14,135 cub. ft. of gas per ton with an illuminating power of 37·36 candle power.

(5) *A manufacturing or iron smelting coal*, though raw coal is seldom used in these days for the smelting of iron, should be low in phosphorus, sulphur and ash content and possess a high fixed carbon content. The phosphorus and sulphur are detrimental to iron, and the ash not only goes to form slag, but detracts from the heating power of the coal. In (6) *steam coal*, the items of chief importance are the heating power, the character of the ash, and the sulphur content. The coal should have a sufficiency of hydrocarbons to permit of easy ignition and consumption of the fixed carbon ; beyond this it conduces to the formation of smoke. Incombustible matter reduces the value, so the ash content should be low and of a non-fusible character, as fusible ash forms clinker which is injurious to the firebars, necessitates considerable stoking and loss of heat. The following may be given as the composition of a first-class smokeless steam coal from the Rhondda Valley :—

Moisture	1·24
Volatile hydrocarbons	18·53
Fixed carbon	77·19
Ash	3·04

Heating value, 7,755 calories.
Evaporation power, 14·44.

(7) *Anthracite* is coal from which nearly all the volatile hydrocarbons have been expelled. (Some steam coals, as, for instance, the *dry steam coals* of South Wales, in the matter of fixed carbon content, stand between the ordinary steam coals and anthracite.)

Anthracite is hard, compact, glossy and difficult to ignite. Its calorific value is very high, attaining to a maximum of 15,531 B.Th.U. (8,624 calories) in the case of the best Welsh anthracite and an evaporative power of 16·06 lbs. of water. The composition of a good anthracite (Stanllyd Vein Blaina Colliery, Carmarthenshire) shows :—

Fixed carbon	93·76 per cent.
Volatile matter	5·09 „ „
Ash	1·15 „ „
Sulphur	0·75 „ „

and an inappreciable percentage of moisture.

Constitution of Coal.

The four constituents of coal, as identified by Dr. Marie Stopes, are :—

(1) Fusain. (2) Durain. (3) Clarain. (4) Vitrain.

Some authorities (*e.g.* Stach, *Glückauf*, 1927, p. 759, Kellett, *Trans. Inst. Min. Eng.*, vol. lxxv., 1928, p. 400) hold that Clarain is merely a mixture of Vitrain and Durain. In the U.S.A. these three classes are generally spoken of as Anthraxylon, Attritus, and Fusain. Fusain forms, usually, under 10 per cent. of a coal seam and is the highest in ash, this ash being always high in lime ; it also usually carries more salt than the rest of the seam. Vitrain is very low in ash, generally under 0·5 per cent. Durain ash contains mainly silica and alumina. Fusain is non-coking, but its calorific power is high and it is valuable for dust-firing. It is now generally admitted that Clarain is not a true constituent. (See papers by Prof. G. A. Hickling in *Trans. Inst. Min. Eng.* and *Proc. S. Wales Inst. Eng.*)

Calorific Power.

The calorific power may be calculated from the ultimate analysis by the 'Deutscher Verband' formula :—

$$\text{Cal. power in calories} = 81\,C + 290\left(H - \frac{O + N}{8}\right) + 25\,S - 6\,W,$$

where C, H, O, N, S, and W are the percentages, respectively, of Carbon, Hydrogen, Oxygen, Nitrogen, Sulphur, and Moisture.

Gontal's formula is applied where only the proximate analysis (percentages of fixed carbon, volatile matter and ash) is known :

Let C be percentage of fixed carbon, V of volatile matter, A of ash ; then

$$\text{Cal. power in calories} = 82\,C + a\,V,$$

where a is a coefficient depending upon the percentage of volatile matter in the ash-free coal, *i.e.* upon $\frac{100\ V}{100 - A}$ as follows :—

$\frac{100\ V}{100-A}$	5	10	15	20	25	30	35	40
a	145	130	117	109	103	98	94	80

The ash of coal varies usually between 2 and 15 per cent.

1 lb. of ordinary coal requires theoretically 130 cub. ft. of air for complete combustion.

COKE.

Ordinary bituminous coal, containing 25 to 30 per cent. of volatile matter, coked in a modern regenerative bye-product coke-oven plant, yields approximately :—

Coke . . .	70–75 per cent.	Ammonia as ammonic sulphate	1–1·25 per cent.
Tar . . .	3·5–4·5 „	Crude benzol	1 „

Surplus gas, 3,000 to 5,000 cub. ft. per ton of coal. Total gas, 9,000 to 10,000 cub. ft. per ton of coal.

The composition of this gas is about as follows :—Hydrogen, 48 to 65 per cent.; methane, 25 to 34 ; carbonic oxide, 6 to 8 ; carbonic acid, 2 to 2·5 ; hydro-carbons, 2 to 4 ; nitrogen, etc., 5 to 12 per cent. The calorific power is about 480 to 600 B.Th.U. gross, or 425 to 550 B.Th.U. net per cub. ft.

Modern bye-product ovens connected with ironworks are often fired with blast-furnace gas, so that the whole of the coke-oven gas is rendered available. Blast-furnace gas contains about : carbonic oxide, 30 per cent.; carbonic acid, 10 per cent. ; hydrogen, 1 per cent. ; nitrogen, etc., 59 per cent.; its net calorific value is about 100 B.Th.U. per cub. ft. See 'Notes on the Utilisation of Blast-furnace Gas,' by W. B. Baxter, *Journ. Iron and Steel Inst.*, vol. cxxviii, p. 71.

WEIGHT OF COAL.

The specific gravity of coal varies from 1·1 to 1·6, most bituminous coals approximating to 1·3. The following table shows the weight of coal contained in one acre of superficial area per inch thickness of seam, etc.

TABLE OF THE WEIGHT OF COAL IN DIFFERENT CIRCUMSTANCES.

Specific Gravity.	Weight in the Natural Bed, per Acre, per Inch Thick, in Tons.	Weight of a Cubic Foot in the Broken State, in Lbs.	
		Large Coal.	Small Coal.
1·10	111·411	42·62	37·12
1·15	116·475	44·56	38·81
1·20	121·540	46·50	40·50
1·25	126·604	48·43	42·18
1·30	131·668	50·37	43·87
1·35	136·732	52·31	45·56
1·40	141·796	54·25	47·25
1·45	146·860	56·18	48·93
1·50	151·925	58·12	50·62

Specific gravity × 101·283 = weight in tons per inch thick per acre.

To find the contents of coal of any given specific gravity per acre multiply the figure in the table by the thickness of the seam in inches; the thickness measured along a vertical line should always be taken, as the correct result will thus be obtained, whatever may be the inclinatior of the coal seam.

The weight of coal in its broken state, that is, as it comes to the surface, in tubs or otherwise, will depend on its largeness; the large has been computed to weigh in proportion to the solid coal as 62 to 100, and the small as 54 to 100.

The calculation now most commonly adopted in the North of England to ascertain the weight of coal in the solid state is to reckon 19 cwts. per cubic yard = 78·815 lbs. per cubic foot. On this calculation, a seam 1 foot in thickness would contain 1,532 tons 13 cwts. 1 qr. 10 lbs. per acre, and a seam 1 yard thick would contain 4,598 tons per acre.

A summary way of reckoning the quantity of available coal in a given area of a seam is to take an acre of coal 1 inch thick to produce 100 tons. This leaves a sufficient margin for faults and losses in barriers and in working.

SIZING OF COAL.

In the coal trade coal is often sold by size, but there is no uniformity in practice even in the same field. Above 4 ins. to 6 ins. it is usually sold as Best Coal, House, or Best House ; a very general scheme of designations is :—

Cobbles . . .	between 5 ins. to 6 ins.	and about 3½ ins. to 3 ins.
Treble nuts . .	„ 3½ „ 3 „	„ 1½ „
Double nuts . .	„ 1½ ins.	„ 1 in. to ¾ in.
Single nuts . .	„ 1 in. to ¾ in.	„ ½ in.

Duff, slack or gum below ½ in.

Some collieries also make beans, say about $\frac{5}{8}$ in. to $\frac{3}{8}$ in. and peas, say about $\frac{3}{8}$ in. to $\frac{3}{16}$ in. The dimensions may signify plate screens with round or with square holes, bar screens or wire gauze; in the two latter they may be measured in the clear or centre to centre. All these different methods are in use.

With reasonably closely sized coal of whatever dimensions it is found that as ordinarily heaped the coal occupies 55 per cent. of the total volume of the heap.

The proper sampling of coal is a difficult problem that is now attracting much attention. The methods of the British Standards Institution are generally used.

METALLIFEROUS MINERALS.

In America they make a further class with what they call Industrial Minerals. (See correspondence in 1936 between Professor H. Louis and the American Institute of Mining and Metallurgical Engineers.)

By *metalliferous* minerals or *metallic* minerals are meant those minerals that are technically utilisable for the extraction of the ordinary commercial heavy metals, metals of the alkalies and alkaline earths being excluded; this is almost identical with the meaning of *ores*, or with Group II, of above classification. In United States of America and Canadian statistics, a wider meaning is given to *non-metallic* minerals, which includes all minerals not actually used for the extraction of metals, *e.g.* ochre, although this is an oxide of iron, because it is used as a pigment and not as a source of iron.

Composition of more important Ores.

Name.	Formula.	Percentage of Metal.
Magnetite	Fe_3O_4	Fe 72·4
Hæmatite	Fe_2O_3	,, 70
Limonite	Fe_2O_3,H_2O	,, 60
Brown hæmatite	Hydrated ferric oxide, with 10–15 per cent. H_2O	,, 45–55
Siderite	$FeCO_3$	,, 48·3
Ilmenite	$(TiFe)_2O_3$	,, 30–40
Pyrites	FeS_2	,, 46·7
Pyrrhotite	Fe_7S_8	,, 60·5
Pyrolusite	MnO_2	Mn 63·3
Manganite	$Mn_2O_3H_2O$	,, 62·5
Wad	Impure hydrated manganese oxide	,, 25–50
Chalcopyrite	$Cu_2SFe_2S_3$	Cu 34·6
Chalcocite	Cu_2S	,, 79·8
Malachite	$2(CuO)CO_2H_2O$	,, 56
Cinnabar	HgS	Hg 86·2
Galena	PbS	Pb 86·6
Cerussite	$PbCO_3$	,, 77·5
Anglesite	$PbSO_4$	,, 68·3
Blende	ZnS	Zn 67
Calamine	$ZnCO_3$	,, 52
Stibnite	Sb_2S_3	Sb 71·8
Mispickel	$FeAs_2S_2$	As 46
Cassiterite	SnO_2	Sn 78·7
Wolfram	$(FeMn)OWO_3$	W 70
Scheelite	$CaOWO_3$	,, 74·8

Ironstone.

Ironstone is the term applied to stratified or symphytic deposits workable for iron, iron ore to epactic deposits, veins or masses. Bessemer ore or hæmatite ore is generally understood to mean iron ore with less than 0·035 per cent. of phosphorus when the ore contains about 50 per cent. of iron. Such ores are usually quoted on a basis of 50 per cent. of iron, with a definite increase or deduction in price for each unit of iron above or below that figure, and a penalty for every unit of silica above a given percentage, often 8 per cent.

The yield of ironstone is often taken roughly at 2,000 tons per foot thick per acre.

TABLE OF BRITISH MESOZOIC IRONSTONES.

Formations.			*Name of Ironstone.*
CRETACEOUS—Neocomian		Lower Greensand	Seend, Wiltshire.
	Wealden	Weald Clay	
		Hastings Sand .	Sussex and Kent. Westbury, Wiltshire, Claxby, Lincolnshire.
JURASSIC	Upper Oolite		
	Middle Oolite . . .	Corallian . . . Oxford Clay	Dover, Kent. Westbury, Wiltshire.
	Lower Oolite . .	Great Oolite Estuarine Beds Northampton Sands	Northamptonshire. Cleveland Dogger or Top Seam.
	Upper Lias	Upper Lias Clay Toarcian . . .	Isle of Raasay.
	Middle Lias . . .	Middle Lias Limestone . .	Cleveland Main Seam. Cleveland Pecten Seam. Cleveland Avicula Seam.
		Marlstone. . .	Blenheim, Oxfordshire. Melton Mowbray, Leicestershire. Caythorpe, Lincolnshire.
	Lower Lias		Frodingham, Lincolnshire.

COMPOSITION OF BRITISH MESOZOIC IRONSTONES, DRIED AT 100° C.

	Iron.	Silica.	Lime.	Sulphur.	Phosphorus.	Loss on Ignition.
	per cent.	per cent.	per cent.	per cent.	per cent.	per cent.
Cleveland Main Seam	30	12	5	0·3	0·5	26
Northamptonshire	38	17	3	0·15	0·7	22
Lincolnshire	25	9	21	0·2	0·35	36
Oxfordshire	28	13	11	0·15	0·3	35
Raasay	25	10	17	0·5	1·0	20

HAEMATITES.

These weigh 3–3·5 tons per cubic metre in place, 2½ tons per cubic metre in mine cars, 2 tons per cubic metre stacked.

COPPER, TIN, LEAD AND ZINC ORES.

Copper, tin, lead and zinc ores are usually sold by assay. The value is calculated on the current market price of the metal multiplied by the percentage of metal present less the percentage estimated to be lost in the operation of smelting; from the value thus found there is then deducted the 'returning charge,' being the estimated cost of smelting the ore, including the smelter's profit, etc. The following were the generally accepted (pre-war) methods :—

COPPER ORES.—Weight paid for is gross weight less moisture and less 1 per cent. draftage. Wet assay less 1 per cent. for ores assaying 3–4 per cent. (deduction varied proportionately for richer or poorer ores); for value per unit (1 per cent.) take 0·01 of price of best selected copper, less returning charge of 2*s*. 3*d*. to 2*s*. 6*d*.

Example.—Cargo of 1,000 tons ore assaying 4 per cent. copper, 1¼ per cent. moisture. Net weight $= 1{,}000 - \left(\frac{1{\cdot}25}{100} \times 1{,}000\right) = 987{\cdot}5$ tons.

4 per cent. − 1 per cent. = 3 per cent. copper to be paid for.
Best selected quoted that day 82*l*. 10*s*. per ton = 16*s*. 6*d*. per unit, less 2*s*. 3*d*. returning charge = 14*s*. 3*d* 3 units at 14*s*. 3*d*. = 42*s*. 9*d*. per ton of ore.
Value of cargo 987·5 tons at 42*s*. 9*d*. = 2,110*l*. 15*s*. 7½*d*.

LEAD ORE.—Weight paid for is net dry weight. Percentage paid for is dry assay less 4 units, on basis of soft foreign lead price in London, subtracting 40*s*. per ton for returning charge. Silver above 3 oz. per ton paid for at current rate for fine silver. Price is understood to be delivered smelter's works.

Example.—150 tons net of galena assaying 78 per cent. of lead and 5 oz. of silver per ton quotations being: lead, 20*l*. per ton; fine silver, 2*s*. 3*d*. per oz. Lead, 150 tons at (78 − 4) per

cent. = 111 tons at 20*l*. = 2,220*l*., less 150 tons at 40*s*. = 1,920*l*. Silver, 150 × (5 − 3) oz. = 300 oz. at 2*s*. 3*d*. = 33*l*. 15*s*. Total value of ore, 1,953*l*. 15*s*. Silver quotations are generally in pence per ounce standard, *i.e.* 925 per mil fine.

Zinc Ore.—Weight paid for is net dry weight. From percentage of zinc, as determined by wet assay, subtract 9 units, multiply by $\frac{18}{20}$ of price of spelter, and subtract returning charge of 3*l*.; difference is price per ton delivered smelter's works.

Example.—If ore assays 50 per cent., London quotation for spelter being 18*l*., the ore is worth £ $\left(\left[\frac{50-9}{100}\cdot\frac{18}{20}\cdot 18\right]-3\right)$ = 4*l*. 0*s*. 3*d*. per ton.

The general gangue of most ores is quartz; solid quartz averages about 165 lbs. per cub. ft. (13½ cub. ft. to the ton), broken about 100 lbs. (22½ cub. ft. to the ton), but varies considerably; the weight of veinstuff containing metalliferous minerals can be calculated from the specific gravity of the minerals.

Table of Specific Gravities and Weights per Cubic Foot of Minerals.

Mineral.	Sp. gr.	Cwts. per cub. ft.	Mineral.	Sp. gr.	Cwts. per cub. ft.
Copper pyrites	4·1–4·3	2·34	Hæmatite	4·5–5·3	2·73
Copper glance	5·5–5·8	3·15	Brown hæmatite	3·4–4·4	2·17
Malachite	3·7–4·0	2·20	Spathic ore	3·4–3·9	2·04
Cinnabar	8·9–9·0	5·00	Pyrolusite	4·8–4·9	2·67
Galena	7·3–7·7	4·18	Cassiterite	6·4–7·1	3·76
Cerussite	6·4–6·5	3·60	Wolfram	7·1–7·6	4·12
Anglesite	6·1–6·4	3·50	Scheelite	5·9–6·1	3·34
Zinc blende	3·9–4·2	2·23	Quartz	2·5–2·8	1·50
Calamine	4·0–4·5	2·40	Calcspar	2·5–2·8	1·50
Stibnite	4·5–4·6	2·50	Dolomite	2·8–2·9	1·56
Pyrites	4·8–5·2	2·78	Fluorspar	3·0–3·2	1·73
Mispickel	6·0–6·4	3·45	Felspar	2·5–2·8	1·50
Magnetite	4·9–5·2	2·80	Barytes	4·3–4·7	2·51
Ilmenite	4·5–5·0	2·64	Witherite	4·3–4·4	2·41

A rib of mineral 1 in. thick = 3 cub. ft. per square fathom.

Calculation of Mineral Mixtures.

The percentages of two minerals present in a mixture can easily be calculated if the sp. gr. of the mixture and of the ingredients is known. A convenient method is that proposed by R. T. Hancock.

Take a weight of x grms. of the mixture containing two ingredients of sp. gr., a and b respectively, where $\frac{1}{x} = \frac{1}{a} - \frac{1}{b}$ or $x = \frac{ab}{b-a}$; then the volume of this weight of a alone will differ from that of the same weight of b alone by 1 c.c., and for every 1 per cent. of a (the lighter substance) present in the mixture, the volume will be greater by 0·01 c.c. than the volume of b alone and for every 1 per cent. of b present, less by 0·01 c.c. than the volume of b alone. It is usually convenient to take $10x$ grms., when the differences will be 0·1 c.c.

Example.—To determine percentage of cassiterite (sp. gr. 7) and 'black sand' (sp. gr. 4·5) in a mixture of these. Since $\frac{1}{4{\cdot}5} - \frac{1}{7} = \frac{1}{12{\cdot}6}$, take 126 grms. Volume of mixture, measured by pouring the mixture into a burette partly filled with water and noting the difference in the burette readings, = 20 c.c. Volume of 126 grms. of black sand would be $\frac{126}{4{\cdot}5}$ = 28 c.c. Hence percentage of cassiterite = 10 (28 − 20) = 80 per cent.

Non-metallic Minerals.

Non-metallic minerals in this country are generally held to include Groups III. and IV. (p. 854), *i.e.* compounds of the alkaline earths and alkalies and gems, together with such minerals as quartz, coal, sulphur; a mineral such as scheelite may be classed either as a metallic mineral or as a salt; in such a case the object for which the mineral is worked is generally taken into account, *e.g.* scheelite would usually be classed as a metalliferous mineral because it would be worked as a source of tungsten and not as a source of lime.

BORING.

Boreholes are put down by one or other of the following methods :—

I. Percussive. (a) Using solid rigid rods.	II. Rotary : A. Core drills.
(b) Using ropes.	(a) Using diamond.
(c) Using hollow rods (water-flush method).	(b) Using steel cutting teeth.
	(c) Using chilled shot.
(d) Using a hydraulic ram.	B. Hydraulic rotary oil drill.

Generally speaking, percussive methods are used when the primary object is to get a hole down rapidly, where information as to the nature of the ground passed through is of less importance, whilst rotary core drills, by the use of which a more or less continuous core is obtained, are employed when an accurate knowledge of the strata passed through is the first consideration. Sometimes both methods are combined ; thus in some of the boreholes put down in Kent, boring through the secondary rocks was performed by percussion, whilst rotary core-boring was substituted after the coal measures had been reached. Percussion boring by the ordinary 'English' method can be used to depths of about 500 ft., costing about 10*s*. to 20*s*. per foot, and advancing 2 ft. to 6 ft. per twenty-four hours. Beyond this depth a free-falling cutter must be used, and in this way depths of 3,000 ft. or more may be reached.

Rope boring with round ropes on the American system is used almost exclusively in boring for oil, salt, etc. The plant for a hole 2,000 ft. deep would weigh about 80 tons complete and cost about 2,000*l*.; holes range from 6 ins. to 15 ins. in diameter. In Pennsylvania and Ohio such boring has been done for as little as 2*s*. per foot, and contracts are usually let at prices ranging from 4*s*. to 8*s*. The rate of boring is from 20 ft. per day upwards. In Middlesbrough, 8-in. holes have been put down to depths of about 1,000 ft. at a cost of about 8*s*. per foot with a speed of 60 ft. per day. Boring with flat ropes by the Mather and Platt system is used for very large boreholes, as for instance for waterworks ; holes up to 45 ins. in diameter have been bored by this method.

The water-flush method, of which there are many varieties (Fauck, Raky, etc.), is used in hard rocks for boreholes of medium size (rarely under 8 ins. in diameter). Very fast boring can be done in this way ; e.g. in the Ruhr district a borehole was put down to a depth of nearly 2,000 ft. at an average speed of 157 ft. per twenty-four hours ; 30 to 50 ft. is quite a usual rate down to depths of about 2,000 ft., and over 20 ft. a day has been bored down to a depth of over 3,000 ft. (See 'North Wales Coalfield, Exploration by Rapid Boring,' by R. Foster. *Trans. Inst. M.E.*, vol. cvii, 1918.)

The hydraulic ram method appears to offer certain important advantages, especially for boreholes of moderately large diameter, but has hardly passed the experimental stage. The best known example of this method of boring is the Wolski and Frich boring apparatus.

Core Boring.

Diamond boring is in use to-day for hard rocks ; the main drawback is the high price of the 'diamonds' (bort, used in the softer rocks, costs 2*l*. to 3*l*.* per carat, carbon costs 6*l*. to 12*l*. per carat), which tells severely against boreholes larger than 6 ins. in diameter. The cost of diamond boring in this country is about 7*s*. 6*d*. per foot down to 500 ft. and 12*s*. to 18*s*. down to 1,500 ft. for a 5-in. borehole. A 5-in. borehole put down in Scotland to 1,800 ft. cost only 6s. 4d. per foot ; the rate of advance was about 6 ft. per day. In British Columbia a borehole 2,200 ft. deep cost 8*s*. 6*d*. per foot, and was bored at the rate of 10 ft. per day. In Victoria the Government does much boring to aid prospectors ; the bores are mostly shallow, but in one case a depth of 1,110 ft. was reached in about 120 days ; the average cost of boring in coal measures in 1912 was 9*s*. 3*d*. and in the harder rocks 14*s*. per foot. In Nova Scotia 2 in. diamond boreholes put down by the Government in moderately hard ground to depths of 200 to 400 ft. averaged about 9*s*. per foot in 1913.

Steel cutting teeth can only be used in tolerably soft ground. The cutter should make 15–25 r.p.m. A 10-in. borehole in Kent was put down by this method by the Calyx Company to a depth of 1,400 ft., the speed at times reaching 10 ft. per hour.

The chilled shot method is very generally applicable, and is used for relatively large bore holes in hard ground. The cutter should run at 100–150 r.p.m. ; about No. 5 shot is generally used. It is usually somewhat slower but cheaper than diamond boring. The consumption of shot with a cutter 4½ ins. in diameter ranges from 0·15 lb. per foot bored in shale to 1·2 lbs. in hard sandstone, but may be much higher in broken ground, where shot escapes into the fissures. In Nova Scotia 6 in. Government boreholes by this method averaged 10*s*. 6*d*. per foot in 1913.

* Under conditions at present existing, e.g. high wages and high cost of materials, the figures quoted under this heading would have to be materially increased in order to be severally applicable for 1949.

The hydraulic rotary drill is the name given to a form now much used in boring oil-wells; it cuts out the whole of the material, leaving no core, the debris being brought to the surface by a water-flush. Over 100 ft. per day has been bored by this method.

Boreholes, especially those bored by rotary methods, are apt to deviate from the vertical, and a number of ingenious appliances have been devised for determining the amount and direction of the deviation. This is specially important when boreholes are put down in connection with the freezing method of shaft sinking (see p. 869, also *Trans. Inst. Min. Eng.* 1929-30, vol. lxxix, p. 309). See also for methods adopted for the determination of the amount of deviation from verticality and deflection of boreholes Vol. 1 'Modern Practice in Mining,' by Sir R. A. S. Redmayne, third edition, Longmans Green & Co., where a good example of the survey of the east borehole sunk by Turf Mines, Ltd., Transvaal, is given.

Location of Bore-holes for Valuation of Alluvial.*

When ground is systematically tested by pits or boreholes at regular intervals, three conventional views are possible:—

1. That there should be so many pits or holes per acre, or
2. That the distance between pits or holes should be so much, or
3. That no point on the ground should lie beyond a certain distance from a pit or hole.

The first two are those most generally held, but the last seems the most logical, and to attain it with the least amount of work the holes should lie at the corners of a series of equilateral triangles, the length of whose sides, or the distance of the pits apart, is equal to the maximum distance under condition (3) multiplied by $\sqrt{3}$. A line is run in any desired direction and staked at intervals equal to the distance between pits. From these stakes lines are turned off at angles of 60° or 120°, and staked at similar intervals starting from the turning-off point. The stakes are numbered to indicate their position on their own line and the number of the turning-off point, so that the pits sunk at them can be readily identified.

The advantages of this method over the more usual square lay-out are considerable. There is 6 per cent. less work involved in the lay-out of the holes, and 23 per cent. less in the sinking and sampling. The computation of values can also be more accurately done. Interpolation of values can be done on the plan, much as in contouring from spot levels, and a reliable assay plan prepared. (*R. T. Hancock.*)

Geophysics for the Location of Minerals.†

Magnetic methods; electric methods (self-potential method, equi-potential method, parallel wire method, single electrode method), electro-magnetic methods; gravitational methods (torsion balance); seismic methods; radio-active methods; geothermal methods. A Chair of Geophysics has been established at the Imperial College of Science, South Kensington. These methods of prospecting for minerals have come increasingly into use of late years, and have been found of service in the work of geological surveying.

BLASTING.

Explosives are divided into low explosives, with a slow rending action and fired by simple ignition, and high explosives, with a rapid shattering action, fired by means of a detonator. The rate of explosion of gunpowder, typical of the former, is 3 to 4 metres per second, and of blasting gelatine, typical of the latter, up to 7,000 metres per second.

For blasting in fiery coal-mines explosives known as safety explosives are employed, which have a lower temperature of detonation. No explosive can, however, be considered really safe in a fiery atmosphere. A shot-hole in coal should never be placed in the solid, i.e. in coal that has not previously been undercut.

If D be the depth of a properly placed shot-hole, the volume of rock blasted out averages with one free face $0{\cdot}37\ D^3$; with two free faces volume $= 0{\cdot}84\ D^3$, and with three free faces $1{\cdot}33\ D^3$. In drifting and shaft sinking the volume blasted out is approximately $0{\cdot}75\ D^3$. If L be the line of least resistance, D = about 1·75 L.

The quantity of explosive required for charging a shothole is determined by experience and though formulae exist (notably that of Chalon and its modifications) small reliability can be placed upon them in practice.

* See 'The Valuation of Alluvial Deposits,' by W. R. Rumbold. *Trans. Inst. Min. Met.* 2nd edn.), vol. xxxvii, 1928, p. 437.

† See 'Applied Geophysics,' by A. S. Eve and D. A. Key, Cambridge University Press (1st Edn.), 1929; also papers *Trans. Inst. Min. Eng.*, vols. lxxxvi and lxxxix, and 'Scope of Geophysical Prospecting' in 'Free State and New Rand Gold,' by D. Jacobsson, 1945.

Low tension detonators are best in series, high tension in parallel. Electric detonators are generally fired by magneto-exploders. Certain explosions are being attributed to faulty exploders, and dry batteries are being recommended in the place of exploders. (*Trans. Inst. Min. Eng.*, vol. lxxxiii., p. 222.)

Electric detonators consist of a fuse-head electrically fired, which communicates the ignition to an ordinary detonator. The fuse-head is generally contained in a paper tube, the empty space being filled with molten sulphur and the fuse-head cemented by a waterproof cement into the detonator. Electric detonators are distinguished as low tension and high tension; the low-tension fuse-head has a resistance of 0·9 to 1·3 ohms., the high tension 1,500 to 50,000 ohms, which, by bad treatment, may be reduced to as low as 50 ohms. Some makers make a fuse-head with an intermediate resistance of about 100 ohms. In the low-tension detonator the terminals of the copper wire leads are connected by a piece of fine wire, which the passage of the current heats to redness, whilst there is no such wire in the high-tension fuse-head. The wire consists of an alloy of copper, manganese and nickel, and the fuse-head proper in a low-tension fuse consists of copper acetylide, whilst in the high tension it is a mixture of this substance and graphite. In both types of fuse the fuse-head terminates with a mixture of chlorate of potash and carbon, which gives a flame to ignite the detonating composition. The low-tension fuse requires current of at least 0·4 amp. to fire it.

Liquid air is used in the ironstone mines of Lorraine; a cartridge containing sawdust, powdered cork, soot, or a similar substance impregnated with liquid air or liquid oxygen is employed. No detonator is required, but the shot must be fired within 15 minutes of charging. The addition of aluminium powder to the charge greatly increases the detonating action. The quantity of liquid air used is 0·3 litre per ton of ironstone, and the cost of this explosive per ton blasted is given as 0·303 per cent. as against 0·571 per cent. with cheddite and 0·597 per cent. with compressed black powder. Liquid oxygen is preferred to liquid air, because the latter is apt to produce fumes containing oxides of nitrogen which injuriously affect the miners, whereas with the former no noxious gases at all are evolved. It has also been used in some of the Lincolnshire * ironstone quarries.

Under Explosives Order No. 1194 of 1921, the production in the mine of an explosive by impregnating carbonaceous material with liquid air or oxygen is permitted under licence, thus modifying Section 50 of the Explosives Act, 1875. (See p. 951.)

STEMMING.

Stemming, also known as tamping, is the act of closing the shot-hole with some material which is intended to resist the action of the charge of explosive. A 'blown out shot' is one in which the stemming is projected from the shot-hole. Coarse sand makes an excellent stemming material. (See *Trans. Inst. Min. Eng.*, vol. lxxxii., p. 497, and vol. lxxxiv., p. 298.)

Professor J. A. S. Ritson suggests the use of sand and clay well mixed with calcium chloride. (See *Trans. Inst. Min. Eng.*)

SHEATHED EXPLOSIVES.

The late Professor Wheeler showed that the production of flame is not instantaneous but is the result of a series of reactions and that certain substances can delay or even inhibit one or other of these intermediate reactions.

Cartridges sheathed with a thin layer of sodium bicarbonate which acts as a cooling agent, thereby rendering the flame of the explosive less incendive, rather than as an 'inhibitor.' (See 'Sheathed Explosives,' by C. A. Naylor and R. V. Wheeler, *Trans. Inst. Min. Eng.*, vol. lxxxvi, p. 345.) These 'sheathed' explosives are now very largely in use in gassy and dusty collieries.

Detonators.

Detonators.—These are known by numbers, the charge of which varies somewhat. The following is about the average practice:—

Detonator numbers . . .	1	2	3	4	5	6	7	8
Weight of mixture in grains .	$4\frac{1}{2}$	6	8	10	12	15	23	30

The mixture consists of 80 per cent. fulminate of mercury, 20 per cent. chlorate of potash. Tetryl and Lead Azide mixture are also now used. (See *Proc. Roy. Soc. N.S.W.*, vol. lxiii, p. 85.)

Detonating Cord (Cordeau détonant) consists of a flat tube of lead, tin or aluminium usually 7 mm. wide, 3 mm. thick, filled with nitro-methylaniline or trinitrotoluol, containing usually about 10 grams of explosive per metre. An explosion is propagated through this at a velocity of about 7,000 metres per second. A piece of this cord is attached along a row of blasting cartridges and ensures their complete and instantaneous detonation, even in wet holes. It is stated that the efficiency of the explosive is increased by 20 per cent., and that experiments at

* *Journ. Iron and Steel Inst.* vol. cxvii., 1928; Pt. I., p. 186.

Lievin have shown that an explosive mixture of firedamp is not ignited by the detonation of 3 metres of cord.

Ordinary fuse burns at the rate of 30 seconds per foot.

(See also Explosives, p. 949.)

Elimination of Misfiring of Centre Shots.

As a means towards the elimination of misfiring of centre shots in large-scale blasting in coal mines, the United States Bureau of Mines recommends that the following practices be used whenever shots are to be fired in wet holes: Use waterproofed electric detonators with enamelled leg wires. When making connections with enamelled leg wires, care must be taken to scrape the ends of the wires well, otherwise the enamel will prevent good electrical contact. Fire the shots from an ungrounded power circuit that has a capacity of at least 30 kw. Use extra care when tamping the holes in order not to damage the insulation of the leg wires. Arrange the connections between the detonators so that they are supported clear of the earth or any other conducting medium. (See *Technical Publication* No. 257, *Amer. Inst. Min. Met. Eng.*, 1929.)

In England the firing of several shots simultaneously, except in the case of Cardox, which is not an explosive, is forbidden, although several authorities consider this the safer method of working.

'Safety Explosives.'

Safety explosives, or Sprengel explosives, are a group of explosives consisting essentially of ammonic nitrate with some organic substance. Other safety explosives consist of various ordinary explosives to which some substance is added to reduce the energy or temperature of the explosion. In this country no explosives may be used in collieries that can be considered 'fiery,' unless they have passed certain prescribed tests, when they are known as 'permitted explosives.'

A new type, though not really an explosive, known as Cardox, has been introduced; it consists of a steel shell charged with liquid carbonic acid which is converted into a gas producing very great pressure by means of a heater ignited by an electric current. The Cardox method of mining coal has for some time past been largely practised in U.S.A., and has recently been increasingly applied in Great Britain, both because of the safety of its use in 'fiery' mines and of the saving effected by the production of a higher percentage of 'large' coal as compared with that produced by other methods. (See 'The Cardox Method of Mining Coal,' published by Cardox (Great Britain), Ltd. 20 Copthall Avenue, E.C. 2.)

'Low density' explosives are being introduced with the object of getting among other presumed benefits a lower speed of explosion. (See *Trans. Inst. Min. Eng.*, vol. lxxiii., p. 216, vol. lxxvi., p. 305, and vol. xci., p. 127.)

SHAFTS.

(See 'Vertical Shaft Sinking' by E. O. Forster Brown, 1927, and vol. ii. of 'Modern Practice in Mining: The Sinking of Shafts,' by Sir R. A. S. Redmayne, K.C.B., third edition.)

Modern shafts are generally either rectangular or circular in cross-section; the former are used in sinking upon narrow veins or seams, where the ground is not very heavy nor the amount of water excessive, or where good timber is cheap; under other conditions round shafts are preferred, but local custom also exercises much influence. Inclined shafts are practically always rectangular; elliptical shafts have been sunk in some places (Forest of Dean) but are now rarely used.

Rectangular Shafts.

Rectangular shafts for large mines may be up to 25 ft. long by 15 ft. wide in the clear. Exceptionally large shafts are used in Pennsylvania, e.g., No. 5 Wilkesbarre anthracite mines, 52 ft. by 12 ft., Exeter Red Ash, 46 ft. 2 ins. by 13 ft. 10 ins.

Round shafts, say from 8 ft. to 24 ft. diameter in the clear; a colliery shaft to wind **1,000 tons per day is usually 16 to 18 ft. diameter.**

Rectangular shafts are mostly wood-lined; they are either cribbed or timbered with sets **holding lagging in place. In Fifeshire shafts up to 30 ft. by 12 ft. are cribbed with** 'barring' **of 9 ins. by 3 ins. up to 6 ins. by 12 ins. The timber for a shaft 23 ft. by** 8 ft., including buntons, **etc., would amount to about 125 cubic feet, and would cost, including nails,** wedges, etc., **about 18*l*. 10*s*. per fathom. The cost of sinking (the ground is usually easy) and putting in the timber is about 30*l*. per fathom. In the iron-ore mines of Cumberland and Lancashire shafts about 10 ft. by 5 ft. are cribbed with 10 ins. × 5 ins. plank, the ends being half checked; the total cost including sinking, timbering, timber, stores, etc., is about 30*l*. per fathom, the ground being carboniferous limestone that requires blasting. Sets are made of squared timber 4 ins. × 6 ins. for a small shaft, up to 9 ins. × 15 ins. for a very large one; sets are 3 ft. to 6 ft. apart, held together by hook bolts of 1½-in. round iron with wing nuts.**

Horned sets or bearer sets are put in about every 6 sets or so, depending on the nature of the **ground; lagging is often 9 ins. × 3 ins. plank.**

Round Shafts.

Round shafts are lined with brick or concrete in ordinary strata; special methods are adopted where shaft traverses water-bearing strata and the lining is required to keep back water. For shafts over 18 ft. in diameter ordinary brick may be used; for smaller shafts wedge bricks should be employed. Let f be the width of the inner face of the brick, b of the outer face, l the length of the brick and R the radius of the shaft. Then $b = \frac{(R + l)f}{R}$; if, as is often the case, $l = 2f$, this becomes $b = f + \frac{2f^2}{R}$. Instead of bricks, firebrick lumps 6 ins. to 8 ins. × 12 ins. to 18 ins. × 10 ins. to 15 ins. may be used. They should also be made wedge shaped in accordance with the above formula. A shaft 10 ft. in diameter lined one brick thick requires about 350 bricks laid in mortar, per foot of depth. The thickness of the lining depends on the nature of the ground and the diameter of the shaft.

COST OF SINKING.*

The cost of sinking may be taken as made up of five main items: Supervision, labour, materials, power, and interest on capital expended.

Walling, including the cost of the bricks, amounts to between 15*s*. and 30*s*. per cubic yard of brickwork. Walling a shaft 13 ft. in the clear, 15 ft. over all, cost about 15*l*. per fathom = 30*s*. per cubic yard; walling a shaft 22 ft. in the clear, 25 ft. over all, cost 24*l*. 5*s*. per fathom = 15*s*. 6*d*. per cubic yard. A walling curb costs about 10*l*. Sinking a shaft 12 ft. in the clear cost 24*l*. per fathom (sinkers and waiters-on, 13*l*. 10*s*.; general labour, 7*l*. 10*s*.; explosives and stores, 3*l*.). Sinking a shaft 13 ft. in the clear costs 23*l*. to 30*l*. for labour in ordinary strata, and 70*l*. to 100*l*. in basalt, per fathom.

A rough and ready rule for estimating the cost of sinking and walling circular shafts of from 18 to 21 ft. in diameter through ordinary coal measure strata, in which little water is encountered, is to allow £2 per ft. diameter per yard in depth. This is inclusive of cost of labour, hammer drills, tools and explosives, but exclusive of walling material.

The following are the particulars as to cost respecting a shaft of finished diameter 16 ft. which commenced sinking in September 1929 and finished September 1932. The shaft was sunk to a depth of 738 yards through ordinary coal measures in North Staffordshire. Progress while working one shift was about 4 yards a week and while working two shifts 8 yards a week. Most of the work was done working one shift per diem. Shaft lining was of brickwork, 14 ins. thick, set on oak curbs concreted into the solid to hold each length of brickwork independently, average length of bricking in each length approximately 18 yards.

Cost per yard sinking	£16.
Bricking	£3 10*s*.
Cost of material in shaft	£6,647.
,, ,, Steel headgear	£890 erected.
,, ,, Pulleys and bearings	£260
,, ,, Guide ropes (8) (half lock coil)	£1,844
,, ,, Winding ropes (2)	£865
,, ,, Cages (2)	£406
,, ,, 30″ × 6″ Markham (termed secondhand) winding engine	£1,500

The cost per yard for wages and materials was just under £30.

Most of the feeders of water were cemented off with liquid cement pumped in at a pressure of 100 lbs. per sq. in.

The contract price of £16 per yard included provision of a winding engineman and banksman and the handling of the dirt at the surface and putting it into its final position. This may be taken as a highly satisfactory result even though it was an instance in which little water was encountered.

The cost of sinking and tubbing a section of 22-ft. shaft at Blackhall, with feeders of water from 300 to 2,885 gallons per minute, was as follows:—

	£	*s*.	*d*.	
Underground labour	70	10	10	**per linear yard.**
Surface ,,	26	13	7	
Materials, coal and stores	87	2	1	
	£184	6	6	

* Under conditions at present existing, *e.g.* high wages and high cost of materials, the figures quoted under this heading would have to be materially increased in order to be severally applicable for 1949.

The cost of sinking a 22-ft. shaft through limestone and tubbing it, against feeders of water averaging 3,000 gallons p.m., may be averaged at about 275*l.* per yard of depth.
(*J. J. Prest*, Tr. Inst. M. E. xlvii. p. 611.)

Shafts may be lined with cement, either by setting sheet iron cylinders and ramming concrete round these, lifting the cylinders when the cement has set, or by walling with thin blocks or slabs (about 2 ft. 6 ins. × 3 ft. 6 ins. × 5 ins.), and ramming cement behind this facing ring. The cost of an 18-in. monolithic lining in a 22-ft. finished shaft was made up as follows :—

	s.	d.	
Mixing concrete	1	0	per cub. yd.
Cement	4	2	"
Lining shaft with concrete	2	6	"
Coal and stores	0	9	"
Steel cylinders and other plant	1	4	"
Total	9	9	per cub. yd.

There were 28 cubic yards to the fathom, hence the cost per fathom was 13*l.* 13*s.* It took seven hours to complete the concreting of one fathom.

The cost of walling a similar pit with brickwork was 24*l.* 3*s.* 3*d.*

Lining a 22 ft. shaft at Blackhall with a monolithic cement concrete (6 : 1) lining cost per linear yard :—

	£	s.	d.
Labour in loading sand and gravel	1	18	6
Enginemen and attending mixer	0	4	1
Cement	3	13	6
Interest and depreciation on shuttering, etc.	0	19	0
Coal, stores, etc.	0	10	6
Labour in shaft	3	1	7
	£10	7	2

Lining with concrete blocks 3 ft. 6 ins. × 2 ft. 6 ins. × 5 ins. backed solid by ramming in concrete, all also 6 : 1, cost practically the same. The average cost of sinking and walling with 18 in. concrete was 26*l.* 1*s.* 8*d.* inclusive. Suitable bricks would have cost 45*s.* per 1,000, and at this price brickwork in cement would have cost 15*l.* per running yard. (*Loc. cit.*)

In a 14 ft. pit the total costs of labour for sinking and concreting came to 15*l.* 15*s.* per fathom. The cost of the concrete was about 21*s.*, and the labour of lining the shaft with it about 1*s.* 6*d.*, or altogether 22*s.* 6*d.* per cubic yard. There were 10½ cubic yards per fathom, so that the cost of the concrete lining amounted to 11*l.* 16*s.* per fathom ; with four men at the surface and four men in the pit, the concrete was put in at the rate of a fathom in six hours. At Karwin, Silesia, an 18 ft. shaft was lined with concrete put in from above downwards at the rate of a fathom in fourteen hours at a cost of 62*l.* per fathom complete ; the same shaft lined in the upper part with brickwork had previously cost 132*l.* per fathom. Ferro-concrete has been used in place of simple concrete. At Zweckel, near Gladbeck, a shaft was lined with ferro-concrete slabs backed with concrete, into which the iron framework of the slabs was continued, for 23*l.* per fathom. At Rheinelbe a shaft 20 ft. in diameter was lined with Monier ferro-concrete 11 ins. thick at a cost of 2*l.* 5*s.* 6*d.* per cubic metre of lining, or 30*l.* 12*s.* per fathom of shaft : for a similar shaft at the Hansa Colliery lined 9½ ins. thick the prices were 2*l.* 3*s.* 6*d.* per cubic yard or 29*l.* 14*s.* per fathom.

Shaft Lining.

In heavily watered strata a lining has to be put in strong enough to resist the pressure of water. The thickness of such lining may be calculated as follows :—

Let R be the internal radius of the shaft, s the maximum admissible stress in the wall, p the pressure to be resisted, all to the same units, which are preferably inches and pounds, then :—

$$\text{Thickness of wall} = R\left(\sqrt{\frac{s}{s-2p}}-1\right)$$

(*Aldis*, N.E. Inst. M. E. xxxii. p. 201.

The resistance to crushing of the various materials used in lining shafts is :—

Common brick	800 to 1,000 lbs. per square inch.
Fire brick	1,500 to 2,000 " " "
Sandstone	1,500 to 5,000 " " "
Concrete 1 : 3 : 6 to 1 : 1 : 3	2,000 to 5,000 " " "
Average	3,000 " " "
Cast iron	90,000 to 120,000 " " "

Safe loads may be taken as $\frac{1}{10}$ of these ; for best brick laid in 1 : 3 Portland cement up to 250 lbs. ; for ferro-concrete up to 600 lbs. per sq. in. ; for frozen sand fully saturated with water at −12° C. up to 340 lbs. per sq. in. (*Alby*, Bull. Soc. Ind. Min. 1895, p. 273.) L. Sauvestre has shown that frozen sand is highly plastic, and that its resistance to crushing increases by 28½ lbs. per sq. in. for each drop of 1° C. (*Annales des Mines*, Ser. xi., vol. ix., 1920, Pt. I, p. 5.)

The pressure to be resisted is usually taken merely as the head of water—namely, 0·434 h lbs. per sq. in., where h is the head expressed in feet. For quicksands Poetsch adopts a formula equivalent to $p = 1{\cdot}042\ h \tan^2\left(45° - \frac{\alpha}{2}\right)$, where α is the angle of repose of the sand; Poetsch gives $\alpha = 35°$ for fine quicksand, and 20° for coarse sand. These figures are used in calculating the thickness of the ice-wall required in the freezing method of sinking.

Many approximations are used for ordinary shaft linings :—

J. J. Atkinson : Thickness in inches $T = \frac{5{\cdot}2\ R\ h}{s - 0{\cdot}434\ h}$, or for cast iron $T = \frac{5{\cdot}2\ R\ h}{9{,}000 - 0{\cdot}434\ h}$.

Chaudron: for cast iron, $T = 0{\cdot}8 + \frac{0{\cdot}434\ R\ h}{600}$ where $h < 165$ ft. and $T = \frac{130}{h} + \frac{0{\cdot}434\ R\ h}{600}$ where $h > 165$ ft. (here s is taken at only 7,200).

Riemer takes $s = 11{,}400$, and writes :—

$$T = 0{\cdot}8 + \frac{0{\cdot}434\ R\ h}{950} \text{ for } h < 165,$$

$$\text{and } T = \frac{130}{h} + \frac{0{\cdot}434\ R\ h}{950} \text{ for } h > 165.$$

Cast-iron tubbing is always stiffened by ribs and flanges, and these should be taken into account. (See *Morrow*, *Trans. Inst. Min. Eng.*, xxxiv., p. 100.)

Coffering.

Coffering may be used for moderate pressures and shallow depths ; costs range from 20*l.* 10*s.* per fathom (16-foot shaft) to 38*l.* per fathom (18-foot shaft), being considerably less than cost of cast iron tubbing.

Tubbing.

Tubbing is put in either by English or German method ; in the former, segments are 2 ft. to 3ft. deep, flanges are external, not machined, and the joints are made with wood sheeting and wedges ; in the latter, segments are 5 ft. to 6 ft. deep, flanges are internal, are machined, and the joints are made with lead sheeting and bolts. In both forms the tubbing is carried upon wedging curbs, followed usually by a foundation ring. In some few recent cases the wedging curb the been dispensed with, and the tubbing set direct upon a good bed.

Cost of Tubbing.

Cost of English tubbing varies with the price of pig iron, etc., from 5*l.* to 7*l.* 10*s.* per ton ; putting in the tubbing costs 2*l.* to 2*l.* 10*s.* per ton of tubbing put in, or 5*l.* to 7*l.* per fathom of shaft. The following are the detailed costs of tubbing a 15 ft. shaft with 12 segments to the ring, 2 ft. deep, ⅜ in. thick, each ring weighing 2 tons 10 cwt. :—

	£	s.	d.
Three rings, 7 tons 10 cwt. at 7*l.* 10*s.*	56	5	0
Sheeting, 213 ft. at 8*s.* per 100	0	17	0
Wedges, 3,500 at 10*s.* per 1,000	1	15	0
Labour, putting in and wedging	6	0	0
Total	64	17	0
Cost per fathom, about	64	0	0

To the above must be added a due proportion of the wedging curb. The castings for the curb of a 15-ft. shaft cost 25*l.* to 35*l.*, sheeting and wedges 5*l.*, putting in and wedging 10*l.* to 15*l.* Dressing the curb seat may cost from 10*l.* to 40*l.* according to circumstances. About ten men are usually employed in the shaft in wedging tubbing, and they can wedge two to three rings per shift.

The cost of 'German' tubbing usually comes out a little less than English tubbing per fathom. The seams and joints are all machined, this costing 3*l.* 10*s.* to 4*l.* per ring ; three shifts of four men can put in and bolt up four rings (20 to 24 ft.) per 24 hours.

The cost of 'German' tubbing in an 18-ft. shaft, the tubbing being 1 in. to 1¼ in. thick, 2 segments to the ring 5 ft. deep, was as follows in this country :—

	£	s.	d.	
Cast-iron tubbing, 13 tons 4 cwt. at 7*l.* 10*s.*	99	0	0	per fathom
Lead for sheeting and washers, 6 cwt.	6	0	0	
Bolts and nuts	3	5	0	
Labour	7	10	0	
Cost per fathom	115	15	0	

In the Minister Achenbach Colliery a shaft 17 ft. in diameter was tubbed with 18 rings of tubbing 8 ft. 2 ins. high, 1 in. to $1\frac{1}{4}$ in. thick, each ring composed of 10 segments. The weights of material employed were:—

Tubbing . . .	192 tons 10 cwt.	Lead washers . . .	1 ton 2 cwt.
Lead strips . .	2 „ 8 „	Bolts, nuts and washers .	6 „ 3 „

Where the ground is weak and apt to run, suspended tubbing can be used to much advantage. This method was used at Astley Green (*Trans. Inst. Min. E.*, xxxix. p. 543), also at Hamsterley (*Trans. Inst. Min. E.*, xxxviii. p. 320); the latter shaft was 10 ft. in diameter and 35 ft. deep. The cost was:—

Materials, 627*l.* 2*s.* 10*d.*, labour, 141*l.* 0*s.* 3*d.*, equal to 83*l.* 16*s.* per fathom. The method is to be recommended for safety, speed and economy.

Of late years cast iron with a small proportion of molybdenum has been found advantageous for tubbing. Where extremely heavy pressures have not to be resisted, ferro-concrete may be used and has been used in several cases. The cost of ferro-concrete tubbing resisting a pressure of some 800 lbs. per sq. in. for an 18-ft. diameter shaft is calculated at about £22 per lin. ft.

All forms of shaft lining are apt to suffer by corrosion, especially in many cases at the outer skin where the shaft traverses water-bearing strata in which the water contains salts in solution that are capable of attacking the lining. Galvanic action can also produce corrosion, and such action may be set up by the not uncommon method of tightening the joints in German tubbing by means of copper wire. In iron tubbing a layer of rust is apt to form, but this should be removed from the interior of the shaft as promptly as possible, because it is injurious to the underlying tubbing and does not form a protective layer as is often supposed. Brickwork is readily destroyed by corrosion, but as it is rarely used to line shafts through water-bearing strata, this is a matter of secondary importance. Neither concrete nor ferro-concrete can be expected to remain permanently water-tight, and both are very liable to corrosion. The best form of tubbing to resist corrosion is apparently double tubbing with concrete between the two cast-iron skins.—'Glückauf,' vol. lxix., p. 161.

The above, which may be called ordinary methods of sinking, can only be used where a moderate influx of water has to be contended with; the greatest quantity ever dealt with by these means has been in the sinkings at Blackhall Rocks, where the Horden Colliery Company has sunk two shafts through the Magnesian Limestone in which 14,300 gallons per minute were being pumped from a depth of about 90 yards. Very much less than this is generally sufficient to cause special methods to be resorted to, 5,000 gallons per minute being generally considered prohibitive.

Cost of Shaft Sinking in Gold Mines.

A shaft 4 ft. by 8 ft. in clear can be sunk in California, where wages are 14*s.* per shift, by hand drilling, at a cost of 4*l.* 10*s.* per foot. This includes labour, timbering, explosives and supplies. The hoisting done by a horse whim and no water to pump. The rate of progress is about one foot per day.

In Nevada a shaft 4 ft. by 8 ft. in clear was sunk 100 ft. in $31\frac{1}{2}$ days by means of hand drilling. The cost per foot was 6*l.* 10*s.* Labour 16*s.* per shift.

In Western Australia shafts vary from 4 ft. by 10 ft. to $5\frac{1}{2}$ ft. by $13\frac{1}{2}$ ft. On eleven of the mines in the district the cost per foot of shaft is from 11*l.* to 16*l.*, and from 2*s.* 9*d.* to 5*s.* 6*d.* per cub. foot of material excavated. The average monthly advance was 34 ft. to 60 ft. with an average of 50 ft. Machine drills used on all above shafts.

See 'Cost of Sinking the East Shaft of the New Kleinfontein Company, Limited,' by H. J. Way, *Trans. Inst. Min. Met. Eng.*, vol. xiii, 1903–04, p. 102.

In Utah, with wages 12*s.* per shift, a shaft 17 ft. by 65 ft. was sunk in ground of medium hardness at a cost of 12*l.* per foot sunk or 1*l.* 16*s.* per ton of material broken. Machine drills used. Pumping from 800ft. Rate of sinking 0·9 ft. per day.

On the Rand a shaft (Brakpan, Ltd.) 43 ft. by 9 ft. in the clear was sunk at a rate of 204 ft. per month. The working cost was 14*l.* 10*s.* per foot and the total cost 22*l.* 14*s.* per foot. Ground soft. Hand drilling employed.

The following are some of the Rand shafts and the cost per ton of material handled:—

Shaft.	Size.	Speed per Month.	Cost per Ton.	Method Employed.
			s. *d.*	
Rand Collieries . .	34 ft. by 9 ft.	120 ft.	10 9	Machine
Kleinfontein . .	34 ft. by 9 ft.	107 ft.	11 3	„
Brakpan . . .	43 ft. by 9 ft.	124 ft.	8 11	Hand.
City Reef	46 ft. by 9 ft.	125 ft. 6 ins.	9 2	„
Hercules . . .	47 ft. by 9 ft.	119 ft.	8 5	„
Wolhunter . . .	46 ft. by 9 ft.	95 ft.	10 6	„

The Inspiration Copper Company, Miami, Arizona, sunk and completed a shaft 12 ft. 4 in. by 16 ft. 8 in., 1,400 ft. deep in 10 months.

Some exceptionally fast sinking has been done on the Rand; thus, in one of the shafts on the Crown Mines 318 ft. were sunk in August, 1925, and in one on the Randfontein Estates 386 ft. were sunk in June 1926.

Winzing and Raising.

At Park City, Utah, winzes 11 ft. by 7·5 ft. in quartzite are sunk at a cost of 6*l*. per foot sunk, or 1*l*. 10*s*. per ton of material broken. The advance is about 1 ft. per day. In same mine raises, 17·5 ft. by 5·5 ft., are put up at a cost of 1*l*. 14*s*. per foot or 4*s*. per ton of material broken. The advance is at the rate of 3·3 ft. per day.

In Western Australia winzes are from 4 to 5 ft. in width and 6 to 8 ft. long. They are 50 ft. to 200 ft. deep, and the average total cost per foot in eleven of the mines is 4*l*. 15*s*.

The rises in the same district extend from 25 ft. to 120 ft. in height, and are 4 ft. to 5 ft. wide and 6 ft. to 8 ft. long. The total cost on the average is 3*l*. 18*s*. 7*d*. per foot.

Special Methods of Sinking.

In hard, very wet ground :—

Kind-Chaudron : maximum diameter of shaft, 15 ft.; greatest depth yet attained by it, 1,335 ft.; Nos. 1 and 2 pits Marsden Colliery (the first successfully sunk in this country by this method) cost respectively 362*l*. 8*s*. and 293*l*. per fathom; in the North of France shafts 8 to 12 ft. diameter to depth of 100 fathoms cost 80*l*. to 240*l*. per fathom, and the rate of sinking varied from 2 to 6 fathoms per month. In Germany it is estimated that to sink 55 fathoms by this process at a depth of about 300 fathoms would cost about 1,400*l*. per fathom for a shaft 14 ft. 6 ins. in diameter. (See *Bull. Ind. Min.*, 1909, ser. 4, vol. ii, p. 342.)

*Poetsch Freezing Process.**—Applicable both to fissured firm ground and to running ground. The brine used is a 25 per cent. solution of magnesium chloride cooled down to − 20° C., its freezing point being about − 35° C. The following table shows the pressure at which ammonia gas liquefies at different temperatures :—

Temperatures.		Pressures in Lbs. per Sq. In.
−20°C.	− 4°F.	27
−10°C.	+14°F.	41
0°C.	+32°F.	62
+10°C.	+50°F.	89
+20°C.	+68°F.	125

It is estimated that to produce 100,000 negative calories per hour at —15 C., 50-60 h.p. are required at the freezing plant, and that 0·25 lb. of ammonia and 2 gallons of brine are required for each yard in length of the freezing circuit.

Saturated sand (165 parts water to 1,000 parts sand) frozen at 32° F. has a crushing strength of 242–341 lbs. per sq. in., and at − 13° F. of 2,845 lbs. per sq. in.

Shafts have been sunk to 130 fathoms by this method, the cost for shafts 13 ft. to 16 ft. in diameter ranging from about 150*l*. to 250*l*. per fathom. The cost of sinking two pits, 12 ft. and 15 ft. 6 ins. respectively in diameter, at Ansin, in 1895, to a depth of 50 fathoms, was as follows :—

	£ s. d.	£ s. d.
Cost of freezing plant		73 2 0
Cost of freezing operations		48 6 0
Cost of sinking and tubbing : labour .	32 16 0	
Cost of tubbing . . .	66 10 0	
		99 6 0
Total cost per fathom		220 14 0

The cost of sinking a pair of 20 ft. shafts by freezing under ordinary conditions should be :—

	£
To a depth of 50 fathoms	17,500
„ „ 100 „	28,000
„ „ 200 „	50,000

(*W. W. Wilson*, Tr. Inst. M.E. xli. 353.)

* See *Rev. Univ. d. Min.*, 1931, p. 185.

Freezing an 18-ft. shaft at Monkwearmouth cost 230*l.* per fathom to a depth of 60 fathoms.

A deep sinking was carried out by this process at Beeringen in 1920; a depth of 1,640 ft. of clay, quicksand, and water-bearing chalk was first sunk through in this way, and subsequently the process was applied to the more difficult task of passing through a bed of running sand 46 ft. thick, at a depth of 2,000 ft. under a pressure of 926 lbs. to the sq. in.

(*Annales des Mines*, Ser. xi., vol. ix., 1920, Pt. I. p. 5.)

At Seaham, Co. Durham, there were three freezing units, one of 300,000 negative calories and two of 150,000 negative calories each per hour (300 and 150 h.p. respectively). There were five double-acting brine pumps 10 in. diam., 10-in. stroke; the brine was calcium chloride solution of sp. gr. 1·25. Two shafts have been sunk 21 ft. diam. in the clear to a depth of about 490 ft. The freezing was carried out in sections by a method which the inventors call the 'rational' method, which has also been applied with much success in the Campine, Belgium. (See *Ann. d. Min. Belg.*, vol. 22, 1921, p. 535; 'The Sinking of Londonderry Colliery, Seaham Harbour, Co. Durham, by the Freezing Process,' by J. L. Henrard and J. T. Whetton, *Trans. Inst. Min. Eng.*, vol. lxxv., 1928, p. 358.)

In the sinking of No. 19 pit at Courrières in 1909, which was sunk by the freezing method, difficulties were encountered in the upper part of the shaft owing to running water, and these were overcome by the injection of cement to a depth of 57 ft.; over 1,000 tons of cement were used. (*Annales des Mines*, Ser. xi., vol. ix., 1920, Pt. I. p. 38.)

In sinking through strata containing brine, *e.g.* in certain potash mines, the temperature must be reduced below − 35° C. Liquid carbonic acid must be used instead of ammonia in the freezing plant and calcium chloride brine. (*Glückauf*, vol. 63, 1927, p. 293.)

Sheet piling can be used for relatively shallow sinking, near the surface. At Bowburn Colliery Co., Durham, 6 fathoms of wet sand were sunk through by wooden piles in 1906 at a cost of 26*l.* per fathom for timber and 34*l.* 10*s.* for labour. Interlocking steel piling of joists of various forms has also been used.

The dropshaft method is employed in various forms, sometimes by weighting a cutting shoe and curb, on which an iron or brickwork cylinder is built up, or more usually by building a brick or concrete pressure block against which work hydraulic rams that force down tubbing of cast iron, or in exceptionally difficult cases of cast steel. The former method (forcing down a shoe by dead-weight) was found much superior to forcing a shoe down by jacks in sinking a shaft 14 ft. by 8 ft. through 70 ft. of loose strata at Des Moines, Iowa (*Trans. Amer. Inst. Min. Eng.*, vol. iv., p. 232). The following table shows probable costs for shafts of normal size in ordinary running ground (Hoffman):—

For depth.	Cost per fathom sunk. £	For depth.	Cost per fathom sunk. £
From 15 fathoms to 27 fathoms .	320	From 82 fathoms to 110 fathoms .	1,280
" 27 " to 55 " .	732	" 110 " to 137 " .	1,554
" 55 " to 82 " .	1,006	" 137 " to 165 " .	1,830

A sinking with a timber drop shaft lining is described in the February, 1922, *Bulletin of the Canadian Institute of Mining and Metallurgy.*

The *Honigmann* system has been used in a few cases down to depths of 70 fathoms on the Continent; the shaft is bored out by a dredge-borer, the walls being kept up by internal hydrostatic pressure, and the shaft is lined with tubbing composed of rings of channel iron riveted together and subsequently lined with concrete.

Cement Injections.—The Sonnenschein cementation process for sinking through quicksands consists in boring a number of holes about 5 m. deep, relatively close together, through the concrete plug in the shaft bottom. When a sufficient number of holes have been bored, water is forced down a bore-hole adjoining the one to be treated, thus forcing the quicksand up the latter, and leaving a conical cavity in the quicksand below the plug. Cement slurry is then forced down the borehole to be treated until the cavity previously formed is filled with cement, and the operation is repeated until a continuous body of cement, about 5 m. deep, is formed below the plug. The cement is then excavated until only enough is left to form a fresh plug, and the entire operation is repeated until the quicksand has been traversed.

The François Cementation Process.—The origin of the use of cement grout in mine shafts is not certain. Probably the first fully authenticated use was in the injection behind shaft lining at the Rheinpreussen shaft, near Homberg, in 1864, at a pressure of 210 lbs. to the square inch. Monsieur H. Portier at Courrières in 1900 successfully injected and sealed a considerable feeder of water behind an old shaft lining of wood. Cement injection was successfully employed at a shaft sinking in Germany by R. A. Weide, at Pohlhauer, Saxony, in 1901. Early examples of its use in France were by M. Reumaux at Lens; and by M. Deffieux at the Mine de Maries in 1908. In 1911 Mr. Albert J. François, who had previously spent some years on the Continent in experimenting with various systems of cement injection behind shaft lining and through various classes of strata, introduced his process into England. His process proved a valuable aid to the development and maintenance of coal mines in wet areas.

The process has since been introduced into France, Belgium, Germany and South Africa. In all, over forty shafts have been sunk by its means. The preliminary silicatisation

of the ground, which is an essential part of the process, was first tried at Hatfield Main, in 1911, with complete success. In porous or finely fissured strata, or alluvial deposits, or generally where the injection of cement slurry proves difficult, the preliminary preparation of the ground by the injection of solutions of silicate of soda and sulphate of alumina is used; the shaft is sunk by ordinary methods until the presence of water is indicated by means of test holes. A number of holes are then bored in the shaft bottom, pipes being cemented into them, their number, position and inclination being dependent upon the size of the shaft and the character of the strata. Through these pipes boreholes are sunk by means of a diamond drill. On the tapping of feeders by one of the holes the boring is immediately stopped and the pipe in the hole connected by means of a length of high pressure flexible hose to a column of pipes in the shaft, leading in turn to a cement pump and mixing tank on the surface. The holes are arranged in series for cementation and silicatisation, and into them cement pulp and mixtures of chemicals are pumped, the strength and proportion of the mixtures of chemicals, the percentage of cement, and the pressure employed being such as experience has shown to be best suited to the class of strata and pressure of water to be treated. The holes are cleaned out and reinjected or further deepened as necessary, the process being repeated with the various holes, boring and injecting of the different holes taking place simultaneously. The ground is treated in this manner until the desired length of ground has been rendered sufficiently watertight to allow mining and walling operations to proceed. The ground is excavated in the ordinary manner and reinforced concrete walling built up, any water not sealed off by the cementation of the boreholes being shut off by injections between the completed lining and the ground. On the walling being completed a second length of ground is excavated and walled, this procedure being continued till the water-bearing strata have been passed through. The lining employed through water-bearing or bad ground is ferro-concrete tubbing of monolithic construction. Cementation in ordinary ground will cost about 50*l.* per yard for a 20-ft. shaft (see *Transactions of Mining Engineers*, vol. liii., 1916, p. 22).

The process has been used in sinking a shaft 16 ft. in diameter at the Holditch Iron Mine, Chesterton, Staffs. The shaft is sunk through the water-bearing Keele strata to a depth of 632 ft., the upper bed giving off 30,000 gallons of water per hour. The total time occupied was 69 weeks, and the costs were 40*l.* 4*s.* 4*d.* for labour and 36*l.* 3*s.* 9*d.* for material per yard of finished shaft (1916–17), or together 76*l.* 8*s.* 1*d.*; it is estimated that sinking with underhung tubbing would have cost 91*l.* 4*s.* 8*d.* The shaft was lined with reinforced concrete. Cement for injection of boreholes 350 tons 13 cwt., for the concrete walling 998 tons 6 cwt., and for the injection of the walling 50 tons 10 cwt. (*A. Hassam and T. T. Mawson*, Trans. Inst. Min. Eng. lviii. p. 16.)

In porous sandstones *The François Cementation Co.* first injects separately solutions of silicate of soda and sulphate of alumina, and follows this up with cement. This process was used successfully at Hatfield Main, Thorne, etc. Two shafts were sunk successfully at Ollerton by this process.

For a more complete description of the process see paper by H. Standish Ball, *Proceedings of South Wales Institute of Engineers*, vol. xxxvi. No. 2, p. 111.

Mr. Eustace Mitton estimates the cost of sinking through the Bunter sandstone by cementation, including the installation of ferro-concrete tubbing to a depth of 200 yards for an 18-ft. shaft, at £350 per lineal yard. (*Midland Counties Institute of Engineers*, October 15, 1921.)

At two shafts at Harworth Main, where there was an indicated quantity of water of 3,000 gallons per minute, the shafts were rapidly and successfully sunk by the François process, with a total residual water in both shafts of 6 gallons per minute (see 'Harworth Sinkings,' *Colliery Guardian*, Nov. 17, 1922); while at two shafts recently completed at Ollerton Colliery, where the estimated amount of water was even greater than this, the shafts were sunk by the François process with an average of residual water of only two gallons per minute in each shaft. (See paper read by Mr. H. Eustace Mitton before the Institution of Mining Engineers (*Transactions*, vol. lxx.), November 4, 1925.)

The François cementation process was successfully employed in preference to sheet piling at Haugh Colliery, Kilsyth, Scotland, for sinking from the surface to some 6 fathoms in depth through running ground.

See also L. Souvestrie, *Annales des Mines*, 12th series; *Colliery Guardian*, Nov. 10, 1922; an article in the *Iron and Coal Trades Review*, Sept. 8, 1922; *Kropf, Industriebau*, Oct. 15, 1922; 'Cementation des Grès Vosgiens,' by F. Arguillère, *Rev. Ind. Min.*, Nov. 1 1927, p. 451. Papers by A. A. Barnes and J. R. Fox, *Inst. of Water Engineers*, June 1927.

The time required to cement a 20-yd. length of shaft in various rocks is as follows:

Bunter Sandstone	16 days	Limestone	18–19 days
Pennant Sandstone	16 ,,	Chalk	12 ,,

and the quantity of cement required per yard of 20–22 ft. shaft has been found to be:—

Pennant Sandstone	15 cwt.–3 tons	Bunter Sandstone	5 tons–6 tons.
Magnesian Limestone	2 tons–4 tons	Chalk	up to 4 tons.
Triassic Sandstone	3 tons–6 tons		

Where silicatisation is employed, the consumption of silicate of soda is 1½ to 5 tons per linear yd., and that of sulphate of alumina about half of this.

(*Vertical Shaft Sinking*, by E. O. Forster Brown, 1927, pp. 147, 148.)

Under special conditions when very rapid setting cement is required, 'ciment fondu' has been used in place of ordinary Portland cement, and in intensely cold ground a mixture of magnesia and magnesic chloride has been used.

LIST OF DEEP MINES* IN THE WORLD.

Mine.	Date (where known).	Locality.	Depth, Ft.	Mineral Wrought.
Robinson Deep	1941	South Africa	8,400	Gold
The Morro Vello Mine of the St. John del Rey Mining Company	1934	Brazil	8,051	"
†Turf Shaft, Robinson Mine	February 1932	South Africa	7,730	"
†Champion Reef Mine, Kolar	February 1932	India	7,160	"
Crown Mine	1941	South Africa	7,000	"
Ooregum Mine, Kolar	—	India	6,970	"
†City Deep, Southern Rand	August 1932	South Africa	6,900	"
†Mysore Mine, Kolar	—	India	6,270	"
†Village Deep Main, Central Rand	—	South Africa	6,263	"
Vlakfontein	1941	"	5,000	"
†Calumet and Hecla	—	L. Superior	5,683	Copper
Victoria Reef Quartz	—	Bendigo	4,614	Gold
Tamarack, No. 4	—	L. Superior	4,450	Copper
New Chum Railway	—	Bendigo	4,318	Gold
Providence (Shaft), Machienne Collieries	1908	Belgium	3,937	Coal
Flenu, Ste. Henriette	—	"	3,809	"
Viviers, Gilly	—	"	3,750	"
Przibram, Maria	—	Bohemia	3,694	Silver, Lead
Przibram, Adalbert	—	"	3,665	"
Lazarus New Chum	—	Bendigo	3,682	Gold
Kimberley Main Rock Shaft	—	Kimberley	3,601	Diamonds
Kimberley Prospect Shaft	—	"	3,520	"
Kennedy	—	California	3,555	Gold
Westfalen I	—	Ahlen, Westphalia	3,566	Coal
Sachsen I	—	Heesen, "	3,560	"
Zwickau, Morgenstern	—	Saxony	3,550	"
Long Tunnel Walhalla (incline shaft)	—	Walhalla	3,450	Gold
Oelsnitz, Frisches Glück	—	Saxony	3,055	Coal
Parsonage Colliery (Mine)	1918	Lancashire	4,000	"
Dolcoath, William Shaft	—	Cornwall	3,000	Tin
Kaiser Wilhelm II.	—	Clausthal	2,960	Silver, Lead
Ashton Moss	—	Lancashire	2,880	Coal
Cooks Kitchen	—	Cornwall	2,580	Tin
De Beers Main Rock Shaft	—	Kimberley	2,466	Diamonds
Rosebridge	—	Lancashire	2,446	Coal

The greatest depth for a single wind is (over) 5,000 ft. at Valkfontein on the Rand.

The Barnsley Bed was cut in the Thorne sinkings at 2,765 ft. on August 22, 1924.

The deepest workings in the world at present are: the Robinson Deep Mine, deepest point 8,400 ft. below shaft collar; Morro Velho, St. John del Rey, 8,051 ft.

In 1924 the total depth of all shafts in mines in Great Britain coming under the C.M.R. Act was 845,000 yds., and the ratio of shaft yardage to average tons of daily output was 1 : 1·183; in the U.S.A. the ratio of output appears to be at least 15 times as great.

Shaft Pillars.

The 'draw' in ordinary Coal Measure strata, practically horizontal, is generally between 5° and 15° to the vertical. In Staffordshire 'draws' between 8° 45′ and 11° 10′ were observed in strata dipping 1 in 24 (W. Hay, *Trans. Inst. Min. Eng.* xxxvi. p. 47). The Final Report of

* In some of these the depth of a single shaft is given but in others the deepest part of the mine from the surface, e.g. the Morro Velho Mine, is worked by a series of shafts sunk from different levels in the mine and in the case of the Robinson Deep Mine there is a vertical shaft of 4,000 ft. which is fed by inclines. The deepest vertical shaft in South Africa is that at Vlakfontein. At the Crown Mines, also in South Africa, a depth of about 7,000 ft. is reached, but here again the arrangement is somewhat similar to that at Morro Velho in that a vertical shaft from the surface is fed by a sub-vertical shaft. At Pendleton in Lancashire the shafts are 1,530 ft. in depth, but owing to the dip of the measure to the south a depth of over 3,460 ft. from the surface is reached.

† *Mining and Metallurgy*. August 1932, p. 370.

the Coal Conservation Committee (Cd. 9084), 1918, p. 64, implies that the maximum angle of 'draw' to be expected is 26° 35′. In secondary strata the 'draw' is generally between 15° and 30°.

There are reasons for assuming that the angle of draw in practically horizontal strata approximates to $45° - \frac{\text{angle of repose}}{2}$. (See *Annales des Mines*, 1931, p. 211.)

In Westphalia (*Zeitsch. f. B. H. u. S. W.*, 1897) the draw in the Coal Measure has been found to be 15° on both rise and dip sides when the dip of the seam is < 15°; for dips exceeding 15°, the draw is 15° on the rise side and increases uniformly up to 35° on the dip side, reaching this value with seams dipping 35°. In the overlying clay deposits the draw is uniformly about 20°. The depth of subsidence is $kT \cos a$, where T is the thickness of the seam and a its inclination; k is a coefficient, which is 0·8 when the goaf is not stowed at all; with a well-stowed goaf, k is 0·4 when a is 0° to 10°, 0·3 when a is 10° to 35°, and 0·25 when a is 35°.

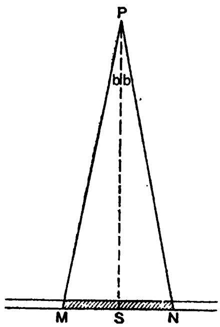


FIG. 1.

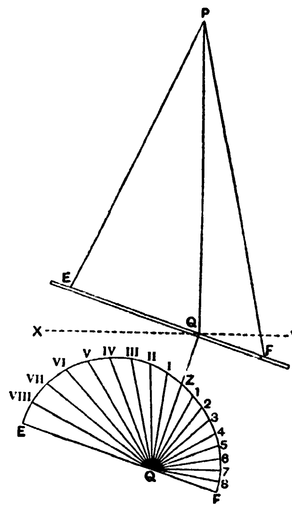


FIG. 2.

When a structure on the surface has to be protected from damage, a portion of the seam of coal or other mineral, usually spoken of as a pillar, has to be left unworked, and its position and dimensions must be calculated according to the draw. The simplest case arises when a single bed of mineral, lying horizontally, is worked out, and if the problem is to support a single point on the surface, obviously the pillar will be circular in plan, having a radius equal to D tan b, where D is the depth of the seam below the point to be protected and b the angle of draw. In fig. 1, PS = D, the depth of the seam, and SM or SN is the radius of the protecting pillar. It follows that if a number of seams one below the other are being similarly worked out, the pillars left in the lower seams will be progressively larger in proportion to the depth below the surface at which the seams occur. This method may be fairly well applied as long as the dip of the seam is less than 3°.

When the seam is inclined the position becomes more complicated, and is not thoroughly understood yet. The following may be taken as a general rule: Let b_l be the angle of draw on the lower side of the excavation, *i.e.*, on the rise side of the pillar, and let b_u be the angle of draw on the upper side of the excavation or the dip side of the pillar, b being the normal draw when the strata are horizontal

then

$$b_l = b + ckf(i) \text{ and } b_u = b - kf(i),$$

where i is the inclination of the seam of mineral measured in the usual way to the horizontal, c and k are coefficients, and $f(i)$ is some function of the inclination. This formula is purely empirical and should not be applied where i exceeds 30°; furthermore, $kf(i)$ can never exceed b.

Apparently $f(i)$ may be taken as sin i, k (in the coal measures) has approximately the value of 8, and c between 1·8 and 2.

Based on this formula, it is possible to calculate the dimensions of the pillar to be left in an inclined seam below a point on the surface which is to be supported, the depth of the seam below the point being, as before, D. It is easiest to calculate the dimensions of the pillar in the plane of the seam, and these can readily be projected on a horizontal plane if required.

The length (in the plane of the seam) of the pillar in the direction of full rise is given by the expression $D \sin b_l / \cos (b_l - i)$ and in the direction of full dip by $D \sin b_u / \cos (b_u + i)$, whilst the width at right angles to the dip will be $D \tan b$ on either side of the centre. Intermediate points must now be calculated as shown in fig. 2, where P is the point to be supported, Q being a point in the seam vertically beneath it. QZ is drawn in the line of strike, QE to the full rise, and QF to the full dip, the lengths of these lines being determined as above. Divide the quadrant ZQE into equal parts by radial lines set out at angles of 10° to each other, such as the lines Q I, Q II . . Q VIII. In an inclined plane dipping at an angle of $i°$, a vertical plane making an angle of $x°$ with the horizontal will cut the inclined plane in a line having a dip of $y°$, such that $\tan y = \sin x \tan i$. Thus the tangent of the inclination of the line Q I $= \sin 10° \tan i$, of the line Q II $= \sin 20° \tan i$, etc. By substituting the values of these angles of dip for i in the formula given above, it is possible to calculate the angles on the rise and dip sides respectively, and in this way the radii of the pillar in the plane of the seam can be determined, as shown in fig. 10, where D is assumed to be 500 ft., i 18°, and b 10°. The shape of the pillar thus obtained is an inconvenient one, and in practice it is best replaced by a rectangle with truncated corners just sufficiently large to include the curved area obtained by the above calculation. For approximations a circle drawn on the line of full dip may be substituted, or spirals increasing regularly from the width of the pillar on the strike to the length of the pillar on the full rise and decreasing in the other direction from the width of the pillar on the strike to the length of the pillar on the full dip.

(See 'Protection Against Damage by Subsidence,' by H. Louis, *Trans. Inst. Min. Surv.*, 1928.

Another method, being practically that of the late Mr. T. A. O'Donahue,* is as follows:—If D be the depth of the seam and T its thickness, in a horizontal seam a pillar of radius R should be set out such that $R = (0{\cdot}19 + 0{\cdot}01\,T)D$, all dimensions being in feet. If the seam is inclined at an angle $d°$ as above, the distance from the vertical through the point to be protected to the edge of the pillar on the rise side should be $R + \frac{2D}{3}\left(\frac{\tan d}{1 + \tan^2 d}\right)$, and on the dip side $R - \frac{D}{3}\left(\frac{\tan d}{1 + \tan^2 d}\right)$. A circular pillar should be set out with radius $R + \frac{D}{6}\left(\frac{\tan d}{1 + \tan^2 d}\right)$, the centre of the circle being at a distance $\frac{D}{2}\left(\frac{\tan d}{1 + \tan^2 d}\right)$ to the rise side of the point S. If a rectangular pillar is preferred, the square circumscribing this circle may be set out. Various other formulas have been proposed, but the two above given appear to be satisfactory.

In practice it is generally an area and not merely a point that has to be protected, the area being usually a circle, a square, or a rectangle. As a general rule this can be done in various ways. If the area is a circle or a square the radii of the pillar can easily be calculated. In other cases a number of suitable points on the periphery of the area (*e.g.*, the corners of a rectangle) should be selected and the protecting pillar for each be determined as above; the irregular figure thus obtained should be enclosed in a figure of some simple form (*e.g.*, a rectangle with truncated corners), re-entering angles being avoided. In cases like this, where the solution of the problem involves so many uncertainties, an approximate result is all that can be obtained, and a liberal margin should be allowed for safety.

Bord and Pillar working.—Pillars may be square but are mostly rectangular, sides varying from 30 to 80 yds.; 30 yds. by 40 yds. to 40 yds. by 60 yds. are usual sizes. Some engineers give areas proportional to depth; for seams 100 to 200 yds. from surface, pillars about 30 by 40 yds.; 200 yds. to 300 yds., 35 by 50 yds.; 300 to 400 yds., 40 by 60 yds.

The Coal Conservation Committee of the Ministry of Reconstruction recommends the following proportions:—

Depth from Surface.	Area of Pillar.
< 100 yds.	300 sq. yds.
100–200 "	500 "
200–300 "	800 "
300–400 "	1100 "
400–500 "	1400 "
500–600 "	1700 "

In *panel working* a rib 30 to 60 yds. wide is left between the districts, the latter being usually 20 to 50 acres in area. In American mines the width of rib is often determined by the expression $T\,(5 + 0{\cdot}01\,D)$, where T is the thickness of the seam and D its depth below surface.

* See 'Subsidence caused by Coal-Mining,' by T. A. O'Donahue, *Trans. Inst. Min. Eng.* vol. lxxviii., 1929, p. 91.

Welsh Stall working.—Single stalls are usually about 12 yds. wide with 12-yd. ribs, double stalls up to 18 yds. wide with 24-yd. ribs between them. In America the following method is sometimes used :—

Width of pillar $W = \frac{T\ D}{100}$, and width of stall $= \frac{D}{W}$, where T is the thickness of seam and D its depth, both in feet.

Subsidences.

(See Second and Final Report of the Royal Commission on Mining Subsidence. Cmd. 2899. 1927. 'Mining Subsidence,' by Hy. Briggs, 1929, also 'Principles of Subsidence' (Lane and Roberts) and 'Subsidence resulting from Mining' (L. E. Young and H. H. Stock), *University of Illinois Bulletin.*)

According to O'Donahue, the working out of a coal seam 4 ft. thick will cause a subsidence of the surface of 3 ft. if the seam be not more than 200 yds. deep, and from 12 to 18 ins. if it be 800 yds. deep. It was shown that at Monkwearmouth, working two seams 6 ft. and 4 ft. thick respectively at depths of 1,725 ft. and 1,600 ft., caused surface subsidences of 6 ft. to 6 ft. 9 ins.

At Dunkirk collieries working two seams 5 ft. and 6 ft. thick, 50 yds. apart, at a depth of 445 yds., caused a subsidence of 6·8 ft. in five years and 8·92 ft. in ten years, after which it ceased.

In Staffordshire working a 5 ft. seam at a depth of 1,626 ft. caused surface subsidences varying from 1·34 to 1·47 ft.: these reached their maximum in about 29 months.

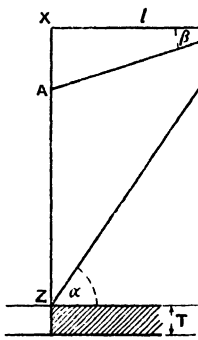


FIG. 3.

In the Doncaster district the surface subsidence may be expected to be about 60 to 70 per cent. of the thickness of the mineral extracted. (Report on Mining and Drainage in an area around the County Borough of Doncaster. Ministry of Agriculture and Fisheries, 1928.)

Final subsidence at the surface is equal to the thickness of the seam actually worked out (*i.e.* the thickness of the seam after allowing for any stowage) less the expansion due to the breaking up of the superincumbent measures. If t is the thickness of the seam and d the depth of the seam, both in feet, and k the coefficient of expansion of the strata, the surface subsidence equals $t - kd$; in ordinary coal measures with seams of moderate thickness and at moderate depths k appears to be of the order of 0·2 per cent., so that if D is the depth of the superincumbent strata in hundreds of feet, then the subsidence equals $t - 0{\cdot}2$ D feet. The coefficient appears to increase with increasing thickness of the seam, which gives the strata more room to break, and it decreases as depth increases, probably because the weight of the overlying strata re-compresses to some extent the strata expanded by fracture, and this effect appears to be sufficiently well marked to prevent $t - kd$ ever coming down to zero, though it is not impossible that it might do so in the case of narrow seams at very great depth.

There are too few data available to give numerical expressions for the above general statements.

Besides the vertical subsidence over the worked-out area, a zone of draw exists over the solid coal, the breadth of which = depth of seam × tan angle of draw as given above. Let XY, fig. 3, represent the zone of draw.

Under normal conditions as to character of superincumbent strata, dip of strata, thickness of seam and absence of faults in the neighbourhood, mining engineers usually reckon on a maximum extension of draw, measured on the surface, equal to one-third of the depth from the surface of the excavation.

Let S (= XA) be full vertical subsidence; l (= XY) be length of draw (width of zone of draw); H = depth of seam; T = thickness of seam; α° = angle of draw measured to the horizontal, said to be about 70° in coal measures; β° = angle of inclination of drawn surface.

There is a horizontal displacement from Y to X, and this is a maximum in the middle of the zone of draw, becoming zero at X and Y.

Tan $\beta = \frac{S}{l}$ l = H cot α. For seams 4ft. to 10 ft. thick, S = (0·5 to 1·0) T, when unstowed, (0·25 to 0·5) T, when hand-stowed. Maximum horizontal displacement varies from 0·05 to 0·3 T and $= \frac{Sl}{4H} = \frac{S \cot \alpha}{4}$.

(*A. H. Goldreich*, '*Die Bodenbewegungen im Kohlenrevier und deren Einfluss auf die Tagesoberfläche.*' 1926.)

In the Zwickau (Saxony) district seams dipping 15°, aggregating 28 ft., worked at depths from 2,000 ft. to 3,000 ft., cause subsidence of 70 per cent. of thickness worked by goafing, 40 per cent. by hand-stowage, 10 to 5 per cent., averaging 7 per cent., by hydraulic stowage. The angle of draw by the two former methods is 80°, flatter by last-named method. In mining under densely built-over areas hydraulic stowage gives better protection against damage than leaving safety pillars. (*Goldreich, op. cit.*)

Coal Getting Machines.

In hand kirving, the average height of a kirving is 13½ ins. in front and 1⅜ in. at back, by a depth of 3 ft., giving a volume of 1·9 cub. ft. of coal kirved out per linear foot of coal face (equal to 16 per cent. on a 4 ft. seam). In a 6-hour shift a hewer in ordinary coal will kirve a bord 5 yds. wide to a depth of 1 yd. or about 7½ sq. ft. per hour of shift. A man will kirve about 9 sq. ft. per hour's actual hewing time after the jud has been nicked in a bord; in a wall he can kirve about three-quarters as much as in a bord, and in longwall work he can kirve one and a-half times as much as in average narrow work. Allowing for his other work a hewer in Northumberland and Durham averages about 38 sq. ft. of kirving per shift.

Machine cutting in those cases where the conditions are favourable to its application gives up to about 65 per cent. more output per man employed; on the average about 10 per cent. more round coal is made by machine than by hand, and a greater output of coal can be obtained from a given seam. Where a mechanical coal cutter can be worked with a face conveyor the ideal is reached in respect of the use of both. The other conditions making for economic application of mechanical cutters are:

(*a*) where the seam is not abnormally thick;
(*b*) where the coal will stand to be cut;
(*c*) where the seam contains a soft floor or a band of soft dirt in which the cut can be made;
(*d*) a good roof;
(*e*) a flat or moderately inclined seam;
(*f*) where a large output is a desideratum and can be profitably disposed of.

Coal-cutting machines for longwall work are chain-cutters; disc-cutters and bar-cutters have gone out of use. Machines may be air-driven, using compressed air at 35 to 65 lbs. per sq. in. at the machine, or electrically-driven (direct, alternating, or tri-phase) at voltages of 250 to 450. The power consumption varies between wide limits, between 12 and 30 h.p. Compressed air coal-cutters cost 250*l.* to 300*l.*, electrically-driven coal-cutters 350*l.* to 450*l.*; a complete plant for a pit using 10 machines would cost about 10,000*l.* with the former and 11,000*l.* with the latter form. The area undercut per hour varies from 45 to 140 sq. ft., 90 sq. ft. being about an average, per hour of working time. An average shift's work may be taken as about 500 sq. ft. Air consumption 1,000 to 3,000 cub. ft. free air per sq. yd. undercut.

In Great Britain, in the year 1943, there were 7,794 coal-cutting machines at work which cut 134,131, 691 tons of coal or 69 per cent. of the total output, as against (year 1941) 89 per cent. of the output of bituminous coal in U.S.A.

According to Mr. Mavor (*Trans. Inst. Min. Eng.*, vol. l., p. 641) a good compressor will compress 6 cub. ft. free air per minute to 60 lbs. per sq. in. per b.h.p. Tests showed coal cutters to use 143 to 1,146 cub. ft. free air per minute, or 620 to 3,642 cub. ft. per sq. yd., at mean pressure of 23 lbs. per sq. in. at machine and 68 lbs. at compressor.

COST.

Mr. Mavor gives average costs as follows:—

	Per shift. s.	d.	Per ton. d.
Interest 5 per cent. on 450*l.*	1	9	0·21
Depreciation 12½ per cent.	4	4	0·52
Labour	33	0	4 00
Air power	23	0	2·77
Picks, oil, etc.	8	4	1·00
Repairs	12	10	1·50
Total	83	3	10·00

In narrow work percussive coal-cutters are used, also chain-heading machines and arc wall machines. Percussive cutters supported on a fixed column are also used; they work with compressed air at 50 to 70 lbs. per sq. in., the air consumption being 100 to 300 cu. ft. of compressed air per sq. yd. undercut; different makes run at 300 to 600 blows per minute. A machine has undercut 30 sq. ft. per hour, but their usual performance is 80 to 100 sq. ft. per shift. Percussive cutters of the Ingersoll type on wheels are also used; these machines weigh 5 to 7½ cwts.; they use air at 70 to 90 lbs. pressure per sq. in., with an average consumption of 100 cu. ft. of free air per min., and strike about 150 blows per min.; they will undercut 15 to 50 sq. ft. per hour of running time, or 50 to 200 sq. ft. per shift.

The use of pneumatic picks both for getting coal and for making height in roadways has become a recognised feature of coal mining, especially in continental collieries. In the year 1938 nearly 13,000 such picks were in application in Great Britain for cutting coal or ripping. These picks are usually from about 17 to 27 lb. in weight, the consumption of air—according to size and weight—varying from 24 to 32 cub. ft. per minute. (A percussive hammer drill consumes at least 33½ cub. ft., whilst a light rotary drilling machine will consume probably 43 cub. ft.)

Chain heading machines may be driven electrically or by compressed air, using 10 to 20 h.p.; these machines can undercut about 50 sq. ft. per hour, but it is rare that 200 sq. ft. can be kirved per shift. See 'The Driving of Narrow Places,' by S. Walton-Brown (*Trans. Inst. Min. Eng.*, vol. lxxiii, 1927, p. 506).

Arcwall machines will cut 10 to 15 places per shift in roads 4 to 5 ft. wide, the usual depth of undercut being about 6 ft.; these machines weigh about 3½ tons, and their motors develop about 40 h.p.

Arc wall coal cutters as used in bord and pillar work are required to flit from place to place 10 to 15 times during a shift. They are mounted on rail-wheels, pneumatic tyres, or caterpillar tracks, and are electrically-driven. One cutting jib and gearhead is used for making vertical cuts and horizontal cuts at any horizon within the range specified, up to about 7 ft. Adjustment of height of cut is made by power-driven screw-jacks, or by hydraulic jacks. The usual depth of cut is 6 ft. Machines of this type weigh about 3·5 tons and are driven by a single motor of about 40 h.p. with clutch engagements to the various motions.

The Stanley header cuts roads circular in cross-section through coal, using 15 to 40 i.h.p. A road 5½ ft. in diameter can be driven through coal at the rate of 10 ft. per 8-hour shift, or six times as fast as handwork, the cost being sometimes rather less, sometimes rather more than hand labour, namely 2*s*. 6*d*. to 4*s*. per foot of advance, everything included, as against 2*s*. 8*d*. by hand.

The Jeffrey Entry driver cuts, breaks down, and loads the coal into tubs.

In the U.S.A. Longwall machines with jibs 40 ft. long cut and load 350 tons per shift. A machine with a 50 ft. jib is worked by five men, and in a 6 ft. seam with a strong sandstone roof cuts and loads 200 tons per shift or 400 tons per day; a machine with a 200 ft. jib, to cut and load 1,000 tons per shift is being built.

Increased mechanisation of coal getting in this country during the 1930's was accelerated by the introduction of American types of machines, especially loaders, during World War II. The N.C.B. aims to develop and apply such methods to the full. The American types of loaders applicable in our conditions are mostly for pillar and stall working, *e.g.* Joy loader, and Duck-bill loader. But it is thought that longwall machines must be developed to suit British conditions, and much has been done to that end.

Much the most successful at present is the A. B. Meco-Moore Cutter Loader. This machine is a development of the common chain coal-cutter; in fact it was originally conceived as two separate units, the loader to follow after the cutter unit. The first model was tried out in 1934, but not until 1941, under stress of war conditions, was intensive development undertaken which led to the trial in 1943 of the prototype of the machines now in use.

The cutter unit of the High Type machine is an ordinary 50 h.p. A. B. coal-cutter fitted with an extra chain jib driven through a gear train, and set to cut at about mid-height of the seam or higher, and immediately above the bottom jib; the latter is 6½ ft. long and makes a 7-in. kerf, and the upper jib is 5½ ft. long and makes a 5-in. kerf. Both jibs can be swung round to cut on either side of the machine, and the chains run and cut in opposite directions.

The loader unit is about the same size as the cutter unit and is attached to its rear end by a hinged coupling. This unit also helps in cutting, by means of a shear jib of triangular shape mounted in the vertical plain in line with the end of the shorter horizontal jib. The sheat jib makes a 5-in. kerf extending from roof to floor, which forms the new face of the longwall. This jib is pivotally mounted so that it can cut when the loader moves in either direction.

The loader elements of the loader unit comprise a short transverse conveyor, a riffle bar, and two gummers. The conveyor is of rubber armed with steel slats and side chains to give a positive drive. It is about 5 ft. long and set some 9 in. above floor level; it feeds the coal on to a normal face conveyor, usually of the bottom-belt type. The rotary riffle bar is mounted horizontally and immediately in front of the conveyor. It carries picks arranged on two helices which in turning break up the coal and lift it on to the conveyor. The gummers are of the screw or fan type and their function is to remove fines. The loader unit has one 50-h.p. motor with shaft drives to shear jib chain and conveyor, etc. Each motor has a separate trailing cable and controls.

In favourable conditions, *e.g.* seam fairly level (the limiting dip of face is about 1 in 4), coal amenable and cleat angle satisfactory, the cut coal collapses on to the conveyor under the action of the riffle bar. Under less favourable conditions in respect of coal or roof for example, overcutting and prior bursting or firing of the coal may be necessary. The *Low Type Meco-Moore Loader* is a facsimile of the High Type built low enough to work in a seam slightly more than 3 ft. thick. Although fitted with two horizontal jibs, the intention is not to use the top jib normally. This machine is 16 ft. 9 in. long by 3 ft. 1 in. wide by 2 ft. 0 in. high, or 2 ft. 7 in. with top jib, and weighs 10 tons. It is driven by two A.C. motors of 60 h.p. each, and takes a cut of 4 ft. 6 in., 5 ft. 0 in. or 5 ft. 6 in. at choice. Experience with the Meco-Moore machines is given in papers by Forrest Anderson, B.Sc.,* and by Messrs. Young and Sansom.†

The Meco-Moore Loader will cut in both directions along the face. But in order to do so it has to be partly dismantled and re-erected at each end of the face, in order to bring the jibs, etc., into correct relative positions again.

The operations involved in reassembling the machine at each end of a face requires considerable pit room. The preparation of these end places or stables is done by hand with ordinary coal cutters and the work often represents a considerable addition to the labour required to operate the cutter-loader itself. For example, a face 140 yds. long may comprise 20 yds. of stables. It is also considered a drawback of the Meco-Moore machine that such a high percentage of the coal got is reduced to gummings by the chain picks. A strong body of opinion holds that a machine using a wedging action would prove more satisfactory for getting and loading on longwall faces in British coals generally, and much thought and development work has been done to that end in recent years.

The Mavor & Coulson slabbing machine, now under development, is more akin to the German Coal Plough in that it shears the coal off the face by a wedging action, without a chain-type of cutter. It is in fact a plough type of machine designed to suit our coal, which is harder than German coal. This machine is hydraulically operated with electric power supply, and is self-propelling. It works on a buttock 2 ft. to 3 ft. wide, and shears the coal off the face by means of multiple wedges which are thrust forward, the machine itself being anchored meanwhile by a powerful vertical jack set between roof and floor. The whole thickness of seam is broken down by this means, and thrust over on to a normal face conveyor.

One Coal-Plough, as developed in Germany during the war, has been working in a suitable seam of soft coal in Durham for about two years, with fairly satisfactory results. This machine shears out the lower part of the seam, to a height of about 18 in. and is hauled to and fro by means of a rope and winches, taking an 8-in. or 12-in. cut each way. A specially constructed scraper conveyor is required, to serve the secondary function of holding the plough up to the face.

Another machine, not unlike the Meco-Moore in principle, has been developed by Messrs. Eickhoff of Bochum. It has two superimposed horizontal jibs ; but the lower one is bent upwards at the end of the horizontal portion, thus forming a vertical shear jib extending to the height of the upper jib. The machine travels on top of a shallow scraper-chain conveyor, and the coal, as it collapses is directed into the conveyor by a series of paddles. Several inches of floor coal left below the bottom jib have to be got up by hand.

In pillar and stall work the common method of mechanisation is to break the coal down first in the usual way with the aid of coal-cutters, and to load with separate machines, often of American make or design. Of these the Duckbill Loader and the Joy Loader are most widely used. Either type may load directly on to a conveyor, or into shuttle cars for transport to a central loading point. Both loaders nose their way into the broken coal at floor level, and the coal is gathered and drawn up on to a conveyor by means of gathering arms or by an oscillating action. The Samson Loader, made by Messrs. Mavor & Coulson is 2 ft. high and 4 ft. 10 in. wide, with gathering arms which sweep 5 ft. 2 in. wide. It is driven by one 30 h.p. motor and moves on crawler treads, and can load up to 4 tons per minute of coal or stone. The Distington Engineering Co., make a duck-bill loader-cum-shaker conveyor.

Air-driven mechanical shovels similar to a surface excavator are coming into use, chiefly for stone-work in headings. The Joy-Sullivan type, which is made in Great Britain, can load from 2 to 3 tons per hour. It is mounted in a four-wheeled truck about the size of a tub, and the bucket swings back over the truck to load the tub standing behind it. Similar shovels are made by the Consolidated Pneumatic Tool Co., and by Messrs. Eimco.

Conveyors have been extensively used for bringing the coal to the gateways and, more recently, along the gateways also. There are four main types : (1) troughs containing scraper-chains, either endless or reciprocating, (2) endless belts, (3) shallow trucks or trays hauled by ropes, (4) shaker conveyors, (5) scraper conveyors, hauled by various machines. The Blackett conveyor is an example of the first type ; the chain is 6-in. pitch and rests in a trough 12 ins. wide at the bottom, 18 ins. at the top, 9 ins. deep, made in 6-ft. lengths ; it will carry up to 30 tons per hour along faces 100 yds. in length. Most of the other forms have lower capacities, and are suited to smaller outputs or shorter faces. In 1943 there were 9,496 conveyors in use at British collieries carrying 66 per cent. of the total output as against (year 1938) 25·8 per cent. in U.S.A.

See 'Colliery Machinery and its Application' (R. A. S. Redmayne), chap. 'Machine Mining of Coal.'

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* 'Some Developments in Simultaneous Cutting and Loading on Longwall Faces.' *Trans. Inst. Min. Eng.*, vol. cvii., Pt. 5, 1948.

† See also 'Simultaneous Cutter and Loader for Longwall Mining,' by T. E. B. Young and W. H. Sansom. *Trans. Inst. Min. Eng.*, vol. civ., 1945.

Working Costs in Getting Coal.

***Bord and Pillar.*—Working 'brokens' (in pillar workings) is usually about 25 per cent. cheaper than working in the whole coal.**

PERCENTAGES OF ITEMS IN PRICE OF COAL AT PIT'S MOUTH.

Year.	1944.	1913.	1901.	1866–96.	1700–50.
	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.
Royalty . .	1·47	4·05	7·96	8·5	12·5
Wages . . .	68·25	62·50	67·37	66·0	37·5
Materials and Administration .	*25·86	*19·43	16 54	17·0	25·0
Interest and Profit .	4·42	14·02	8·13	8·5	25·0
Total . . .	**100·00	*100·00	100·00	100·00	100·00

* These are inclusive of Insurance, Workmen's Compensation, Depreciation, Repairs and Renewals, Clerical and Administrative Staff, Coal and power purchased, Local Rates, Welfare levy (instituted in 1920), and General Expenses amounting in all to 9·58 per cent. in 1913 and 14·72 per cent. in 1944. The balance is in respect of stores and timber, namely 9·85 per cent. and 11·14 per cent. for 1913 and 1944 respectively.

** The 1913 figure was 11*s.*, representing the average proceeds per ton from the sale of coal. The 1944 figure was 34*s.* 7·35*d.* made up of 33*s.* 2·07*d.* average proceeds per ton from the sale of coal and 1*s.* 4·48*d.* net contribution from the Coal Charges Account.

Costs are given as follows:—

	Average for							
	1913.†		Year ending Mar. 31 1920.†		1922.		1923.‡	1924.
	s.	*d.*	*s.*	*d.*	*s.*	*d.*	*s.*	*s.*
Labour . .	6	4	19	7¼	12	1	11·90	13·24
Timber and stores		0	3	10	2	4	1·97	1·99
Other costs. .	0	11	1	6			2·72	2·86
Administration, etc. . .	—		0	1	3	2½	—	—
Control and contingencies .	—		0	2			—	—
Royalties . .	0	5½	0	6½	0	7	0·42	0·44
Capital adjustments . .	—		0	4	—		—	—
Owners' profits .	1	5	1	2	0	11½	0·53	0·54
Pithead value .	10	1½	27	3¼	19	2	17·54	19·07

	1938.‖		1944.‡‡	
	s.	*d.*	*s.*	*d.*
Wages .	10	6	23	8
Stores and Timber	2	3	3	10
Royalties		6		6
Miners' Welfare Fund		1		1
Other costs	2	9	5	2
Total costs	16	1	33	3
PROCEEDS—				
Commercial disposal	17	4	33	3
Miners' coal		1		2
BALANCE—				
Debits .	—		—	
Credits .	1	4	1	4
BALANCE ADJUSTED				
Credits .	1	4	1	6

† *Third Report of Fuel Economy Committee, Brit. Assoc.*, 1920.
‡ Royal Commission on Coal Industry (1925).
‖ From the Annual Statistical Summary for the year ended December 31, 1938, issued by the Department of Mines. These summaries ceased to be issued during the War period, but were resumed in 1945.
‡‡ The Coal Industry being under a system of Government semi-control.

The Support of Workings.

Supports of various kinds are used to achieve one or more of these objects given in order :—

(a) To maintain ways and working room in a condition and large enough for their purpose.
(b) To prevent falls of ground, both as a source of injury to personnel and of obstruction to movement.
(c) To control the closure of the strata subsequent to mining, and to utilise their energy as far as possible in the extraction of the mineral.
(d) To reduce in amount and to mitigate the effects of surface subsidence.

Although (a) and (b) are quite distinct functions they are often fulfilled at once by the same supports. A typical instance is that of steel arches as used to maintain some 5,000 miles of roads in British collieries. Their strength is infinitesimal compared with the weight of cover at, say, 4,000 ft., yet they provide resistance sufficient to support the surrounding annulus of rock and maintain the road in a safe condition and of size adequate for its purpose. Steel arches have proved the best solution yet found for this most difficult problem, largely because they combine strength with flexibility and capacity to offer continuous resistance. In metalliferous mines the roads are more often made in strong rock, which may not be disturbed by other mining operations. In such cases a concrete lining may be used for safety (a) and to reduce air resistance (b), at depths as great as 6,000 ft. or 8,000 ft.

With regard to (c) a good instance is in the method of longwall working of coal as practised in this country and on the Continent. The face supports serve the functions (a) and (b) and prevent the premature collapse of the beds immediately overlying the seam. The packs or other permanent supports, starting some 12 to 15 ft. or more from the face, slowly acquire the full weight of the overlying beds. Both the face and permanent supports yield as the face advances, usually to between 50 and 80 per cent. of the seam thickness at 100 yds. from the face. Hence the coal face is subjected to vertical loading, as it were by a huge lever. Subject to consideration of the strength of roof and floor and of the coal itself, the method of support can be adjusted to suit the system of getting and the type of machines used for that operation.

Permanent supports, distinct from mineral left *in situ* to protect the surface or shaft, etc., are usually built of mine waste and the like in collieries, with or without some stiffening or reinforcement. In metalliferous work filled or empty timber cogs or mats are used. In this country packing is almost entirely done by hand, in the form of strips advancing with the face and filling upwards of 50 per cent. of the waste. Attempts to mechanise this bulk movement of material are in progress, using German type of equipment.

Stowing by Machine.

Experience with methods of packing or stowing (other than by hand) is more extensive on the Continent than in Great Britain ; in Germany this was in part due to more severe regulations in respect of surface subsidence. The methods tried include hydraulic, pneumatic and mechanical stowage. In the former, the water proved troublesome, and this method has now only limited use, chiefly in Silesia. In the Ruhr 130 pneumatic stowing plants were in use at 52 collieries in 1944.

The choice of method is influenced by mining conditions, in particular by the packing material available and the percentage of stowing required. Solid stowing is possible by the pneumatic method, thus limiting subsidence to about 50 per cent. of the initial height of pack, as compared with 60 per cent. for solid hand packing.

In a typical pneumatic stowing machine as used at Bulcroft Colliery* mine waste from various parts of the pit is passed through a crusher and then dedusted, damped and fed to the blower which causes it to be conveyed along the pipe line and ejected through a nozzle into the pack. The nozzle is directed by one or two men, and shuttering or brattice is attached to props to control the size of pack.

The driving air is obtained from the compressed air mains. A pressure of 30 to 50 lb. per sq. in. is required, depending on the nature of the material. The velocity of the air stream increases between inlet and exit from about 125 to 400 ft. per second. The stowing material lags behind the air velocity by about 30 to 100 ft. per second depending on its size and nature ; some is air-borne, some rubs or rolls along the pipes. In most cases cast-iron pipes of 6 ins. to 8 ins. diameter are used, with quick-clip joints ; the elbows may have to be protected by internal wearing pads of stone or other hard substance, if the material is abrasive. Depending on size of pipe and its length and type of material, air consumption lies usually between 2,000 and 5,000 cub. ft. per cub. yd. of stowing. Delivery up to 300 yds. is normal, and 600 yds. is practicable.

Material up to 3 or 4 ins. can be conveyed, but smaller maximum sizes and a fair grading gives the best results. Up to 2,500 cub. ft. per hour can be stowed with suitable material, but only half that amount with a sticking or clogging type of material. The total cost of pneumatic stowing is about 6*d*. to 1*s*. 6*d*. per ton of coal mined, chiefly due to wages, air, repairs and maintenance of plant.

* 'Pneumatic Stowing at Bulcroft Main Colliery,' by J. Hunter. *Trans. Inst. Min. Eng.*, vol. cv., Pt. 3, 1945.

An alternative method of stowing is wholly mechanical. Suitably crushed material is fed on to a high-speed belt in such a way that it acquires a comparable speed before being flung out of the machine in the direction required. Wet or dry material up to 3-in. size can be cast 30 ft. by machines now under development.

The advantages of stowing by machine are denser packs with less labour and total cost. The greater density has proved highly beneficial in reducing air leakage and the incidence of spontaneous combustion. In so far as mechanisation facilitates solid stowing it reduces the amount and ill-effects of surface subsidence.

The resistance developed by hand-built packs has been investigated with mechanically-operated dynamometers let into the middle of a piece of floor before that part of the pack was built. In general the curve of load plotted against face advance increased slowly at first, and then more rapidly until it reached a peak value. This occurred at 50 to 60 yds. from the face in seams at shallow depths, 200–300 ft. and at 70 to 80 yds. in seams at moderate depths, 1,000 to 2,000 ft. A considerable fall in load then occurred, followed by a further increase until a steady load was reached, usually somewhat less than the first peak value. The final stable loads measured on strip packs occupying about 50 to 60 per cent. of the waste were usually 1·5 to 3 times greater than those estimated for the depth of cover acting on the unmined seam.

Compression at the centre of the pack started at a high rate, but it was much reduced when the load reached peak value, and subsequently reached a stable value between 30 and 40 per cent.*

Stowing by means of filled bags made of jute or wire-netting is used extensively at some British collieries, chiefly for building the gate-side pack-walls when large stones are lacking, and in conjunction with pneumatic stowing. The bags are sent in filled with washery waste, or they are filled on site by hand, using a simple frame to hold the bag in position. Used in gate-side walls the bags have the advantage that steel arches can be erected on the walls in such a way that the initial subsidence of the road can be accommodated without damage to the arches.†

TIMBER PROPS.

The strength of timber props depends largely on the species of timber and its condition of seasoning, *i.e.* moisture content; crookedness and decay can weaken them incalculably, and knots are deleterious. As for all struts, the slenderness ratio (length over radius of gyration, or 4L/D for round props) is also a critical factor. The strength of props with slenderness ratios of 10 or more (*i.e.* L = 2·5D) is less than that of short specimens in which L = 2D. Compression tests on such short specimens of clear timber provide the basic strength figure f_c, or fibre crushing stress, as given below for some typical softwoods and hardwoods.‡

Species.	f_c, lb. per sq. in.	$f_c \times 75$ per cent.
SOFTWOODS.		
Larch, European	3,500	2,600
„ Japanese	2,800	2,100
Fir, Douglas	3,300	2,500
„ Silver	3,000	2,250
Pine, Corsican	2,700	2,100
„ Scots	3,000	2,250
Spruce, European	2,600	2,000
„ Sitka	2,300	1,700
HARDWOODS.		
Ash	3,900	3,000
Beech	3,700	2,800
Oak	3,800	2,850
Chestnut (Sweet)	3,400	2,500

These values of f_c are for unseasoned timber, *i.e.* having a moisture content exceeding 30 per cent. of the dry weight of wood. Air-dry timber, seasoned to a moisture content of 12 per cent., is about twice as strong, *i.e.* of the order of 5,000 to 7,000 lb. per sq. in. for softwoods and 7,000 to 8,000 lb. per sq. in. for hardwoods.

For good average quality props of the usual slenderness ratios the maximum stress f developed is considerably less than the fibre crushing strength. This loss is due to the effects of slenderness, bow, and knots. The maximum prop stress f varies from 60 to 80 per cent. of the fibre crushing stress f_c for props of 60 to 40 slenderness ratios. For average props of 48 slenderness ratio, *i.e.* L = 12D, the value of f may be taken as 75 per cent. of f_c.

In the result a useful approximation for the strength of unseasoned softwood props is 1 ton per sq. in. of area; and for unseasoned hardwood props, 1·25 tons per sq. in.

* 'An Investigation of the Loads on Packs at Moderate Depths, II,' by W. H. Evans and T. J. Jones. *Trans. Inst. Min. Eng.*, vol. cv, Pt. 6, 1946.

† 'Underground Packing by Means of Bags,' by David Jeffreys. *Proc. S.W.I.E.*, vol. lii., 1935.

'Wire-netting Bags for Packing Purposes,' by L. C. Timms. *Coll. Gdn.*, June 1942.

‡ See 'A Handbook of Home-grown Timbers.' D.S.I.R. H.M.S.O.

Experience in testing large consignments of timber props shows that in any one group of 20 or more similar props an overall variation in strength of 2 to 1 may occur ; *i.e.* from 70 per cent. up to 140 per cent. of the mean strength.

The strength of timber in compression across the grain, as used in cogs, chocks and cribs, etc., depends on the species of timber and its condition, on the initial areas of contact between adjacent pieces and on the amount of compression or yield sustained.

Chocks.

Chocks built of sawn softwood 4 ins. to 6 ins. square will carry loads producing stresses of 300 to 450 lb. per sq. in. of contact area when compressed about 3 per cent. The stress increases to a maximum of 600 to 800 lb. per sq. in. at about 20 per cent. compression. For chocks built of sawn hardwood the corresponding stresses at 3 per cent. and 20 per cent. yield are of the order of 800 to 1,000 and 1,500 to 1,750 lb. per sq. in. respectively. Steel chocks are now widely used.

Cogs.

Cogs built of round softwood 6 ins. to 8 ins. in dia. two sticks to each layer and 6 ft. square have a linear load-yield curve, reaching 40 tons at 20 per cent. and 60 tons at 30 per cent. yield. Similar cogs filled with mine rock also have a linear load-yield curve but are immensely stronger. Depending on the strength of the rock and care in building loads of 200 to 400 tons are sustained when the yield is 20 per cent.

Steel Props.

Steel props are much more reliable than timber props ; they are usually of H sections which are often specially rolled with broad flanges and thick webs. A usual ratio of length in feet to width in inches is 1·5 to 1, as compared with unity for timber props. Eccentric loading is much reduced by folding the flanges to close and round the ends of the props, by the use of timber lids, and the relatively softer floor. Depending on the quality of steel, *i.e.* whether of 30, 40 or 50 tons approximate ultimate tensile strength, a prop will carry a maximum load of about 17, 21, 24 tons per sq. in. or about 5, 6·2, or 7 tons per lb. weight per ft. of section. After maximum load a steel prop will buckle but still carry about half its maximum load when the centre deflection is equal to the width of section.

Special Steel Props.

Several types of mechanically-operated steel props are used successfully under particular conditions, both in this country and abroad. One type is merely a screw-jack which is convenient as a temporary support in conjunction with power loaders. On the continent props capable of adjustment in height, tightening in position, yield under a suitable load, and release, depend usually on frictional resistance between telescoping parts. High transverse forces are required to provide suitable resistance, and this implies a strong wedge box and a heavy prop. The G.H.H. is a prop of this type used with the German coal plough, and it weighs 25 lb. per ft. for a prop 3 ft. long. In this country the most successful mechanical props are not adjustable or yielding ; they merely have capacity for tight setting, with or without means of quick release from a safe position. Of the former class the Bathgate prop is typical ; it comprises a suitable length of H steel section with a cast steel head which is raised or lowered about 1 in. by means of a transverse wedge, also of cast steel. The Sylvester prop is similar, but has the additional feature that on drawing a catch the head collapses and the prop is ejected.

Dowty Hydraulic Props.

An important recent introduction is the hydraulic prop, which after some three years' intensive development by Dowty Mining Equipment, Ltd., is being used in extensive trials in the East Midlands. It appears to have more than fulfilled the economies anticipated in reduced costs for material and labour underground, and its use is likely to become widespread. The prop comprises two steel tubes filled with oil, the lower one forming the working cylinder. It is extended by forcing oil out of the upper tube (or ram) into the cylinder by means of a pump built into the ram. The pump is operated by a detachable handle engaging a crankshaft carried in a housing in the side of the ram. Thus the prop can be extended 2 or 3 ft. if required, and a tight setting obtained.

At a predetermined load of about 20 tons the prop yields by passing oil from the cylinder back to the upper tube. This is done via an internal pipe connecting the piston head of the ram to a spring-loaded control valve set in a housing near the top of the prop. This same valve can be pulled off its seat by hand (using a cable or rod hooked to a shackle in the valve housing) in order to release the load on the prop, which can then be withdrawn. The prop weighs about 13 lb. per ft. and may be had in practically any length. The present price is £10 for props of normal length, but this may be reduced to £7 10*s*. when manufacture on a suitably large scale can be begun.

The economies effected by the Dowty hydraulic prop are due to the saving in time in setting and withdrawing operations, and economy in supplying lids and new props, and in restraightening bars. The improved roof control and increased safety are likely to yield equally satisfactory returns.

Steel Arches.

The specification for ' steel arches for use in mines ' given in B.S.S. 227 (1934) is now outdated in several respects. Considerable quantities of arches are now made in steel of about 40 tons

ultimate tensile strength, in addition to those of 30 tons and 50 tons specified. Also many new sections have been introduced, some of them designed for this specific purpose; and of the three shapes of arch specified, the horse-shoe shape proved inferior and is not now in use.

The supporting value of a steel arch depends on the strength of the section, on the efficiency of the strutting between arches, and on the size of road and mode of incidence of the ground pressure. The strength of the section varies with the weight per ft., the overall dimensions, and on the quality of the steel.

The actual strengths of 8 ft. diameter straight-legged arches when subjected to load fairly well distributed over the crown, and with the sides firmly supported, varies greatly for the reasons given above. Arches of 50 tons tensile steel and 4 ins. by 2·5 ins. by 2 ins. section weighing 11 lb. per ft. and erected with mediocre strutting carried 27 tons per arch maximum load, equivalent to about 3·3 tons per ft. diameter of arch or 1 ton per sq. ft. of roof area with arches set at 3 ft. crs. The strength of arches of this section was increased 2·6 times, to 70 tons, when the arches were well strutted, as by a continuous brick lining 3 ins. thick, built between the flanges, or by steel struts and corrugated sheet lagging. Arches of 5 ins. by 4½ ins. section 18 lb. per ft. wt. and rolled in 30 tons steel set with mediocre strutting carried 56 tons per arch, and 108 tons or nearly twice as much when well supported by continuous brickwork lining 4½ ins. thick. In this case the increase in strength given by good strutting was not so large as in the case of arches of the slimmer section.

The actual strengths are perhaps of less interest than relative strengths in terms of the intensity of ground pressure resisted. This probably varies inversely as D^2, D being the diameter or width in feet; if p is the ground pressure, then the relative strengths are as follows:—

$D =$	8	10	12	14	16	18	20
$p =$	100	66	44	33	25	20	16

Under normal conditions on roads 14 ft. wide well-lined arches of 5 ins. by 4½ ins. section and of mild steel should carry up to 36 tons per arch, which is equivalent to the dead weight of 12 ft. of rock, on arches set at 3 ft. crs. Under similar conditions arches of 4-in. by 2-in. by 2½-in. section in high-tensile steel should carry 8 ft. of cover.

These figures for actual depth of ground or strata supported may appear slight, but they are of the same order for face supports.* As pointed out above, in relation to the weight or thickness of superincumbent strata the strength of most supports is negligible in itself. But by judicious application they can provide that stabilising influence which enables the fractured strata to cohere sufficiently to prevent collapse. In the case of steel arches the annulus of ground around the road which they control may be well able to support both itself and the weight of higher beds as well. Not infrequently the strength of the floor of the road proves the limiting factor.

Colliery Consumption.

According to the Coal Conservation Committee of the Ministry of Reconstruction, 1918, colliery consumption in Britain varies from 5 to 9·43 per cent., averaging 6·8 per cent., of which 6·2 per cent. is produced by the direct combustion of coal under boilers, and the balance from outside sources. In 1923 colliery consumption amounted to 6·11 per cent. and miners' coal to 2·34 per cent. of the output, and in 1924 to 6·21 per cent. and 2·46 per cent. respectively. The average colliery consumption at British collieries at the present time (1947) is estimated to be 6 per cent. At a modern colliery at which not much water is made the colliery consumption should not exceed 3 per cent. of the output.

Mining in Veins and Masses.

Methods employed in veins are:

- Underhand stoping.
- Overhand stoping.
- Combined underhand and overhand stoping.
- Rill stoping.
- Shrinkage stoping or magazine mining.

Irregular deposits, wide veins, thick and steeply inclined seams may be worked by the following methods:—

- Lost pillars.
- Vertical slices.
- Horizontal slices taken transversely.
- Horizontal slices taken longitudinally.
- Top slicing.
- Caving.
- Magazine mining.
- Square set timbering.

Of these, overhand stoping is the most general, underhand stoping being only used in quite special cases, as when there is only a small quantity of payground to be taken below a given level. Overhand stoping can be used for veins of medium width, say up to 10 ft. on the average.

* See 'The Strength of Undermined Strata,' by W. H. Evans, Ph.D., A.M.I.Mech.E. *Trans. Inst. Min. and Met.*, *Vol. L.* 1940–41.

Rill stoping is mainly employed in Western Australia. Magazine mining suits veins of any width, provided that they are not too flat, that the walls are fairly good, and that the whole of the veinstuff has to be extracted. Slicing and caving are used in wide veins and irregular masses when the value of the mineral is not great and where cheap methods of working are essential; it is largely used in the hæmatite mines of Cumberland and Lancashire. Square sets can only be used where the mineral is valuable and where large timber is abundant and cheap.

The following classification has been drawn up by the Committee on Mining Methods of the *Amer. Inst. Min. Met. Engs.*, vol. lxxii. 1925.

- **Open cut with or without stripping**
  - Benches
  - Glory holes and milling
  - Placer mining
    - Hand sluicing
    - Dredging
    - Hydraulicing
- **Underground methods**
  - Breast stoping (tabular flat-dipping deposits or beds)
    - Continuous horizon
    - Casual pillars
    - Room-and-pillar
  - Underhand stoping (veins and large masses)
    - Glory hole and milling
    - Open stope
      - With pillars
      - With stalls
      - With square set timbering
  - Overhand stoping (steep-dipping veins or masses)
    - Open stope
      - With pillars
      - With stalls
      - With square set timbering
    - Filled stope
      - Horizontal cut and fill (flat-back)
      - Inclined cut and fill (rill)
      - Square set timbering
      - Piercing or stripping
    - Shrinkage stope
    - Sub-level stopes
  - Top slicing (wide veins or masses)
    - Continuous horizon
    - Panel slicing
    - Block slicing
    - Inclined slicing
  - Caving (large masses)
    - Sub-level caving
    - Block caving
      - Undercut from main level
      - Undercut from sub-levels
      - Back caving into chutes
      - With branch raises
- **Combined methods (large masses)**
  - Alternate shrinkage stope. and pillar
  - Pillars mined by: top slicing, sub-level caving, or block caving
  - Square set timbering: pillars mined by slicing

(See 'The Working of Unstratified Mineral Deposits' by G. J. Young, 1927.)

The costs of stoping vary within very wide limits, and may be illustrated by typical examples.

In Western Australia, with the contract system in force, a miner breaks 7·31 tons of ore per shift. Formerly a miner on day's pay broke only 4 tons per shift.

In eleven of the West Australian mines treating from 9,000 to 22,000 tons of ore per month with veins varying from 4–20 ft. to 6–60 ft. in width the stoping costs per ton of ore range from *6s. 4d.* to *9s.*, averaging *7s.* This includes breaking ore, filling stopes, trucking, hoisting, etc.

On the Rand it costs from 3*s*. 6*d*. to 5*s*. 6*d*. to stope a ton of milling ore in stopes 40 to 60 inches in height. To these figures must be added cost of compressed air, rock drill maintenance, steel sharpening, and administration underground. The first three factors amount roughly to about 1*s*. per ton, while the last would not exceed 1*d*. In some Rand mines one man can break 12 tons per shift per machine, and uses one case (50 lbs.) of blasting gelatine to break 3·5 fathoms of ground. At other mines only 7 tons are broken per man and one case of explosives is used to break 2·5 fathoms.

In Nevada at the Silver Peak Mines each miner breaks 8·6 tons of rock per shift. The total stoping cost is 5*s*. per ton. Miners here receive 16*s*. per shift.

At the Braden Copper Mines native miners break 12 tons of ore per man per day with rock drills in moderately hard rock. The total cost per ton of ore broken, including superintendence and general charges, is given as 1*s*. 9*d*.

In Utah in quartzite with a stoping width of 10 ft., square set system, small air drills, the stoping cost per ton varies from 6*s*. to 8*s*.

At Douglas Island, Alaska, the ore is a mineralised igneous rock of fair hardness, the workable portions of which vary upwards from 30 ft. in width. Wages 12*s*. Machine used 3¼ inches piston diameter. The following figures are taken over a year's working :—

COST OF BREAKING GROUND AT DOUGLAS ISLAND, ALASKA.

General Details.	Ready Bullion.	700 Ft.	Mexican.	Treadwell.
Average number of drills used	10·12	11·586	14·917	39·13
Tons ore broken	151,460	238,696	232,875	830,551
Tons waste broken	2,756	—	10,509	1,670
Tons broken, total	154,216	238,696	243,384	832,221
Total footage drilled	241,679	269,990	362,519	803,542
Tons ground broken per machine per shift:				
Stoping	47·36	42·998	41·26	—
Developing	8·65	11·323	6·58	—
Cutting out	14·30	7·334	10·34	—
Average	23·06	30·41	24·19	28·47
Footage drilled per machine per shift:				
Stoping	29·39	33·587	33·50	—
Developing	39·14	37·78	39·71	—
Cutting out	40·11	33·954	34·79	—
Average	36·12	34·37	35·7	29·41
Tons broken per foot of hole drilled:				
Stoping	1·61	1·28	1·23	—
Developing	0·22	0·19	0·165	—
Cutting out	0·31	0·33	0·30	—
Average	0·64	0·88	0·669	0·96
Costs of breaking ground per shift per machine:	£ *s*. *d*.	£ *s*. *d*.	£ *s*. *d*.	£ *s*. *d*.
Labour	1 14 0	1 8 0	1 10 0	1 9 0
Explosives	9 8	11 4	12 2	11 4
Supplies and power	1 0 0	18 2	1 0 0	1 0 0
Total	3 3 8	2 17 6	3 2 2	3 0 4
Cost per foot of hole drilled	1 10	1 8	1 9	2 0
Cost per ton of ground broken	2 10	11	2 8	2 1
Total cost of mining, including development	4 0	4 9	4 10	4 6
Percentage of total mining cost	2 9	8	2 2	1 11

In the Granite Gold Mine, B.C., on a vein of hard quartz from 8 inches to 5 feet wide dipping 35° to 50°, the stoping costs per ton are as follows :—

	s.	*d*.
Stoping by hand	14	0
„ by ¾-in. piston drills	11	6
„ by 2¼-in. „ „	8	4
„ by 2-in. hammer drills	6	4

Mining costs at the Elko Prince Mine, Nevada, in 1917, stoping a 2 ft. 6 in. vertical vein of hard quartz by magazine mining, using hammer drills with 1¼ in. cruciform steel, were as follows:—

	Per Ton of Ore
	$
Timbering, including ore shoots	0·5588
Blacksmith	0·0995
Stoping	0·8083
Tramming	0·2822
Hoisting	0·1956
Compressed air	0·1288
Explosives, detonators and fuse	0·4452
Repairs	0·0486
Drill steel	0·0570
Supervision, taxes, assaying, etc.	0·4987
Cost per ton	$3·1227 = 13s.

(*Bull. Amer. Inst. Min. Eng.*, Aug. 1918.)

Costs of stoping in 1903 in two mines on Cripple Creek, Col., in narrow veins: machinemen, $4; miners, $3 for 8 hour shift; coal, $4·60 per ton; timber, $20 per mil.

	I.		II.	
Machinemen and miners	$0·34		$0·369	
Shovellers and trammers	0·41		0·401	
Timbermen	0·23		0·215	
Total labour underground		$0·98		$0·985
Compressed air, rock drills, sharpening		0·15		0·179
Tramming, trams, etc.		0·03		0·034
Explosives		0·13		0·139
Timber		0·28		0·310
Hoisting and surface transport		0·23		0·254
Miscellaneous stores		0·04		0·291
Supervision and general expenses		0·23		
		$2·07 = 8s. 7½d.		$2·192 = 9s 1½d.

Drilling.

Ordinarily two men can drill double-handed 25 to 33 ft. of 1-inch holes per day in limestone, 4 to 8 ft. in granite. With double hammers 10 ft. of 2-inch holes can be drilled in medium rock.

On the Rand generally one white man and five Kaffirs run two air-drills; in some cases one white man and seven Kaffirs run three drills. Eight 6-ft. holes is considered a shift's work for two machines; these holes break on an average two fathoms of ground. The cost is as follows:—

	£	s.	d.	
One white man	1	5	0	
Five Kaffirs		15	0	
Power for two machines		16	0	
Maintenance		4	0	
Total	3	0	0	per fathom.

In Chili, with native labour, contracts are let on the basis of the number of holes drilled. The rock is moderately hard.

By hand.—1,008 man-days, by single hand work, drilled 13,855 ft. at a cost of 4·1*d.* per foot. Average amount drilled per man per day 13·8 ft.

By machine.—The actual cost of drilling for one month was as follows:—164 man-days (9 hours) with two ¾-in. air drills, drilled 5,082 ft. at a cost of 1·01*d.* per foot. Average amount drilled per man-day, 31 ft.

It is well known that drilling in siliceous rocks produces a dangerous dust, recognised as the origin of Miners' Phthisis. The amount of dust suspended in the air is tested by the Kotzé konimeter; the atmosphere is considered safe if there are less than 300 dust particles per c.c. (Final Report of Miners' Phthisis Prevention Committee, Pretoria, S. Africa; see also recent papers on Silicosis in *Trans. Inst. Min. Met.*)

The Safety in Mines Research Board has devised a filter bag connected by a flexible tube with the hole which is being drilled; this catches the particles of dust produced in drilling. (Paper No. 28, 1926.)

See 'Silicosis and its Prevention,' Haldane, Haynes, Shaw and Graham. *Trans. Inst. M.E.*, vol. xcvii., 1939. 'Dust-suppression (Pneumokoniosis) Experiments at a Kent Colliery,' by F. H. Price. *Trans. Inst. M.E.*, vol. cv., 1945, and Annual Reports of S.M.R.B., B.C.U.R.A.

Mechanical Drilling.

Drills either rotary or percussive. Rotary drills mostly of the twist-drill type confined to the softer rocks; Steavenson electric drill averages 40 ins. per minute in ironstone, uses 6 h.p. Siemens and others make small electric drills, taking 1 to 2 h.p., boring 6 to 10 ins. per minute in coal. For boring in hard rock the Brandt machine has been used in the Simplon tunnel and in some German collieries; hydraulically driven under head of 1,500 lbs. per sq. in.; bores 15 ft. per hour in gneiss. The Kellow drill, used in North Wales, is similar; has drilled 5 ft. per minute in slates.

Percussive drills are almost exclusively worked by compressed air. The drills are of two kinds: piston drills, in which the drill bit is rigidly connected with the piston rod, and hammer drills, in which the piston rod strikes on the end of the drill bit, the latter being lighter, smaller, and running at much higher speeds.

Drills may be classified by their valve action:—

1. Valve moved by external tappets.
2. " " internal "
3. " " air with auxiliary valve moved by tappet.
4. Valve moved by air.
5. Valveless, the piston itself acting as a valve.

Most modern drills belong to Class 3, but several types belonging to Classes 2 and 4 are also in use; very many hammer drills belong to Class 4.

Piston drills are further classified by their piston diameter, ranging from 2 to 4 ins.; 2¼-in. drills are largely used for stoping, 2¾ to 3¼ ins. for drifting, 3 to 3½ ins. for sinking. In 1902 70 per cent. of the drills in use on the Rand were 3¼ drills. Such drills use 60 to 90 cu. ft. of free air per minute at pressure between 60 and 90 lbs. per sq. in. A 3-in. drill evolves about 1¾ h.p., for which about 5 h.p. must be applied to the drill piston and 10 to 25 h.p. to the steam end of a direct-driven air compressor.

At the Rose Deep Mine, Transvaal, a trial run of 31 drills for 6 hours gave air consumption for 3¼-in. drills 81 cu. ft. compressed air, free air compressed to 70 lbs. pressure per minute; each drill required the consumption of 43 lbs. coal per hour to supply this air, 3·4 lbs. coal developing 1 h.p. hour. The average power of the compressor engine was 12·7 i.h.p. per drill; average drilling speed 4·5 ft. drill hole per drill per hour.

Relative air consumption of drills: –2¼-in. drill, 0·445; 3¼-in. drill, 1·000; 3 5/16-in. drill, 1·069; 3⅜-in. drill, 1·123.

Average speed of boring, including setting up, is 10 to 25 ft. per 10-hour shift. Costs of boring in hard rock in Sweden are about 8*d*. to 10*d*. per foot bored, of which 40 per cent. is for wages and 30 per cent. for power; average boring speed is 5 ft. 6 ins. to 8 ft. per hour for downward holes and 4 ft. to 6 ft. 6 ins. for upward holes.

Hammer drills use 25 to 50 cubic feet of free air per minute; speed of boring about the same as piston drills, but holes are usually smaller. Hammer drills with a relatively rapid turning action of the bit and using bits with a chisel edge, but specially twisted steel, are used very successfully in coal. Hammer drills are replacing piston drills for stoping, etc.

A complete 6-piston drill plant costs about 500*l*.

Two-stage compressors, delivering air at 100 lbs. gauge pressure weigh on the average 27 lbs. and cost on the average 12*s*. in the U.S.A. per cubic foot piston displacement.

In the case of small quarry plants or contract work, where drilling is done in the open, dry steam at a pressure of from 80 to 100 lbs. can be used, but will not give such good results as compressed air. Where a number of drills are used, a compressed air plant should preferably be installed, and is indispensable in underground drilling. With a properly laid pipe line of suitable size, there is very little loss in conveying compressed air from a central plant over very large distances.

Detachable drill bits are said to give good results at several large mines, saving in cost of transport of drills. (*Trans. Amer. Inst. Min. Eng.*, 1930 Year-book, p. 139.)

Sharpening Steel Bits.

A good blacksmith and striker can sharpen 30 to 40 bits per hour.

By the use of power drill-sharpeners, many of the shortcomings attendant upon the hand-sharpening process are got rid of, and drills can be kept true to gauge, with the result that where these machines are used it is possible to accomplish more drilling, whilst the cost of sharpening is considerably reduced. There are a number of drill-sharpening machines on the market.

TABLE I.—CUBIC FEET OF FREE AIR REQUIRED TO RUN ONE DRILL.

Gauge Pressure.	Size and Cylinder Diameter of Drill.												
	A35	A32 A86	B	C	D	D	D	E	F	F	G	H	H9
	2 ins.	$2\frac{1}{4}$ ins.	$2\frac{1}{2}$ ins.	$2\frac{3}{4}$ ins.	3 ins.	$3\frac{1}{8}$ ins.	$3\frac{3}{16}$ ins.	$3\frac{1}{4}$ ins.	$3\frac{1}{2}$ ins.	$3\frac{5}{8}$ ins.	$4\frac{1}{4}$ ins.	5 ins.	$5\frac{1}{2}$ ins.
60	50	60	68	82	90	95	97	100	108	113	130	150	164
70	56	68	77	93	102	108	110	113	124	129	147	170	181
80	63	76	86	104	114	120	123	127	131	143	164	190	207
90	70	84	95	115	126	133	136	141	152	159	182	210	230
100	77	92	104	126	138	146	149	154	166	174	199	240	252

TABLE II.—MULTIPLIERS TO DETERMINE CAPACITY OF COMPRESSOR REQUIRED TO OPERATE FROM 1 TO 70 ROCK DRILLS AT ALTITUDES COMPARED WITH SEA LEVEL.

Altitude Above Sea Level.	Number of Drills.																		
	1	2	3	4	5	6	7	8	9	10	12	15	20	25	30	40	50	60	70
	Multipliers.																		
0	1·0	1·8	2·7	3·4	4·1	4·8	5·4	6·0	6·5	7·1	8·1	9·5	11·7	13·7	15·8	21·4	25·5	29·4	33·2
1,000	1·03	1·85	2·78	3·5	4·22	4·94	5·56	6·18	6·69	7·3	8·34	9·78	12·05	14·1	16·3	22·0	26·26	30·3	34·2
2,000	1·07	1·92	2·89	3·64	4·39	5·14	5·78	6·42	6·95	7·60	8·67	10·17	12·52	14·66	16·9	22·9	27·28	31·46	35·52
3,000	1·10	1·98	2·97	3·74	4·51	5·28	5·94	6·6	7·15	7·81	8·91	10·45	12·87	15·07	17·38	23·54	28·05	32·34	36·52
4,000	1·14	2·05	3·08	3·88	4·67	5·47	6·15	6·84	7·41	8·09	9·23	10·83	13·34	15·62	18·01	24·4	29·07	33·52	37·8
5,000	1·17	2·10	3·16	3·98	4·8	5·62	6·32	7·02	7·61	8·31	9·48	11·12	13·69	16·03	18·49	25·04	29·84	34·4	38·84
6,000	1·20	2·16	3·24	4·08	4·9	5·76	6·48	7·2	7·8	8·52	9·72	11·4	14·04	16·44	18·96	25·68	30·6	35·4	39·84
7,000	1·23	2·21	3·32	4·18	5·04	5·9	6·64	7·38	7·99	8·73	9·96	11·68	14·39	16·85	19·43	26·32	31·36	36·16	40·84
8,000	1·26	2·27	3·40	4·28	5·17	6·05	6·8	7·56	8·19	8·95	10·21	11·97	14·74	17·26	19·9	26·96	32·13	37·04	41·83
9,000	1·29	2·32	3·48	4·39	5·29	6·19	6·96	7·74	8·38	9·16	10·45	12·26	15·09	17·67	20·38	27·6	32·9	37·92	42·83
10,000	1·32	2·38	3·56	4·49	5·41	6·34	7·13	7·92	8·58	9·37	10·69	12·54	15·44	18·08	20·86	28·25	33·66	38·8	43·82
12,000	1·37	2·47	3·7	4·66	5·62	6·57	7·4	8·22	8·9	9·73	11·1	13·02	16·03	18·77	21·64	29·32	34·94	40·28	45·48
15,000	1·43	2·57	3·86	4·86	5·86	6·86	7·72	8·58	9·3	10·15	11·58	13·58	16·73	19·59	22·59	30·6	36·46	42·04	47·47

EXAMPLE.—Required the amount of free air necessary to operate thirty 5-in. 'H' drills at 9,000 ft. altitude, using to operate these drills air at a gauge pressure of 80 pounds per sq. in.

From Table I we find, when operating the drills at 80 pounds gauge pressure at sea level, that one 5-inch 'H' drill requires 190 cubic feet of free air per minute.

From Table II we also find that the factor for 30 drills at 9,000 feet altitude is 20·38; multiplying 190 cubic feet by 20·38 gives 3,872 cubic feet air per minute, which is the displacement of a compressor for the above outfit under average conditions, to which must be added pipe line losses, such as friction and leakage. See also 'Problems of Mechanical Coal-Mining,' by S. Mavor, *Trans. Inst. Min. Eng.*, vol. lxvii., p. 456.

According to experiment, the fastest cutting bit is the Carr (double chisel bit with central hole), the Z, + and 6 edged star in this order; cutting speed $\propto$ (drill diameter)$^{-2}$; cutting speed $\propto$ air pressure; pressure of 85 lbs. per square inch most generally satisfactory.

(*Bull. Amer. Inst. Min. Eng.*, August 1917, p. 1079.)

Air Receivers.

An air receiver is not intended so much for storage as to equalise pressure, and to allow of an even flow of air through the pipe, there being a certain amount of pulsation from the air compressor discharge itself. In long pipe lines a smaller receiver should be placed in the pipe line near the point of application of the compressed air, furnished with suitable drain cocks, for the purpose of drawing off the superfluous moisture which might collect, and the consequent prevention of freezing in the line. The following table gives sizes and details of receivers which are usually considered standard practice :—

Table III.—Air Receivers and Pressure Tanks for Standard Working Pressures.
Tested to 165 lbs. Water Pressure.

Diam. in Ins.	Length in Ft.	Contents Cub. Ft. (about).	Thickness of Shell. Ins.	Thickness of Heads. Ins.	Weight Lbs. (about).	Diam. of Safety Valve. Ins.	Diam. of Inlet and Discharge Openings. Ins.	Compressor Capacity Receiver is Best Adapted for in Cub. Ft. Free Air per Minute.
18	6	10	$\frac{3}{16}$	$\frac{5}{16}$	350	1	$2\frac{1}{2}$	90
24	6	18	$\frac{7}{32}$	$\frac{5}{16}$	575	$1\frac{1}{4}$	$2\frac{1}{2}$	120
30	6	29	$\frac{1}{4}$	$\frac{3}{8}$	950	$1\frac{1}{2}$	3	150
36	6	42	$\frac{1}{4}$	$\frac{3}{8}$	1,000	$1\frac{1}{2}$	$3\frac{1}{2}$	150 to 200
36	8	56	$\frac{1}{4}$	$\frac{3}{8}$	1,350	$1\frac{1}{2}$	4	200 to 300
42	8	77	$\frac{9}{32}$	$\frac{3}{8}$	1,750	2	5	300 to 500
42	10	96	$\frac{9}{32}$	$\frac{3}{8}$	2,000	2	6	500 to 700
48	8	100	$\frac{11}{32}$	$\frac{7}{16}$	2,480	$2\frac{1}{2}$	6	500 to 800
48	12	150	$\frac{11}{32}$	$\frac{7}{16}$	3,000	$2\frac{1}{2}$	7	700 to 1,200
54	12	190	$\frac{3}{8}$	$\frac{7}{16}$	3,300	$2\frac{1}{2}$	8	1,200 to 2,100
60	14	275	$\frac{13}{32}$	$\frac{1}{2}$	5,500	$2\frac{1}{2}$	9	2,000 to 3,000
66	18	437	$\frac{7}{16}$	$\frac{9}{16}$	7,500	$2\frac{1}{2}$	10	3,000 and over
24	6	18	$\frac{7}{32}$	$\frac{5}{16}$	625	$1\frac{1}{2}$	4	These are only furnished horizontal style and are used as water traps in air lines.
36	6	42	$\frac{1}{4}$	$\frac{3}{8}$	1,100	$1\frac{1}{2}$	6	
48	8	100	$\frac{11}{32}$	$\frac{7}{16}$	2,200	$2\frac{1}{2}$		

Large-scale Distribution of Power by Compressed Air.

The Powell Duffryn Colliery Co. in South Wales distribute compressed air to collieries 8 to 10 miles away from their Bargoed Colliery at 80 to 85 lbs. pressure by a system of welded steel pipes varying in diameter from 28 to 12 ins., the compressed air being produced by three 40,000 cub. ft. per minute steam turbo-compressors, each turbine developing 7,000 h.p. at full load. It has been calculated that the pressure drop at the pit top would not exceed 10 per cent. when all the collieries were taking their maximum load. Under normal conditions the drop in pressure over 9 miles transmission is about 5 per cent. This good result is secured by using large pipes with consequent low velocity of air. A number of expansion joints are provided in the pipe line.

HAULAGE.

Practice in haulage varies greatly from the best to worst in British collieries. The great majority of collieries use rope and tub haulage, with or without belt conveyors to a central loading point, and the coal stays in the tub until it is tipped at the screens. The obvious waste involved in winding tubs as well as coal can be overcome only by extensive re-equipment, which may not be justified in many cases. Similarly with underground haulage, the cost of reconditioning roads and change of layout required to suit improved methods of haulage is often prohibitive.

As an example of best present day practise in a pit not recently sunk that installed at Manvers Main Colliery serves well.*

Coal from the Haigh Moor Seam, $4\frac{1}{2}$ ft. thick at 347 yds. depth and average dip 1 in 17, had been wound in tubs in ordinary cages from 1942 to 1945, when skip winding was put in operation,

* 'Coal Handling from Face to Screens at Manvers Main Colliery.' Payne & Kimmins. *Trans. Inst. Min. Eng.*, November 1948.

together with an improved haulage system. The four actual workings, all in one district, are advancing longwall faces, each 250 yds. long double-unit, and producing some 500 tons per turnover of 5 ft. Face equipment, which is identical on all four faces, comprises two 24-in. troughed belt conveyors driven by 15 h.p. motors. These face conveyors deliver on to a short-scraper chain conveyor in the centre-gate, which feeds the gate conveyor proper. The interposed all-steel conveyor saves the gate belt from damage at the delivery end, and its cost and charges are fully recovered thereby. The gate conveyors are 30-in. fully troughed belts carried 12 ins. above floor level and driven by 40 h.p. motors at 310 ft. per min.; when the length of haul necessitates its extension conveyors will be arranged in tandem.

The layout of the faces is such that two gate conveyors deliver on to the trunk belt at the same position, from opposite sides, on to a belt already carrying the output of faces inbye. In order to mix the coals from different faces, and to load the belt centrally to avoid spillage, transfer points were arranged. The trunk belt is deflected in such a way that all three belts feed into a hopper which discharges the mixed coal on to the trunk belt again. Dust is suppressed by hoods and water sprays.

The trunk belt is 42 ins. wide and 650 yds. long, driven by a 75 h.p. 3,300 volt slip-ring motor at 380 ft. per min. through a worm reduction and fluid coupling. An ultimate length of 2,500 yds. is envisaged, requiring tandem belts up to about 800 yds. each. Concrete was used for the foundations of the drive head and motor, and for the conveyor structure, which carries the bottom belt 1 ft. clear of the floor. The belt is of six-ply 32 oz. duck with $\frac{4}{64}$-in. top and bottom covers, and the joints are vulcanised.

At the loading point the trunk belt delivers to the mine cars through a shaker chute of 8 tons capacity, which serves both as a reservoir and feeder to the mine cars. The chute is driven by a 15 h.p. motor at 88 by $3\frac{1}{2}$-in. strokes per min., and dust is suppressed by complete hooding and four water sprays.

The all-steel cars, which are of 6 tons capacity, during their passage through the loading point are attached to a twin chain creeper driven by a 13·5 h.p. variable speed air motor fitted with an air brake. Coal is loaded at the rate of 625 tons per hour at peak periods, and 65 to 70 cars are filled each hour at the busiest part of the shift. Control of the loading point is carried out from a station built on the opposite side of the road from the trunk belt, where the operator has all the controls arranged before him, and can inform the pit bottom of the number of cars filled per hour.

The cars are 18 ft. long over the couplings, 16 ft. 3 ins. long in the body by 5 ft. 3 ins. wide and 4 ft. 6 ins. high; they are of fabricated construction, and 2 tons 16 cwt. tare weight. Automatic couplings are used and the clearance between the tops of adjacent cars is 1 ft. 8 ins. The two axles are at 5 ft. 6 ins. crs. have 16-in. dia. wheels.

Two diesel locomotives are required to operate the trains, and one is kept in reserve. They are 100 h.p. 0–6–0 type of 15 tons weight, with four gears, giving a top speed about 15 m.p.h. and a maximum tractive effort about 8,000 lb. An underground depot is provided for servicing and maintenance repairs to the locomotives.

The engines each use about 0·43 gals. of fuel oil per mile, and the total cost per ton of coal hauled is:—

0·14*d.* for fuel oil.
0·01*d.* ,, lubricating oil.
0·10*d.* ,, maintenance;

these costs include non-productive mileage as in hauling mine dirt on back-shifts.

A 36-in. gauge track laid with 65 lb. flat-bottom rails dogged to larch sleepers 5 ft. 10 ins. long by 10 ins. by 5 ins. set at 2 ft. 10 ins. crs. Ballast is of mine rock and ashes graded from $1\frac{1}{2}$ ins. down, and laid 10 ins. deep from 1 in. below sleeper level. Track maintenance required 2 men for 5 shifts a week initially, but this should be halved as the track settles.

Five trains of 15 cars each are used, each train in charge of an engine driver and a guard, who stands on a platform at the rear of the train and signals the driver by coloured lamp and by whistle.

The full trains gravitate on a gradient of 1 in 80 from the creeper at the loading point into a shunt. There they are coupled to the locomotive and hauled 800 yds. against an adverse gradient. The train of cars alone is then lowered back down a slant road 375 yds. long and leading off the main road, to the dumping point for the skip; after discharging the train returns by gravity to the loading point, 370 yds. farther along that slant.

The dump consists of a hopper built under the track, with capacity rather greater than that of one car. Into this the car discharges through bottom doors, which are closed again by a ramp as the car leaves the dump. The cars are moved across the dump by a creeper working against a gradient of 1 in 100, and driven by a 10 h.p. electric motor which is controlled by a pneumatic clutch. One onsetter controls the movement and discharge of the cars into the dump, and the subsequent charging of the coal into the skip then in juxtaposition.

The skip winding plant was adopted to replace the original cage. The skip itself is 34 ft. long by 9 ft. by 3 ft. 2·5 ins., it weighs 8·5 tons and holds 6 tons of coal, *i.e.* it has the same capacity as a mine car. A locked coil winding rope 1·625 in. dia. gives factors of safety 8·5 static and 6·8 kinetic. The scheduled number of winds is 50 per hour, giving a lift capacity of 300 tons per hour.

At the surface the skip automatically discharges itself through a bottom door into a bunker. from which the coal is conveyed to the screens by a 48-in. belt running at 190 ft. per min. At the surface one banksman alone is required ; otherwise conveyance from pit-head to screens is wholly automatic.

In the result it is claimed that this reorganisation requires 74 less men than would intenser working of the previous system of rope haulage and cage-winding, and the O.M.S. on conveying from coal face to screens increased from 19 tons to 87 tons.

Rails.

Mine rails for tub haulage weigh 10 to 10 lbs. per yard. The following table shows the least weight that should be used for the given loads (weight of 1 loaded tub) and low speeds :—

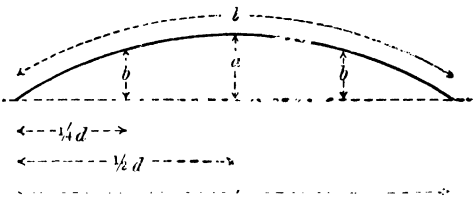
Loads, in cwts.	7	10	15	20	25	30
Weight of rails per yard, lbs.	10	12	15	18	20	22
Gauge of track, in inches	15–18	15–18	16–20	18–22	20–26	20–30

For speeds above 5 miles per hour heavier rails are to be preferred.

Weight of 1 mile of single track in tons = 1·571 × weight of rail in lbs. per yard.

Sleepers 2 ft. 6 ins. to 3 ft. 6 ins., usually 3 ft. apart.

Sleepers 8 ins. to 12 ins. longer than gauge of track, 3–4 ins. deep, 6 ins. wide. A pair of fishplates and 4 bolts for 18 lb. rails, weigh 4–8 lbs., usually 5 lbs. ; 100 dogspikes for same weigh **about 14 lbs.** Weight of fishplates, bolts and spikes is approximately 1 cwt. per ton of rails.



To bend rails to any curve, let l be length of rail, r radius of curve, both in feet, a the middle ordinate in the centre of the curve, b the quarter ordinate, midway between a and the end of the rail (both a and b in inches):—

Then

$$a = 1{\cdot}56\,\frac{l^2}{r}, \qquad b = 1{\cdot}17\,\frac{l^2}{r}.$$

Width of track ¼ in. to ¾ in. greater than gauge of wheels.
Additional width round curves :—

Gauge of track	15	18	24	30 ins.
Additional width	¼	½	⅝	¾ in.

In the case of curves, in all forms of haulage except in regard to main-and-tail-rope haulage (because of the pull exerted by the fore-end rope) the outer rail is raised, in order to prevent the tendency of the tubs to fly off the track. An empirical formula for a suitable elevation in inches E is given by :

$$E = \frac{3DV^2}{3r} \text{ where } V = \text{velocity of tubs in feet per sec.; } D = \text{gauge of track in feet; } r = \text{mean radius of curve in feet.}$$

Tub Control.

A pneumatic system for the remote control of tubs has been in use for some time.

The principle is that two channels fitted with replaceable rubbing strips, arranged parallel with the track and one on each side, engage with the outer faces of the four wheels above the level of the axle, thus applying a steady braking action.

The two channels in each unit are mechanically coupled to pneumatic cylinders situated centrally beneath the track, and set in concrete, 12 ins. excavation being necessary for this purpose.

According to the position of the control valve, either the pneumatic cylinder will be subject to pressure and will close the channels into the braking position, or the pressure will be released in the cylinder, and the channels will be opened out by the return springs.

A similar system, operated hydraulically, has been introduced by Dowty Mining Equipment, Ltd., in which ten such braking units can be run from one 5 h.p. electric motor and pump.

Coefficient of friction of tubs on well-kept mine road (level), 1 per cent. to 3 per cent., average say 1·5 per cent. ; with tubs badly greased on roads in bad condition may be two to three times as much. Coefficient of sliding friction (wheel spragged) is from 12 to 20 per cent. ; if m be coefficient of rolling friction, m' of sliding friction, the coefficient of friction with :—

$$1 \text{ wheel spragged} = \frac{3m + m'}{4}; \quad 2 \text{ wheels spragged} = \frac{m + m'}{2} \quad 3 \text{ wheels spragged} = \frac{m + 3m'}{4}.$$

Additional friction in going round curve of radius r, where the gauge of the track = s (both in feet) is $\frac{s}{2r}\,m'$.

Tractive effort with 18-in. wheels on good level track, 40-in. gauge, at 5 to 6 m.p.h. = 24·3 lbs. per ton with plain bearings, 12·8 lbs. with roller bearings. (*Trans. Amer. Inst. Min. Eng.* vol. lv., p. 29.)

A wooden tub with plain bearings at speed of 1·75 m.p.h. showed coefficients of friction :—

$\frac{1}{30}$ with poor, $\frac{1}{54}$ with average, $\frac{1}{78}$ with good lubrication.

Coefficient on starting from rest with average lubrication $\frac{1}{30}$; tests with plain bearings at collieries gave $\frac{1}{20}$, $\frac{1}{33}$, and $\frac{1}{58}$ respectively. A steel tub with plain bearings gave $\frac{1}{58}$, with Climax bearings $\frac{1}{60}$, with Skefko ball bearings $\frac{1}{135}$; tub with Rowbotham wheels $\frac{1}{86}$.

A 9-ft. radius curve trebles, a 12 ft. radius curve doubles, the coefficient of friction with fixed wheels; loose wheels give ¾ of increase of fixed wheels.

Starting effort to get up to a speed of 1·75 m.p.h. = 2 to 3 times the effort required to maintain that speed with plain bearings, 1½ to 2 times with Rowbotham wheels, and 1 to 1½ times with Skefko ball bearings. (J. Wilson, *Trans. Inst. Min. Eng.*, lxiii., p. 294.)

With 'wheel' greasers the consumption of grease averages about ¼ lb. per tub per diem for ordinary collieries.

On the average a putter can put 10 tons of coal over a distance of 200 yards (level) in an 8-hour shift.

A horse does its best work when travelling at a speed in feet per second $= \frac{1}{4} h$, where h is the height of the horse in hands, and exerting a pull in lbs. of $\frac{60 c^2}{h}$, where c is the circumference of the horse's chest in feet (*Baron*). Average pull is about 1¼ cwt.

Maximum gradient for hand putting, 17 per cent.

Maximum gradient for horse haulage, 12 per cent.

Gradient at which resistance of full tub downhill equals that of empty tub uphill:—Let W be the weight of a loaded tub, w that of an empty one, θ the inclination of the road, μ the coefficient of friction :—$\tan \theta = \frac{\mu (W - w)}{W + w}$ (usually $\tan \theta = 0{\cdot}015$ about).

Work done in hauling a tub of weight W up or down an incline of length $l = l(\mu W \cos \theta \pm W \sin \theta)$

Self-acting Incline.

Self-acting Incline.—Let the sets consist of n tubs each of weight w lbs., and capable of carrying W lbs. of mineral. In the case of over-tub haulage the weight of the rope (or of the chain as the case may be) must be included in W. Let the slope of the incline, assumed uniform, be $\alpha°$. Let l be the length of the rope (= length of incline) in feet, c its weight in lbs. per foot.

Let the rope be supported on rollers, the weight of which per foot of rope (= weight of one roller divided by the distance in feet between the centres of successive rollers) is r lbs.; r must include the weight per ft. of the rope resting on the rollers. Let D be the diameter of the rollers d of the spindle. Let μ be the coefficient of friction of the tub on rails (usually say about 0·015) and μ_1 the coefficient of friction of lubricated bearings (say 0·05 to 0·08). Let F be the frictional resistance of the drum at the head of the incline, including that of the rope coiled upon it—a figure that is always relatively small and may often be neglected. Then at any distance x feet from the starting point the full set is just able to move down the incline when :—

$$\tan \alpha = \frac{\mu n (W + 2w) + \mu_1 l (c + r) \frac{d}{D} + F}{nW - c(l - 2x)}$$

By putting $x = 0$ the above expression gives the angle at which the set will just start from the top of the incline.

The time occupied in descending a distance x will be :—

$$t \text{ seconds} = \sqrt{\frac{2x[n(W + 2w) + (c + r)l + M]}{g\left[\sin \alpha (nw - cl) - \cos \alpha \left(l(c + r)\frac{d}{D}\mu_1 + n(W + 2w)\mu\right) - F\right]}},$$

and the velocity in feet per second v attained after travelling a distance of x feet will be :—

$$v = \sqrt{\frac{2gx\left[\sin \alpha (nw - cl) - \cos \alpha \left(l(c + r)\frac{d}{D}\mu_1 + n(W + 2w)\mu\right) - F\right]}{n(W + 2w) + (c + r)l + M}},$$

where M is the weight of the drum and its fittings and g the acceleration due to gravity.

All the above expressions are approximations, but are accurate enough (too large) for practical purposes.

When the slope is not uniform, the calculations become more complicated. If H be the total vertical fall of the incline and L its horizontal length, it is usually sufficient, with the low gradients prevailing in collieries, to use the above formulas, putting $\alpha = \tan^{-1} \frac{H}{L}$. As the denominator of the expression given above for $\tan \alpha$ increases with x, it is obvious that the incline should, whenever possible, flatten going downwards, or be concave, though not so much

so as to cause the rope to rise too high above the rails. The exact shape of the curve is not very important; it may with advantage be a parabola, which may be set out by making the height h at any point distant y ft. horizontally from the foot of the incline $= \frac{y^2H}{L^2}$. The rope will then follow the curve whenever the relations are such that $H = \frac{cl^2}{4T}$, where T is the tension of the rope. From the above expressions the power required to drive any type of haulage can readily be obtained.

Main-and-Tail Rope Haulage.

Main-and-Tail Rope Haulage.—Loaded tubs in sets up to 40 are hauled outbye by motor (steam-engines, compressed-air engines or electro-motors) at speeds up to 20 miles per hour, whilst the empty tubs are hauled inbye by means of the tail rope. The size of the engine depends upon the pull to be exercised when one set (nearly always the full set) is in its least favourable position. Let T be the tension in the rope in such position, made up of the weight of the set plus rope × sine of angle on inclination of the roadway, the friction of the set and of both ropes on the rollers and the pull required to accelerate the moving parts. Let O be the diameter of the steam cylinder in inches, one of which is alone assumed to be effective at the start, let K O be the stroke of the engine in feet (K is usually $\frac{1}{4}$ to $\frac{1}{3}$), S the steam pressure in lbs. per sq. in., s the ratio of the number of teeth in the spur wheel on the drum shaft to the number of teeth in the pinion on the crankshaft, D the diameter of the drum in feet, then

$$O = \sqrt[3]{\frac{4DT}{\pi SKs}}$$

For any portion of the trip except the start (i.e. with set already in motion), both cylinders are effective, and the value as found above must be divided by $\sqrt{2}$.

These engines are usually small and run at a piston speed of 500 to 800 ft. per min.

Endless Chain and Endless Rope.

Endless Chain and Endless Rope.—Endless chain is now rarely used except on steep gradients and for short distances; it is less reliable than endless rope on long roads, requires more power and is rather more expensive to work, but admits of rather higher speeds; it may work up to 6 miles per hour; endless rope rarely exceeds 4 miles.

The following calculations apply to both methods unless otherwise stated:—

If Q be the quantity of mineral in tons to be transported per hour, W the weight in lbs. carried by one tub, V the speed of haulage in miles per hour, f the distance apart of the tubs in feet, then

$$f = 2{\cdot}36\,\frac{V \cdot W}{Q}.$$

Endless chain is always over-tub haulage, endless rope may be over-tub, under-tub or side-tub. In over-tub haulage the tubs must be so near together that the rope does not touch the ground between them. Let T be the least tension in the hauling rope or chain (*i.e.* between the two tubs nearest the driving pulley on the empty side), let c be the weight of the rope or chain per foot, h the height of the tub, then the distance f_1 of the tubs apart is given by

$$f_1 = \sqrt{\frac{8hT}{c}}.$$

If f is greater than f_1 empty tubs must be run on the full side.

The pull to be exerted by the engine driving the haulage is

$$\mu\left[n\,(2w + W) + 2c\,.\,l\right]\cos a \pm W \sin a,$$

using the same notation as above, n being the number of tubs on any one side of the rope at any time, so that $n = \frac{l}{f}$; the lower sign is used when the gradient is with the full tubs, the upper one when against it. If this expression has the − sign, the arrangement is self-acting.

The return wheel of an endless rope haulage is often used as a tightening pulley, being capable of sliding and pulled by a weight W.

Let the radius of the pulley, fig. 4, be R and of its spindle r.

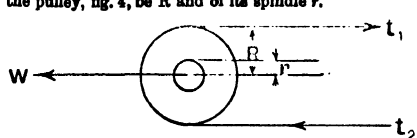


FIG. 4.

Let the coefficient of friction between the spindle and its bearing be μ.

Let the tension in the rope coming off the pulley be t_1, and in the rope leading to the pulley be t_2,

then,

$$W = t_1 + t_2; \qquad R(t_1 - t_2) = r\mu W$$

whence

$$t_1 = \frac{W}{2}\left(1 + \frac{\mu r}{R}\right); \qquad t_2 = \frac{W}{2}\left(1 - \frac{\mu r}{R}\right)$$

Locomotive Haulage.

Locomotives are being used on an increasing scale to replace older methods of rail transport and conveying on main roads. They require large, well-graded, and fairly straight roads and the initial cost of installation can be quickly recovered in suitable applications. The advantages obtained are higher speed and weight of coal hauled with fewer men engaged; two-way hauling and man-riding are feasible, and may effect very considerable economy; also greater flexibility is obtained with less power used. Average speeds of 8 to 10 m.p.h. can be obtained, on the flat or with gradients not exceeding 1 in 15. Where braking is a controlling factor a maximum gradient of 1 in 25 is recommended.

The rolling resistance of the loaded tubs or cars is of vital importance in locomotive haulage On straight level track, with both track and cars in good condition, the rolling resistance varies from 15 lb. to 30 lb. per ton for plain bearings; with taper roller bearings the resistance is about 7 lb. per ton. Starting resistances may be higher by 10 lb. per ton, and curves of the order of 45 ft. radius may increase these resistances by 15 lb. per ton. Curves of about 22 ft. minimum radius can be negotiated when hauling loaded trains on tracks up to 36-in. gauge (see also p. 891). The choice between the several different types of locomotives, *e.g.* electric, diesel, etc., depends chiefly on the gradient, load to be hauled, and mining conditions—with special regard to ignition hazards and health requirements.

Electric Locomotives.—It is accepted that the overhead wire, or 'trolley' locomotive is the most efficient form of traction for use in mines when all conditions are favourable. This arises from the fact that it derives its power from an external source, and that power is readily converted to the rotary motion required for propulsion. Maintenance charges are less than in locomotives with reciprocating parts, and no noxious fumes are emitted. But, owing to the risk of ignition by contact arcing, no trolley locomotives are used in British collieries under existing regulations under the Coal Mines Act; the pertinent regulations are, however, being reconsidered.

Either A.C. or D.C. may be used.

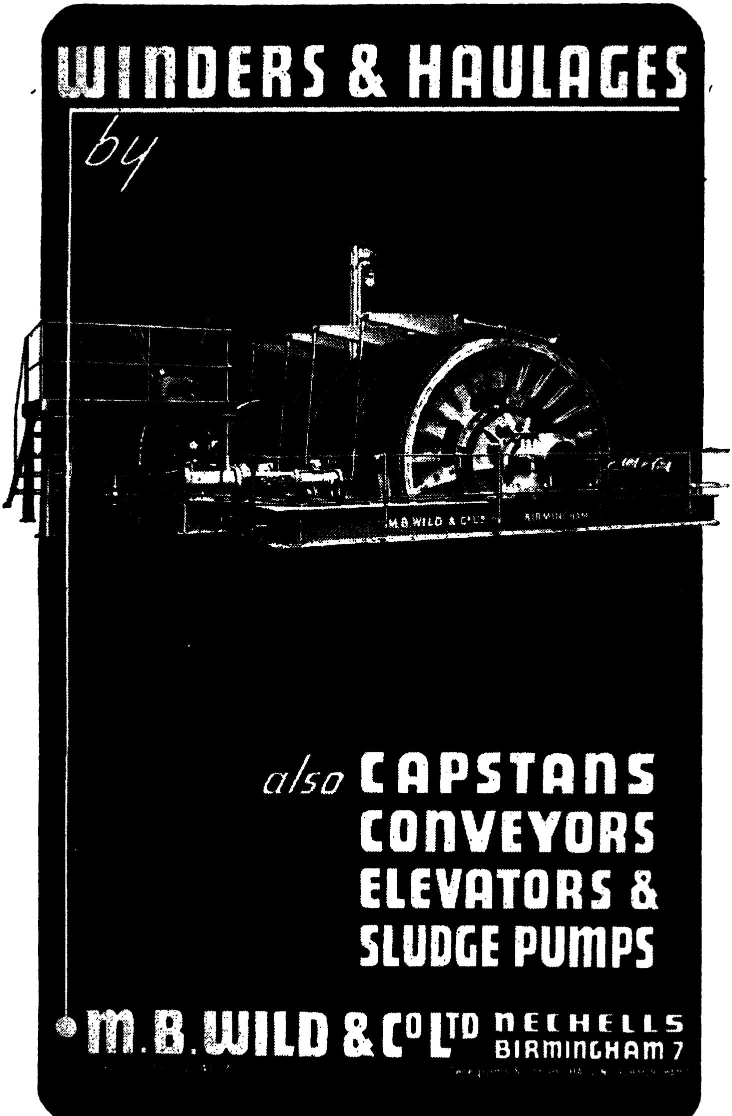
Locomotives of 85 h.p. are used in the Ruhr, and in U.S.A. some have two motors of 150 h.p. each or four of 95 h.p. with weights up to 32 tons. At 75 per cent. efficiency of conversion and 25 per cent. coefficient of adhesion the latter type has a tractive effort of 18,000 lb. and a maximum speed of 8 miles per hour.

Battery-operated Electric Locomotives.—Lead-acid accumulators with a useful life of at least three years are used, and the power output is from 3 to 6 h.p. per ton of locomotive weight. The few used in this country are of small power, but engines up to 40 and 65 h.p. are used in Germany and in U.S.A. Within a limited field the electric battery locomotive is an economical proposition.

Compressed Air Locomotives.—Engines of 40 h.p. with a length of haul of 1,500 yds. are used in the Ruhr. They are intrinsically safe and the exhaust refreshes the atmosphere underground. The air supply is drawn from reservoirs at either end of the run at pressures of 600 to 1,000 lb. per sq. in. It is reduced to 150 lb. at the valve chest. The engine and valve motions are similar to those for a steam locomotive of similar size.

Diesel Locomotives.—The use of this type of engine is extending rapidly in Great Britain, as it provides an extremely flexible system of haulage, and it is the best available under the present Electricity Regulations of the Coal Mines Act, 1911. When properly equipped and maintained these engines can be operated with a high degree of safety in return or intake airways.

Rail gauges of 24 ins., 30 ins., and 36 ins., are to be standardised by the N.C.B. and all of these will probably be used for Diesel haulages. The 0–4–0 and 0–6–0 types of locomotive are used, driven by gearing through a jack shaft with coupling rods, or by chain, to 24 in. dia. road wheels. Several manufacturers are now making them, *e.g.* North British, Fowler, Hunslet, Ruston & Hornsby, in units up to 165 h.p. The Hunslet Engine Co., produces four types, of 16–24, 65, 70, and 100 h.p. In some of these the entire frame is a single-piece casting in steel or high-duty cast-iron, the casting including considerable parts of the gear-box and other components. A friction clutch and four gears provide speeds of about 4, 6, 9 and 14 m.p.h. The 70 h.p. engine weighs 10 tons and has a maximum tractive effort of 5,400 lb. while the 100 h.p. engine weighs 15 tons and has tractive effort of 8,000 lb. An air compressor is driven from the engine to supply air for Westinghouse brakes and for the starting motor: a screw hand brake is also provided. A flame-proof 12 volt dynamo supplies the lamps.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The engine exhaust goes through a stainless steel water bath, and then passes out through a stainless steel flame arrestor to atmosphere on the side remote from the driver. Other makers use a water-cooled exhaust system with special filters and flame traps. These devices ensure safety in respect of flame-proofing. The effect of CO and NO_2 and obnoxious smells on health has to be remedied by dilution by adequate ventilation.

New regulations under the Coal Mines Act, 1911, are shortly to be issued by the Minister of Fuel and Power to govern the use of locomotives underground ; in particular the permissible percentage of CO and NO_2 emitted by diesel engines will be stipulated.

Diesel locomotives have outstanding advantages over other systems ; the initial cost is much less : operating costs for fuel and maintenance are low, and these apply to large or small and concentrated or diffused installations, whether working intensively or at less than full capacity.

Conveyor Haulage.—The normal practice on longwall faces in British mines is to convey the coal off the face, and deliver it at the centre gate either direct into tubs or on to a gate conveyor : in the latter case the conveyor fills the tubs at a central loading station. Belt conveyors are more in favour and more widely used for all purposes than are the older types of scraper-chain or shaker conveyors. The troughed-belt type which is mostly used in the gates, will negotiate inclinations of 18°.

The conveyor system has the drawbacks of high initial cost, and it must be fully loaded for economical working ; also the transport of materials and supplies inbye usually requires a tub-track, and man riding is prohibited. Its advantages are silent and continuous running, its ability to follow the gradients of the seam, less men employed and fewer accidents. Roadway maintenance is also easier and less expensive with conveyor than with tub or locomotive haulage.

WINDING.

In sinking pits, the excavated material is usually hoisted in steel buckets or hoppits, generally unguided, and commonly single. The load is thus wholly unbalanced, geared engines are often used, and the winding speed is slow. Such buckets have capacities of 25 to 50 cubic feet, say 30 to 60 cwts., the weight of the bucket being from ¼ to ⅓ of its load. For ordinary colliery work cages carrying two to twelve tubs are used, always in pairs running on wooden, rail or rope guides ; in collieries in U.S.A. large skips are sometimes used instead of cages. (See paper by H. F. Hebley, *Trans. Inst. Min. Eng.*, vol. lxxxiv., p. 222.)

Practically the whole output of coal in Great Britain is still cage wound, the tubs ascending with their loads of coal. The policy of the National Coal Board is to apply skip winding where practicable in new sinkings and conversions. This system is already used in this country on a small scale and on a larger scale in the Ruhr, Poland, and in U.S.A. where skips of up to 12 tons capacity are used, as compared with 10 tons in this country.

Where cage winding is installed and giving satisfactory service there is no object in applying skip winding. But in cases where an increase in output is not possible with cage winding, then skip winding shows the following advantages.*

The proportion of pay load to total load is raised by about 40 per cent. (*e.g.* from 34 per cent. to 48 per cent.), and loading and unloading are faster, thus increasing the capacity of the shaft ; less labour is required at decking levels ; simpler lay-out of pit at top and bottom ; fewer tubs or cars needed and the size of car used underground is not restricted by the size of cage ; also the rope is in tension always and not subject to snatch. The chief disadvantages are the difficulty of keeping separate coals of different qualities, and that of winding men and materials with a skip, which may often require one shaft to be equipped with cages. Anti-breakage devices appear competent to prevent excessive breakage with skip winding. Plants with capacities for winding up to 3,000 tons in a 7 hr. shift from depths exceeding 2,000 ft. are at work in this country, and much higher capacities, up to 1,000 tons per hour from a depth of 600 ft. are used in U.S.A.

Nearly all of the skips used in collieries in this country are adapted to shafts designed for cage winding (sometimes with one cage still working), with the main object of increasing the shaft capacity, and their design has been influenced thereby. The plan dimensions are similar to those of an equivalent cage, but the length may be greater. The length has to accommodate the side discharge chute and its control doors, as well as the side loading aperture.

Ordinary structural steel sections and plates ¼ in. to ⅜ in. are used for the body, with either welded or riveted construction. High tensile steel plates backed by softwood linings 1-in. thick may be applied to surfaces subject to wear by abrasion ; they last from one to two years.†

Anti-breakage devices are usually in the form of hinged flaps operated by spring and dashpot gear, or the latter combined with counterpoises. These devices also reduce the shock on the rope when the skip is loaded. The bottom doors of the skip are operated and locked by levers with

* For a description of one of the first plants in this country, see 'Skip Winding at Barnborough Main Colliery,' by G. O. Payne, B.Sc., M.I.Min.E., N.A.C.M., 1944.

† See 'Seven Years' Experience of Skip Winding,' by Col. E. Hart, M.C. *Trans. I. Min. E.*, vol. cv., Pt. 6, 1946.

spring catches; and in the Ruhr an improperly closed door trips a catch in the shaft which stops the wind.

When a rope haulage system is retained, with tubs holding 10 cwts. to 20 cwts., tipplers are used to load the skip. The tubs are picked up by a creeper elevator and passed on at a regular rate into the tippler, two tubs at a time. The coal enters a pair of hoppers or measuring pockets, of which one feeds each of the two skips. One onsetter attends to all these operations at pit bottom. The empty tubs pass from the tippler back to a standing line for redistribution to the districts. At the bank, the skip is discharged automatically into a bunker, from which it is fed on to a belt conveyor and passed to the screens.

When feasible a haulage system using large drop-bottom cars holding 3 to 6 tons is preferred. Loading arrangements comprise a dumping hopper of rather greater capacity than the skip at each loading level. The cars are moved over the hopper by a creeper or a retarder, or by air or oil operated piston pushers. The operation of these devices is interlinked with that of the auxiliary equipment, such as door strips and anti-spillage plates, and is placed under the control of one man.

In Great Britain the weight of coal relative to total weight of coal and cage, etc., is about 35 per cent. for cage winding, and about 50 per cent. for skip winding. On the Continent lighter equipment enables these figures to be increased to about 40 per cent. and 55 per cent.

Cages made of duralumin instead of steel have been used for the sake of lightness.

Slack adds greatly to strain on rope.

The velocity of winding varies from 15 to 75 ft. per second; a rule given by O'Donahue is: allow a speed of 20 ft. per second for shafts up to 200 yards in depth, and add 5 ft. per second for every additional 100 yards.

Wooden guides are usually 3 ins. × 4 ins. to 4 ins. × 6 ins.; rail guides 50 to 70 lbs. per yard. Rope guides made of single strand of 7 to 15 wires each $\frac{3}{8}$ to $\frac{1}{2}$ in. diameter, stretched by weight of 1½ to 5 tons; weight is often calculated at 1 ton for every 250 yards depth of shaft. The weights should differ somewhat from rope to rope. The friction on the guides is but small: it usually amounts to about 0·025 of the total load **on the rope with wooden or rail guides and 0·02 of the load with wire rope guides.**

Koepe Pulleys.

The Koepe system is almost universal in Germany and Holland as an economical, safe, simple and flexible method of winding, better suited to their needs than drum winding; it is now arousing growing interest in this country. This system consists of a steam or electrically-driven wheel with a suitably lagged groove in its periphery to take a single winding rope. The Koepe wheel may be mounted in an engine house at ground level, or on the headgear, when it replaces the usual pithead pulley. The cages or skips are suspended one on each end of the rope which passes over the Koepe wheel; the arc of content of the rope around the wheel is always less than 360°, and usually between 180° and 230°. To effect complete balance of the system, a tail rope is used, joining the bottoms of the cages. This rope which is usually heavier than the winding rope and may be flat or round, hangs freely in the shaft and passes round a guided pulley (or a mere timber baulk) set at a position below the lowest winding position of the cages. The ratio of total suspended load to the useful or out-of-balance load is 3·2 to 1 minimum.

Hence all weights other than the useful load are balanced, and the power required is that to raise the load, overcome friction and accelerate or retard the moving masses; the latter are, of course, much less than with drum winders.

The Koepe system is best suited for winding heavy nett loads from deep levels (not less than 1,000 ft.) and is not suitable for multi-level winding. It is, however, more readily adaptable to winding from deeper levels in the same shaft than is drum winding. The risk of slip or creep between the Koepe wheel and the winding rope can be effectively guarded against; first, by using suitable material in the rope groove (*e.g.* leather, rubber, bonded fibre, or special aluminium alloy inserts); secondly, by limiting the rates of acceleration and retardation to induce frictional forces not exceeding 0·2 compared with 0·3 to 0·8 coefficients of friction given by the linings named above; and thirdly, by the right choice of rope and proper attention to its treatment and cleaning.

Regulations in Germany allow a statutory life of 2 years for a rope on a Koepe winder, as compared with 3 years and 6 months under British regulations for a rope on a drum winder; but the latter uses two ropes and the former only one.

The first Koepe winder, supported directly over the shaft, thus dispensing with the usual headgear and pulleys, to be erected in this country, is in operation at Plenmeller Colliery, Northumberland. The Koepe driving pulley is 11 ft. 6 ins. in diameter, driven through single reduction gearing at 200 r.p.m. by a 240 h.p., d.c. motor, fitted with Ward-Leonard control. Current is supplied by a 200 kW., 350 volt generator direct coupled to a 3-phase 50 cycle 600 volt induction motor running at 730 r.p.m. The depth of the shaft is 815 ft., and single deck cages taking 3 tubs holding altogether 48 cwts. of coal are used. The whole of **the hoisting machinery**

is contained in a two-storied steel-framed house with corrugated iron sides and roof mounted on a tall lattice-girder framework. An overall efficiency of 60 per cent. is attained.

(*Trans. Inst. Min. Eng.*, lv., p. 170.)

Another has been working for 25 years at Murton Colliery, Co. Durham. The motor here is 400 h.p. on a 2,000 volt, 40 period current; it draws 4 tons of coals per wind from a depth of 1,431 ft. in 80 secs. The coefficient of friction between rope and driving pulley has been found to be 0·407. The overall efficiency is 61 per cent. (G. Raw, *Trans. Inst. Min. Eng.*, vol. lxxiii., 1927, p. 380).

On the Rand hoisting with skips costs 8*d*. to 1*s*. per ton from depth of about 1,000 ft.

ROPES FOR HAULAGE AND WINDING—See Section XXI, Part III, page 933 (Vol. I).

The time of banking varies considerably; it may be averaged at about 10 to 15 seconds for one deck and 25 to 45 seconds for double decks. Up to 3,000 tons may be wound in 10 hours, but 2,000 tons is not often exceeded.

The consumption of coal in winding engines varies from 0·5 to 1·5 per cent. of the weight of coals wound. Steam consumption 50 to 100 lbs. per shaft h.p. hour.

The cost of a winding engine and boilers erected varies from about 3,500*l*. to about 10,000*l*.

The cost of winding, exclusive of interest and depreciation on the winding plant, varies from about 3*s*. to 10*s*. 6*d*. per 100 tons wound.

Professor S. M. Dixon and Mr. M. A. Hogan have published some measurements of the kinetic loads on colliery winding ropes (Safety in Mines Research Board Paper, No. 78). They state that with steam winders in shafts of moderate depth the total tension on the rope may be as much as two and a half times the dead load, furthermore that the fatigue resistance of hard-drawn steel wire to repeatedly applied tensile stresses is of the order of one-half of the strength shown by a static tensile test. Hence the factor of safety of such ropes is reduced from 8 or 10 to 1·6 or 2, and may be still further reduced by surface damage and corrosion. It has been found that a few inches of slack add considerably to the strain on the rope, as shown by the following table, which gives the average multiples of dead load for a given amount of slack:—

Slack.	*Multiples of dead load.*
3 inches	1·61
6 „	2·14
9 „	2·47
12 „	2·71

Much information will also be found in a paper on 'Mechanical Braking and its Influence on Winding Equipment,' by J. F. Perry and D. M. Smith. *Proc. I. Mech. E.*, vol. cxxiii., pp. 537–620; 1932.

Electric Winding Plants.

Electrically actuated winding engines are being increasingly used at collieries and metalliferous mines. The alternating current (A.C.) generated or received at the mine may be either utilised as such to drive the winding motor or it may be converted into direct current (D.C.). Although the efficiency of the direct drive by three-phase motor is usually higher than in systems where conversion to D.C. is necessary, it sometimes happens that the loss of energy during the acceleration period outweighs the loss of converting to direct current.

ELECTRIC WINDERS ON WITWATERSRAND.

Name.	Depth in ft.	Tons (2,000 lbs.) per hour.	Weight of Mineral.	Weight of Skip.	Maximum Rope Speed.	Drum Diameter.	Winding Rope Diameter.	Tail Rope Diameter.	H.P.	System.
			lbs.	lbs.	ft. per min.	ft	in.	in.		
City Deep . .	3,210	157·5	9,000	8,000	3,500	11	$1\frac{9}{16}$	$1\frac{9}{16}$	1,600	A.C.
Village Deep . .	4,200	152·0	10,600	7,000	3,500	11	$1\frac{9}{16}$	$1\frac{5}{8}$	1,600	A.C.
Brakpan . . .	3,800	180·0	10,000	6,000	3,500	11	$1\frac{1}{2}$	$1\frac{1}{4}$	1,450	Ward-Leonard
Village Deep, Turf Shaft	4,093	150·0	10,000	7,000	3,500	12	$1\frac{1}{2}$	—	1,600	A.C.

The last-named winder (Whiting hoist) shows an overall efficiency of 77 per cent.

(J. F. Perry, *Trans. Nat. Assoc. Col. Man.*, xv., p. 299.)

At the East Geduld Mines an electric winder hoists from a depth of 3,350 ft. 1,400 lbs. per wind, 230 (short) tons per hr.; drum cylindrical 14 ft. diameter, 5 ft. between flanges, maximum 3,580 speed ft. per min.: maximum peak capacity to be 4,570 h.p.

At Aberdare colliery, N.S.W., an electric winder raises 7½ tons net per wind, 490 tons per hr. from a depth of 1,260 ft.; it is rated at 4,500–11,250 h.p. Another shaft at this colliery has an electric winder making 133 trips per hour, with a net load of 4¼ tons, the maximum winding speed being 2,555 ft. per min.

An electric winder is used at Orient No. 2 mine at West Frankfort, Ill. (Chicago, Wilmington & Franklin Coal Co.). Coal skips normally weigh 10 tons, maximum weight, 11·5 tons; coal dumped at shaft bottom from mine cars by tipplers; depth of shaft, 202 yds.; normal output, 1,500 tons per hour; maximum, 2,000 tons per hour; time of wind, 16 secs.; maximum winding speed, 4,000 ft. per minute; peak load during acceleration, 8,000 h.p.; two motors, each 2,000 h.p. one placed at each end of drum shaft; motor generator set; 40 tons fly wheel, 12 ft. diameter; equipped with two 1,800 kW., 600 volt, 600 r.p.m., d.c. generators, driven by a 2,200 h.p. induction motor 220 volts, 3-phase, 60 cycles, 600 r.p.m. At the Nemacolin Mines, Pa., the seam is 250 ft. below and the tipping platform 90 ft. above ground level; the skip carries 16 tons and the plant can wind 1,500 tons per hour; the electric winder develops 1,400 h.p., the drum being cylindro-conical.

The largest electric winder in Gt. Britain is at the Carlton Main Colliery, Grimethorpe; it raises 300 tons per hour from a depth of 2,610 ft., and can develop 7,500 h.p. at 49·6 r.p.m.: it has a 23-ton flywheel Ilgner set.

For full information on electric winders see 'Electric Winders' by H. H. Broughton, 1927, and chapter on 'Electrical Winding in Colliery Machinery and its Application' by R. A. S. Redmayne, 1932.

Electric winders at British collieries have become far more numerous as the 'grid' system has developed.

Automatic control, often electric, is being used. Usually the control of the winding motor is effected by an oil-immersed quick make-and-break reversing switch in the static circuit and a liquid controller in the rotor circuit.

Headgears.

Headgears.—Steel-framed headgears usually range from ¾ to 1 ton per foot of height, and cost erected 12*l.* to 18*l.* per ton; headgears composed of angles and joists only, not lattice girders, weight ½ to ¾ ton per foot of height. Wooden headgears weight about ½ ton per foot of height and cost 10*l.* to 12*l.* per ton. The size of the main legs of timber headgears is given approximately by the expression $A = 0\cdot03\ H^2 \sqrt{W}$, where W is the total working load on the winding ropes in tons, H the height in feet of the headgear from the ground to the pulley centres, and A the area of the cross-section of the main leg; the area of the backstay $= 0\cdot64$ A.

(R. Boyle, *Scott. Inst. Min. Stud.* 1908.)

Steel headgears are now often welded and not riveted.

Ferro-concrete headgears have been erected in several places and have proved quite satisfactory.

Pulleys.

Pulleys.—Usually 10 to 20 ft. in diameter, mostly of the same diameter as the drum. The size depends more on the diameter of the wire used in making the rope than that of the rope itself—and should, according to one eminent authority, be from 1,000 to 1,400 diameters for iron and 2,000 to 2,100 for steel.

If F = the force in lbs. applied to the rim of the pulley required to overcome friction of axle.
W = weight in lbs. upon the axle of the pulley.
D = diameter of the pulley in inches.
d = diameter of axle in inches.
m = coefficient of friction, say 0·07.

$$\text{Then } F = \frac{Wmd}{D}$$

$$\text{or} \quad D = \frac{F}{Wmd}$$

Pillow blocks may be carried on strong springs to diminish shock at starting.

If W be the weight of a pulley in cwts. and d its diameter in ft., $W = 9\cdot2\,(d - 8)$ approximately for large colliery headgear pulleys. $W = \frac{d^2}{4}$ is a useful approximation, but is too low for pulleys under 6 ft. in diameter.

The angle of lead from the drum to the head pulleys should not exceed 2° on either side of the centre line; the horizontal distance between the drum and pulley centres should not exceed 150 ft., otherwise the rope is apt to surge.

Electrical Cables for Collieries.

See MINES AND QUARRIES FORM NO. 11, MINES DEPT. (*Apri.* 1921).*

PUMPS AND PUMPING.

Drainage of Mines.

MEASUREMENT OF WATER FEEDERS.

1. The meaaurement of the flow of water over a double notch weir, the upper portion of which is 12 ins. wide and 4 ins. deep and the lower 6 ins. wide and $4\frac{1}{2}$ ins. deep. The same formula is used in each case treating the weir in vertical sections.

$$Q = 3 \cdot 2\, l h^{\frac{3}{2}}$$

where Q = flow in cubic feet per second.
l = length of notch (or weir) in feet.
h = depth of water flowing over notch in feet.

2. Flow of water in pipes. $Q = 0 \cdot 62 a \sqrt{2gh}$

where V = velocity in feet per second.
g = acceleration due to gravity or $32 \cdot 2$.
h = height of head in feet.
a = area of orifice and $0 \cdot 62a$ is the coefficient for contraction (*vena contracta*).

3. The effect of friction is to reduce the pressure in the latter part of the tube or pipe below that which would exist if friction were absent.

$$\text{Loss of head} = 4f \times \frac{l}{d} \times \frac{V^2}{2g}$$

where l = length of pipe in feet.
d = diameter of the pipe in feet.
$f = 0 \cdot 0075$ approximately.

Drainage of Siphon.—Hawksley's rule for determining the flow of water through siphons is as follows :—

Let L = length of siphon in feet.
d = diameter of siphon in feet.
E = effective fall of siphon in feet.
W = velocity of flow in feet per second.

$$\text{Then V} = \left(\sqrt{d\frac{E}{L}}\right) \times 48.$$

Cornish pumps are never installed at mines now, and very few remain in existence. The foundations are massive and costly, as is the engine also, and the mechanism does not experience uniform strain or work at a uniform speed. Pumping engines are now mostly erected underground and force the water to the surface, but one form of surface-erected pumping engine, the Hathorn-Davey Differential Engine, is still in use as it meets all the disabilities of the Cornish Beam Engine, being a horizontal, reciprocating compound condensing engine which when placed at the surface works a double range of pump rods through the medium of a wooden connecting rod attached to a quadrant.

Other types of mine pumps which are placed below ground are the Tangye, Evans, Worthington, the Reidler, and centrifugal pumps worked electrically. Ram pumps work at speeds of 100 to 200 ft. per minute.

If n be number of complete strokes (i.e. up and down strokes) per min., and d the diameter of the ram or bucket, the theoretical discharge G in gallons per min. is given by $G = 0 \cdot 034 d^2 n.l.$ The actual discharge is less than this by the slip, which ranges from 3 per cent. to 30 per cent. in extreme cases. In practice it is usually between 10 per cent. and 15 per cent. It must be deducted from the theoretical discharge to get the actual discharge. Maximum velocity of water in rising main 200 to 300 ft. per min. rarely exceeds 100 ft. in small pumps; maximum velocity in windbore, 40 ft. per min.

Pressure at bottom of rising main = pressure due to static head + pressure required to overcome resistance of pipe.

If v be the velocity of discharge in feet per second, H the vertical head in feet, L the length in feet of the pipe, D its diameter in inches, and P the total pressure at bottom of rising main, then,

$$P = 0 \cdot 341\, D^2 \left(H + 0 \cdot 024 \left(\frac{L}{12D}\right)^{4} \frac{v^2}{2g}\right).$$

* Printed and published by H.M. Stationery Office. Price 1*s.* net; 1*s.* 2*d.* post free.

The pressure in the steam cylinder must be capable of overcoming this pressure, allowing for the mechanical efficiency of the engine (say 0·66).

The content of a pipe in gallons of water = $0{\cdot}034\ D^2L$.

The thickness of a pipe can be calculated thus :—

Thickness in inches = 0·000125 P.D. + 0·5.

Thickness of pump-barrel = $0{\cdot}5 + \frac{P.D.}{2{,}000}$, where P is pressure in lbs. per sq. in.

Weight of pipe in lbs. = $2{\cdot}5\ L\ (D^2 - d^2)$, where D and d are external and internal diameters. The flange = weight of 1 ft. of pipe.

Horse-power in the water lifted = $\frac{g\ H}{3{,}300}$, where g is the actual delivery in gallons per min.

The 'duty' of a Cornish pumping engine is the number of millions of pounds of water lifted one foot high by the consumption of 112 lbs. of coal (before 1856 of one bushel=94 lbs., and in America of 100 lbs. of coal) under the boilers. To convert duty into lbs. of coal burnt per h.p. in the water, divide 222 by the duty. It is always theoretical discharge that is taken in calculating duty. Cornish engines have worked with as little as 1·5 lb. coal per h.p.; 2·5 to 3 lbs. is a usual figure.

For double-acting pumps the formula for discharge previously given becomes

$$G = 0{\cdot}068\ d^2 n.l.$$

If A be the area of the piston, n number of strokes per minute, S length of stroke, b the height of the water barometer, l the length of the windbore, a its area, h the vertical height of the suction lift, g the acceleration of gravity,

$$a > 0{\cdot}0055\ \frac{An^2 Sl}{g(b-h)} \text{ for single-acting pumps, and } > 0{\cdot}00275\ \frac{An^2 Sl}{g(b-h)} \text{ for double-acting pumps.}$$

or the piston will rise faster than the water can follow it, and hammering will result.

The air-vessel should have a volume = 4 times that of barrel for single-acting, and 1·6 times for double-acting pumps. *(Weisbach.)*

The most economic diameter of a rising main can be calculated by the method given in *Trans Inst. Min. Eng.*, vol. lxxvii., p. 75.

Centrifugal Pumps.

These are the types of pumps now chiefly employed in the drainage of mines.

(See *Trans. Inst. Min. Eng.*, vol. lxxxviii., p. 9.)

Let u be peripheral velocity of impeller in ft. per sec.

- r radius of the impeller in feet.
- n number of revolutions per second.
- d depth of openings in impeller in feet.
- i number of impellers in series.
- h height of total lift in feet.

then

$u = 2\,n\,\pi\,r$. Velocity of radial water discharge from impeller = K u, where K is a coefficient between 0·1 and 0·3, averaging 0·2. Quantity of water discharged per second = $K\,u\,2\,\pi\,r\,d = 4\,K\,n\,\pi^2 r^2 d$ cub. ft. $h = c\,\frac{i\,u^2}{2g}$, where c is a coefficient that varies considerably, but is mostly between 0·75 and 1.

Mechanical efficiency (m) is between 0·5 and 0·8, usually about 0·7.

Hence h.p. required = $\frac{8\,K\,c\,\pi^4 n^3 r^4\,i\,d}{8{\cdot}8\,m\,g}$, or, taking average values, h.p. = about $\frac{3}{4}\,n^3 r^4\,i\,d$.

Direct coupled motor-driven centrifugal pumps to pump 1,500 gallons p.m. to a height of 1,000 ft. have been found to have an overall efficiency of 71·6 per cent.; electrically driven ram pumps would have an overall efficiency of 80 per cent. *(Prest & Leggat.)*

(See also p. 810, Vol. I.)

VENTILATION.

Mine Gases and Ventilation.

Pure dry air according to Haldane consists by volume of:—

Oxygen	20·93 per cent.
Nitrogen	78·10 „
Argon	0·94 „
Carbonic acid	0·03 „
	100·00 „

With 17·0 vol. a candle is extinguished, but breathing is still possible; with 15·2 vols. of oxygen breathing becomes more difficult; with 10 vols. of oxygen, very difficult; and with 5 or 6 vols. practically impossible. Each person requires ordinarily when at work 20 cubic feet of air per min., or when in a working place giving off ½ per cent. of gas 40 cubic feet per min. Each horse requires 90 cubic feet per min. and each flame safety lamp ⅛ cubic foot per min.

Approximately in Great Britain 5·8 tons of air pass through coal mines per ton of coal raised.

The blood in a man's body is capable of absorbing about one pint (about 20 per cent. of its volume) of either oxygen or carbonic oxide when saturated. If it be half saturated with carbonic oxide, unconsciousness sets in, and with higher proportions death follows. In air containing 0·1 per cent. of carbonic oxide the blood would become half saturated in about an hour in the case of a man at rest, and in about half an hour in the case of a man taking exercise. Anything above 0·15 per cent. of carbonic oxide must be considered dangerous; 0·16 per cent. will cause slight distress to mice in an hour, 0·2 per cent. distress in 8 minutes, 0·3 per cent. distress in 4 minutes, 0·45 per cent. distress in 2 minutes and collapse in 4 minutes, 0·57 per cent. distress in 1 minute, collapse in 6½ minutes, and death in 12½ minutes. With canaries 0·09 per cent. causes distress in one hour, 0·15 per cent. distress in 3 minutes, fall from perch in 18 minutes, 0·2 per cent. distress in 1½ minutes, fall from perch in 5 minutes, and 0·3 per cent. fall from perch in 2½ minutes.

Fire-damp.—Fire-damp consists essentially of marsh gas or methane, CH_4, the remainder consisting of nitrogen and carbonic acid; hydrogen and some of the higher hydrocarbons have also been found in some samples. Haller and Moureu have found helium with traces of neon (0·003–0·05 per cent.), and argon with traces of krypton and xenon (0·003–0·04 per cent.) in fire-damp. The high explosive range of a mixture of methane gas and air is with mixtures containing 8 to 14 of methane; when present to the extent of 22 per cent. methane the mixture is non-explosive, the lower limit of inflammability is 6·1 per cent. methane. The igniting point is between 650° and 750° C., and there is always a perceptible 'lag' or delay in the ignition, the amount of which depends upon the temperature. According to Taffanel and Le Floch, mixtures containing only 3 per cent. of marsh gas will ignite at 680° C. and as much as 30 per cent. at 875°. In a mixture of 6·5 per cent. of marsh gas with air, quantity of gas burnt per second varies from 0·00075 per cent. at 535° C. to 0·05 per cent. at 640° C. According to H. Le Chatelier the igniting point of acetylene is 450° C., hydrogen 550° C., methane and carbonic oxide 650° C. The velocity of propagation of ignition of methane ranges from 0·03 m. per sec. with 6 per cent. of methane to a maximum of 0·61 m. per sec. with 12 per cent., and falls again to 0·1 m. per sec. with 16 per cent. of methane in air. The velocity of the explosive wave in a mixture of methane with pure oxygen is 2,500 m. per sec. Marsh gas has been given off in quantities up to 4,500 cubic feet per ton of coal wrought, and has been shown to exist in coal under pressure up to 40 atmospheres. It is on record that in one mine as much as four million cubic feet of fire-damp has been given off in 24 hours.*

Experiments on the ignition of mixtures of methane and air by electric sparks have been recorded by Dr. Thornton (*Trans. Inst. Min. Eng.*, xliv. p. 145, and xlvi. p. 112). Dr. Wheeler found in experiments carried out at the instigation of Sir R. Redmayne in 1914 that with 3 Dania cells giving a current (on closed circuit) of 0·45 amps. under a pressure (on open circuit) of 4·5 volts it is possible to produce a spark by short circuiting the current in signal wires to give a signal, which will explode a mixture of air and methane when methane is present to the extent of 8·2 per cent.

It has been shown that mixtures of methane and air can be ignited by the impact of siliceous rocks on each other. (Safety in Mines Research Board, Paper No. 46.)

Firedamp Detectors.

Coal Mines.

On May 1, 1935, the Board of Trade issued an order that in every ventilating district in which safety lamps are required, a sufficient number of appliances, approved by the Board of Trade, for detecting the presence in the air of inflammable gas, shall be provided by the owner of the mine for use by the workmen employed. In general the number of detectors to be provided for use shall not be less than in the proportion of one detector, in longwall workings for every eight, and in other workings every four, of the total number of persons who are wholly or mainly employed at the working faces.

* See *Trans. Inst. Min. Eng.*, vol. lxxxviii., p. 120.

The provisions for the examination of safety lamps shall apply to these detectors.

In place of the safety lamp required in certain circumstances for use with electric motors General Regulations, 1913), a detector may be provided and used.

These Regulations came into force on October 1, 1935, and may be cited as the Coal Mines General Regulations (Firedamp Detectors), 1935.

The 'Ringrose' is the only approved 'automatic' detector.

PRINCIPLES OF VENTILATION.

(See 'The Ventilation of Mines,' by Hy. Briggs, 1928 'Ventilation of Mines,' by R. A. S. Redmayne; also *Trans. Inst. Min. Eng.*, vol. lxxxviii., p. 109.)

The difference of pressure producing ventilating currents in mines is measured by the water gauge; a column of 1 in. of water corresponds to about 66 ft. of air at 60° F. and 30 ins. of barometric pressure. The theoretical velocity of air due to such pressure is given by the expression $v = 66\sqrt{h}$, where v is the velocity in feet per second and h the water-gauge in inches. If the friction in the airways of a mine is taken into account, according to Murgue—

$$h = K\frac{l\,p\,v^2}{a} \text{ or } h = K\frac{l\,p\,Q^2}{a^3}$$

where l is the length of the air passage, p its perimeter, a its area, all in feet, and Q the quantity of air in cubic feet per second. Values of coefficient K: for drifts, bricked and arched, K = 0·0000012; in ordinary rock, untimbered, K = 0·0000034: timbered in the usual way, dimensions taken inside timbers, K = 0·0000057. If the quantities and velocities be given in thousands of feet per minute, these coefficients become 0·00033, 0·00094 and 0·00156 respectively; some engineers use Mr. Fairley's coefficient, which is 0·002. The critical velocity of air flowing in circular pipes $= \frac{18}{D}$ feet per min., where D is the diameter of the pipe in feet.

The work required to move a quantity Q of air under a water-gauge of h inches = 5·2 hQ foot-pounds.

Mr. J. T. Storrow has found that the quantity of air flowing through material under 0·25 in. in size at a slow rate varies directly with the pressure and not with the square root.

(*Trans. Inst. Min. Eng.*, lv. 313.)

The loss by scaling through packwalls, etc., would therefore, for the same difference in pressure, be greater than in an open airway.

Measurement of flow of air.—In a pipe circular in section a velocity equal to the mean flow is at 0·15 of radius from the wall of the pipe. In a mine drift the mean of two measurements halfway between roof and floor and one-sixth of the width of the airway from either side will approximate to the mean velocity of the air in the drift. The mean velocity is between 0·7 and 0·85 of the maximum velocity at the centre of the drift, averaging 0·78.

If the pressure of air on either side of a regulator or other diaphragm with an aperture of e square feet be determined by water-gauge, and the difference of gauge pressure be d inches, the quantity of air flowing is 40 $a\sqrt{d}$ cubic feet.

The anemometer is used generally for measuring the velocity of the air current, but the Pilot tube may be adopted in cases where high accuracy is desirable when for practicable purposes the velocity may be calculated from the formula:—

$$V = 1098\sqrt{\frac{WG}{W}}.$$

where V = velocity in feet per min.
W.G. = velocity pressure in ins. of water gauge.
w = weight in lb. of 1 cubic feet of air flowing.

Where the measurement of low velocities is necessary the Kata thermometer is sometimes used, especially in the South African Gold Mines.*

In order that the rate of heat loss may be expressed in heat units the total heat loss while cooling between the marks 100° and 95° F. (or corresponding Centigrade) is determined for each instrument by the number of marks and is divided by the surface area of the Kata expressed in sq. cm. The heat unit being a mille calorie ($\frac{1}{1000}$ of 1 grm. calorie). The mean time of cooling of the Kata having been obtained, the factor number is divided by this, and the result is the rate of heat loss per sq. cm. of bulb per second, or cooling power H. To determine air velocity the temperature of the air θ is also taken, then

$$\frac{H}{\theta} = \frac{\text{cooling power}}{36\cdot5 - \text{air temperature, °C}}$$

From this observed value for $\frac{H}{\theta}$ the velocity of air movement can be obtained in cm. per second.

Ventilation surveys are being used increasingly.†

* See 'The Measurement of Low Air Velocities in Mines,' J. P. Rees, *Trans. Inst. Min. Eng.*, vol. lxxiv., 1927–28, p. 359.

† See *Trans. Inst. Min. Eng.*, vol. lxxxviii., p. 181.

Natural Ventilation and Furnace Ventilation.

Natural Ventilation and Furnace Ventilation.—If there be two shafts, the mean temperature of the air in each of which is respectively T and t (in degrees Fahrenheit), the water-gauge due to this difference of temperature will be $\frac{H}{66}\left(\frac{T-t}{459+T}\right)$, where H is the depth of the pit in feet, assuming that the temperature of the atmosphere is 60° and the pressure 30 ins., under which conditions a cubic foot of air weighs 0·0765 lb.; under any barometic pressure of β inches and temperature $t°$ Fahrenheit, the weight of a cubic foot of air is $\frac{1\cdot326\ \beta}{459+t}$, by means of which formula the necessary correction can be applied. The number of heat units required to produce the above water gauge will be $\frac{1\cdot326\ Q\ \beta\ (T-t)}{459+t} \times 0\cdot2375 = \frac{0\cdot315\ Q\ \beta\ (T-t)}{459+t}$ heat units per second.

The grate area is often taken $= \frac{34}{\sqrt{H}} \times \frac{5\cdot2\ hQ}{550}$ sq. ft.

In practice, to pass 300,000 cubic feet of air per min. with a 2-in. water-gauge about one ton of coal is burnt per hour.

Ventilation of coal mines by furnace in Great Britain is permissible only in the case of a small mine, i.e., in which the total number of persons employed below ground does not exceed 30 or if the mine was opened out prior to the year 1911; very few mines now use it.

There is a limit to the efficiency of furnace ventilation, because of the expansion of the air in the shaft due to heat and increasing velocity augments the frictional resistance. Pectet thus stated that when the upcast air has expanded to twice its original volume, the limit of furnace ventilation has been attained. Theoretically about 150 cubic feet of air are necessary for the combustion of 1 lb. of average coal. In practice, however, it takes about twice this amount of air. In the case of an extensive colliery ventilated by furnace which caused 271,356 cubic feet of air to pass down the downcast shaft, the power of the furnace was equal to 111 h.p., but this must be regarded as an extraordinary case.

Fan Ventilation.*

(See Reports of the Institution of Mining Engineers on Mine Ventilation, *Trans. Inst. Min. Eng.*, vol. lxvii., pp. 268, 273; vol. lxx., p. 162; vol. lxxi., p. 337; vol. lxxiii., p. 78; vol. lxxvii., p. 188; vol. lxxxi., p. 313; vol. lxxxvii., p. 9.)

Centrifugal Fan.—If u be the peripheral velocity of the fan blades in feet per second, the theoretical water-gauge produced $= \frac{u^2}{68g} = \frac{u^2}{2180}$.

The manometrical efficiency of a fan ranges from about 0·45 for an open-running fan up to about 0·9 for a good modern enclosed fan.

The volumetric efficiency under given conditions is the volume of air actually passed by the fan divided by the volume of the fan × number of revolutions.

The mechanical efficiency is the horse-power required to drive the fan (or of the fan and fan-engine, as the case may be) divided by the horse-power in the quantity of air moved $= \frac{5\cdot2\ hQ}{550}$. It varies from about 45 to about 75 per cent. in different types of fans.

The equivalent orifice of a mine $o = 0\cdot4\ \frac{q}{\sqrt{h}}$, where q represents the quantity of air in thousands of feet per minute, and o the area in sq. ft. The equivalent orifice of British mines is usually between 20 and 30 sq. ft.

If H be the initial water-gauge of a given fan (maximum water-gauge, produced on exhausting out of a closed vessel), the equivalent orifice of a fan, working on a mine with equivalent orifice o, should be $= o\sqrt{\frac{h}{H-h}}$. The inlet area of a fan should be about 20 per cent. greater than its equivalent orifice. The equivalent orifice of a fan should be about three times that of the mine on which it works. The velocity at which air enters the fan orifice should be between 1,000 and 1,500 ft. per min. If k be the manometric efficiency of any fan, according to Murgue—

$$u = \sqrt{\frac{g\,68\,h}{k}}.$$

To find the horse-power of a motor or engine required to drive a fan under average mine conditions, multiply the number of cubic feet of air by the water-gauge and divide this product

* See Report of the Fan Standardisation Committee of the Inst. of Heat and Ventil. Eng., June 1927; also Min Inst. Report on Testing of Mine Fans, *Trans. Inst. Min. Eng.*, vol. lxxxviii., p. 170; also Mine Ventilation, a Review of Present Theory and Practice by R. Clive, W. Hay and I. G. F. Statham. Tenth Report of the Midland Institute Committee on the Ventilation of Mines.

by 4,500. Example: If a fan is delivering 100,000 cub. ft. of air at 2-in. water-gauge, what should be the horse-power of the motor or engine required to drive it?

$$\text{H.P.} = \frac{100{,}000 \times 2}{4{,}500} = 44 \cdot 4. \qquad (\textit{Machinery.})$$

The Steart fan is an air-screw or propeller fan which is stated to give a very high efficiency and to be readily adaptable to the changing conditions of a colliery. See *Journ. Chem. Met. and Min. Soc. So. Afr.*, vol. xxiv., 1923, p. 31; *Trans. Inst. Min. Eng.*, vol. lxviii., 1924–5, p. 310.

The Report of the Technical Advisory Committee (the Reid Report of March 1945) recommends the wider use of axial-flow type of fans in British coal mines.

Axial-flow fans with variable pitch propellers are now being supplied to collieries and mines in this country and abroad. One of the standard fans made by Walker Bros. (Wigan), Ltd., has a capacity of 250,000 cub. ft. per min. at 6-in. water gauge.

REGULATORS.

Regulators.—If A be the area in sq. ft. of a regulator which is required to produce as much resistance to the flow of air as a roadway of length l, perimeter p, both in feet, a area in sq. ft., K = Murgue's coefficient of friction as above when the quantity of air is expressed in thousands of cubic feet per minute and the pressure in inches of water-gauge, then $A = 0 \cdot 4 \sqrt{\frac{a^3}{Kpl}}$. The regulator should be placed behind the last man in the split.

Safety Lamps.

Until recently it was held that a lamp gauze should be of wire $\frac{1}{70}$ in. diameter and contain 28 meshes to the linear in. (784 meshes per sq. in.); recent researches by the Miners' Lamps Committee have resulted in recommending wire $\frac{1}{50}$ in. diameter with 20 meshes to the linear in. (400 meshes per sq. in.).

See 'Mine Lighting,' by J. W. Whitaker, 1928.

The Technical Advisory Committee in the Report (Reid Report, March 1945) regard the present standard of underground lighting at British collieries as being too low, and consider that a standard of the order of 0·4 ft.-candles in the general working area should be aimed at, that as a source of illumination the flame safety-lamp is obsolete, that a system of general lighting by power-fed lights supplemented, preferably by cap lamps, is necessary at the coal face.

TABLE SHOWING AVERAGE CANDLE-POWER OF VARIOUS LAMPS (*a*) AT BEGINNING, AND (*b*) AT END OF TEN HOURS' BURNING WITH FIVE DIFFERENT OILS.

Lamp.	No. 1 Oil.		No. 2 Oil.		No. 3 Oil.		No. 4 Oil.		No. 5 Oil.	
	(*a*)	(*b*)	(*a*)	(*b*)	(*a*)	(*b*)	(*a*)	(*b*)	(*a*)	(*b*)
Marsaut	0·73	0·63	0·75	0·64	0·74	0·63	0·72	0·64	0·72	0·64
Donald	0·50	0·37	0·49	0·43	0·48	0·43	0·48	0·44	0·48	0·42
Mueseler	0·54	0·42	0·49	0·40	0·46	0·41	0·49	0·40	0·49	0·38
Davy	0·54	0·44	0·83	0·69	0·78	0·70	0·82	0·75	0·81	0·72
'Geordie'	0·46	0·36	—	—	—	—	—	—	—	—
Clanny	0·78	0·72	—	—	—	—	—	—	—	—
Scotch Davy	0·59	0·48	—	—	—	—	—	—	—	—
Deflector	0·96	0·90	—	—	—	—	—	—	—	—

TABLE SHOWING COST IN PENCE PER 1,000 LAMPS OF ONE AVERAGE CANDLE-POWER HOUR, WITH DIFFERENT OILS.

Oil.	Marsaut.	Donald.	Mueseler.	Davy.	'Geordie.'	Clanny.	Scotch Davy.	Deflector.
No. 1	11½	22	14½	18	16	11½	17½	11½
No. 2	19½	33¾	32	19	—	—	—	—
No. 3	25½	43	43½	26½	—	—	—	—
No. 4	35	56	59¾	42½	—	—	—	—
No. 5	45½	73	75¾	43½	—	—	—	—

The oils used were as follows :—

Oil.	Specific Gravity.	Composition, per Cent.		Cost per Gallon.	
		Mineral.	Vegetable.	*s.*	*d.*
No. 1	0·820	100	0	0	9
No. 2	0·841	70	30	1	6
No. 3	0·844	50	50	1	9
No. 4	0·865	30	70	2	3
No. 5	0·889	0	100	2	9

Lamps burning benzine of sp. gr. 0·765, costing 1*s.* per gallon, gave the following results :—

	Candle-power.		Cost in pence per 1,000 lamps of one average c.p. hour.
	At Beginning.	At End.	
Wolf	0·99	0·94	16¼
Protector . . .	0·72	0·66	25¼

Flood lighting at the colliery face is advocated by several authorities. (See *Trans. Inst. Min. Eng.*, vol. lxxxiv.)

Portable Electric Lamps.

It is considered that the ultimate improvement of mine lighting required involves the adoption of mains lighting throughout the pit. However, even in this event, a certain number of portable lamps will still be used, and during the period of years needed in the development of mains lighting, portable lamps will still constitute the chief supply of illumination.

The problem, in the case of portable lamps, is to provide increased lighting without excessive weight, bearing in mind the fact that increased wattage requires a larger battery to feed it throughout the working shift.

A cap-lamp throws the entire light forward in the direction that the wearer is normally looking, and for the same power gives about three times the light in the area of the beam as would an ordinary hand-lamp. However, the general contribution to illumination at the face is no greater.

Bulbs filled with krypton gas give 20 per cent. more light than argon-filled bulbs for the same power consumption. Likewise reduction in the specified life of bulbs would make for greater output of light.

Fluorescent Lighting.

Fluorescent Lighting underground on roadways and at the face is a development on which a number of experiments have been made.

Some of the advantages over filament lamps are the greater efficiency, longer life, and with the much larger area from which light is emitted, the consequently reduced glare—a specially important feature considering the low mounting height possible in a mine. The undesirable sharp shadow effect is also eliminated.

Fluorescent lighting, however, is not intrinsically safe since there is a slight possibility of ignition in gassy mines in case of breakage of the glass tube, though the danger is far less than with filament lamps. Likewise the chances of short life caused by vibration are much smaller, since the starter filaments at each end of the fluorescent tube are very small compared with the filament in an ordinary lamp of the same power.

Nevertheless, regarding replacements and maintenance in general, together with the refitting of lights at the face as it moves forward, there are still a great number of drawbacks to overcome.

See 'Fluorescent Lighting Underground in Mines,' by F. Widnall, *T.I.M.E.*, vol. cvi., pp. 593–605.

Gas Caps.

***Gas caps* are shown in an ordinary safety lamp with the flame turned down so as just not to show any white light, when introduced into an atmosphere containing small amounts of marsh gas or methane, approximately as follows :—**

2 % of marsh gas. Cap ½ in., just visible except at extreme tip.	4½% of marsh gas. Cap 1½ in., well defined.
3 % " " " ½ in., fully formed.	5 % " " " 2½ ins. " "
4 % " " " ¾ in., well defined	5½% " " " 4 ins. " "
	6 % " " " fills the safety lamp.

Gas-testing lamps burning alcohol will show clearly as little as 0·2 per cent. of marsh gas.

Coal Dust.

(See 'The Collection and Analysis of Air-borne Dust during the Driving of Hard Headings,' by J. Ivon Graham and F. Lawrence, *Trans. Inst. Min. Eng.*, vol. xcii., p. 1.)

Coal-dust.—Three ounces of coal-dust per cubic yard of gallery space is sufficient to produce an explosive mixture; excess of coal-dust prevents neither the initiation nor the propagation of an explosion. Coal-dust giving less than 12 per cent. of volatile matter is practically incapable of initiating an explosion. At least 50 per cent. of inert stone-dust is required to prevent the initiation of an explosion. If less than 75 per cent. of the dust passes the 200 sieve it is difficult to initiate an explosion. (*Abstracted from the Lieven experiments.*)

The temperature of ignition of coal-dust is about 1,000° C. Mixed with air and introduced into a heated vessel the temperature may be as low as 590°; there is very little lag. (*Taffanel and Le Floch.*)

The General Regulations provide that in all coal mines except anthracite mines, the mine roadways shall be so treated with incombustible stone dust that contains a percentage of incombustible matter to the extent of 65 per cent. (incombustible matter includes moisture). The incombustible dust used for dusting the mine shall be of such a fineness that of the dry dust which passes through a 60 mesh sieve not less than 50 per cent. by weight, and not more than 75 per cent. by weight shall pass through a 240 mesh sieve and shall be of such a character that it is readily dispersable into the air. Representative tests must be made by the management at intervals of not less than one month, and the results posted at the pit head. The method in which the tests are to be made is fully prescribed by the Statutory Orders (see those issued December 8, 1939). Samples of the dust must be systematically collected and analysed and, in respect of roads used for the transport of coal, and of return airway within 200 yds. of the working face, the number of such samples during each calendar month shall not be fewer than in the proportion of ten per mile of those roads and airways.

In any seam in which, however, inflammable gas is unknown and in which no explosive other than a sheathed permitted explosive is used in any road or ripping or any dry and dusty part of the mine, the percentage of incombustible matter shall not be required to be more than 50 per cent., if that is the natural condition of the dust throughout the road, or more than 60 per cent. if the road is treated with incombustible dust.

Rescue Apparatus.*

Rescue apparatus may be divided into three classes:—

(1) Respirators, which do not supply oxygen, and which absorb by means of a chemical, specific noxious gases.

(2) Tube apparatus, in which the wearer is supplied with fresh air through a tube by means of a bellows or fan.

(3) Self-contained breathing apparatus in which the wearer carried all the means of supporting respiration over a period of two hours independent of any other person or persons.

Respirators are not approved for use in mines by the British Mines Department, although the American Bureau of Mines have approved of their use in U.S.A.

Tube apparatus of the smoke helmet and equaliser types are now approved for use in mines in this country, and there must be maintained at each mine with over 100 underground employees, one of such type of appliance ready for use. The length of tubing employed is 120 ft.

Self-contained breathing apparatus are of two main types depending on the nature of the oxygen supply.

(1) Where the oxygen is carried in steel cylinders at a pressure of 120 atmospheres.

(2) Where the oxygen is carried in the form of liquid air, the liquid occupying one eight-hundredth part of its gaseous equivalent.

The weight of this class of apparatus ranges from 36 to 40 lbs. The expired carbon dioxide is absorbed by from 2 to 4 lbs. of granulated caustic soda, and in a few cases specially prepared soda lime is used.

According to the Rescue Regulations, C.M.A. 1911, the supply of oxygen to the wearer must not be less than two litres of oxygen per minute.

The chief types in use of this class are:—

Great Britain.—Proto: Siebe, Gorman & Co., Ltd., London. Aerophor: Guest & Chrimes, Ltd., Rotherham.

* Kindly supplied by F. P. Mills, Chief Officer, the Durham and Northumberland Collieries Fire and Rescue Brigade.

America.—Gibbs: Mines Safety Appliances, Pittsburg, U.S.A. McCaa: Mines Safety Appliances, Pittsburg, U.S.A.

Germany.—Draeger: Draegerwerke, Lübeck, Germany.

To Detect Carbon Monoxide in the Air.—The most satisfactory and practical method of detecting CO is by means of a small bird. The bird is affected from 7 to 10 times quicker than man, and gives ample warning by its behaviour.

Numerous chemical devices have been evolved for this purpose but none has been adopted for general use in mines.

For accurate quantitative estimations there is perhaps no more satisfactory method than that suggested by Haldane (*Journ. of Physiology*, vol. xviii. p. 430). This is a colorometric test and depends on the colour of blood which has been exposed to the CO under test, relative to that of an equal amount of blood fully saturated with the gas, the matching of the former to the colour tint of the latter being effected by a standard solution of carmine. The spectroscopic test of blood which has been saturated with the suspected gas for the spectrum of carboxy-hæmoglobin is a very simple and rapid qualitative test (*Gas Poisoning in Mining and Other Industries*, p. 381). (*Glaister & Logan.*)

It is estimated that 120 cubic inches of oxygen per min. is ample for the severest exercise; a man at rest requires about 15 inches, and climbing hills 90 cubic inches per min.

The following has been found to be a man's average consumption of oxygen per minute, gas dry, at N.T.P.

Rate of doing Work.	Breathing.			
	Air.		Oxygen.	
	Litres.	Cub. in.	Litres.	Cub. in.
At rest	0·36	22	0·32	19·5
3,000 ft.-lbs. per min. . .	1·12	68·3	0·97	59·2
6,000 " "	1·74	106·2	1·54	95
9,000 " "	2·33	142·2	2·16	131·8

The most economical rate of walking when travelling a flat road and wearing a rescue apparatus—*i.e.* the rate at which a maximum distance can be covered for a given supply of oxygen —is about 2½ miles per hour.

(' *Second Report of the Mine Rescue Research Committee of the Department of Scientific and Industrial Research* ')

When the wet-bulb temperature is much above 80°F., a man cannot remain long at work; he can stand a wet-bulb temperature of 120°F. for about five minutes. (*J. S. Haldane.*)

Recently the 'Hoolamite' detector has been used. This consists of iodine pentoxide dissolved in fuming sulphuric acid, the changes of colour in which indicate the amount of carbonic oxide present; it is said to be able to detect 0·07 per cent. A carbonic-oxide recorder has also been developed, which works by recording the rise in temperature in 'Hopcalite.' This consists of a catalyst combined with oxide of manganese which oxidises carbonic oxide to carbonic acid; the heat developed by this reaction is a measure of the amount of carbonic oxide oxidised. (See 'Devices for Detecting Dangerous Gases in Mine Air,' by J. T. Ryan. '*Trans. Amer. Inst. Min. Met. Eng.*' vol. lxxv., p. 599.) Mr. Ivon Graham has shown that carbon monoxide is a normal constituent of the air of coal mines over 1,500 ft. deep, being produced by the slow oxidation of carbonaceous material at the coal face, in the roads, and in the goaf; it is usually under 0·005 per cent., but in a few cases 0·012 per cent. has been found.

('*Trans. Inst. Min. Eng.*,' vol. lix., p. 222.)

Spontaneous Combustion.

See also Appendix A and B Final Report of the Committee on Spontaneous Combustion of Coal in Mines, 1921, H.M. Stationery Office.

Work in the Doncaster Coal Owners' Laboratory has shown that the liability of a coal to spontaneous combustion in the goaf depends, other things being equal, upon its capacity for absorbing oxygen. It appears that a coal, 100 grammes of which, ground to 200 mesh, absorb less than 200 c.c. in 96 hours at 30°C., is only liable to gob fires when the working conditions especially favour the latter. Coals liable to fire absorb over 300 c.c. in the same time. Presence of FeS_2, especially when finely divided, increases the liability to spontaneous combustion.

Pneumatic stowing, by reducing air leakage across the wastes, is a satisfactory preventative measure against spontaneous combustion.

Fire Extinction.

Investigations have been made into the scientific side of the mud-jet method of fire extinction in colliery workings, and it has been decided that the material of the mud, some of which should be of a lime and some of a clay character, should consist of sifted earth of 5 mm. mesh, boiler ash, dust from the cleaning of blast-furnace gas, and even combustible material, such as the refuse of coal washing, can be added to make up bulk without any risk should nothing more suitable be at hand. In the majority of instances the mud tank can be so placed as to operate by gravity, but steam or compressed air pressure can be used. It has been found that if a tank is placed at a height of, say, 15 yards, it will force the mud a horizontal distance of 150 yards, even allowing for fairly sharp bends in the conducting pipes. (See also Safety in Mines Research Board papers, Nos. 75 and 76.)

Electric Signalling in Coal Mines.

In British coal mines bells and relays have to be such as will not 'spark' (see General Regulation 132). It has been ascertained that with the type of bell and magnitude of battery which used to be employed in mines, nearly every break-flash that occurred when the bare signal wires were separated would ignite a mixture of fire-damp and air containing between 7·5 and 9·5 per cent. fire-damp, were such a mixture to surround the wires at the time.

The Report of the Advisory Technical Committee to the Ministry of Fuel and Power (1945), draws attention to the fact that in this country signalling systems have been developed chiefly to meet the needs of rope haulage and add that the efficiency of some leaves much to be desired, and points particularly to the bare-wire system which demands a high standard of maintenance. The enclosed, or insulated, system requires less maintenance and reduces delays, but necessitates the provision of contact makers at each signalling point and does not furnish the same measure of control throughout as the bare-wire system, although this may be overcome by providing pull-wire control between successive contact makers. This system has, the Report states, been developed especially for use with man-riding haulages.

The introduction of locomotive haulage—the Report proceeds—requires a different system of signalling, essentially based upon visual signals in conjunction with a suitable telephone system, affording communication between haulage junctions.

Signalling.—By the Regulations issued by the Ministry of Fuel and Power, where electricity is used for signalling the pressure in any one circuit must not exceed 25 volts, and contact makers shall be so constructed as to prevent the accidental closing of the circuit; and adequate precautions shall be taken to prevent signal and telephone wires from touching cables and other apparatus. In a seam or part of a seam in which safety lamps are required to be used, if any part of a circuit containing electrical apparatus is in use for signalling (other than apparatus used solely to control the raising and lowering of cages in shafts), it must be of a type approved by the Minister of Fuel and Power. The apparatus has to be constructed so as to conform in all respects with the certificate and with the drawings and specifications appended thereto.

The source of current to be used for operating the apparatus shall be :—

(*a*) For direct current signalling, a battery of 3-pint porous-pot Leclanché cells connected in simple series, or such other source of current as may be certified for the purpose by the Minister of Fuel and Power.

(*b*) For alternate current signalling a transformer of a type certified for the purpose by the Minister of Fuel and Power.

(*c*) For magneto-call telephones, the generator included in the certified apparatus.

The circuits comprising apparatus must be arranged so that—

(*a*) No part of the circuit is connected to earth.

(*b*) Circuits supplied with current from different sources are not interconnected except that it is permissible to use the same line wires for the calling and speaking circuits of telephones.

(*c*) Direct current bells or relays when connected in parallel must be supplied from a single source of current.

(*d*) Direct current bells or relays when connected in series are to be supplied either from a single source of current, or from two identical of current connected in opposition.

(*e*) Magneto-call telephones have to be connected in parallel.

(*f*) Where magneto-call telephones of different types are connected in the same circuit, each instrument has to include a condenser of the type certified for the purpose and connected in shunt with the calling bell.

A list of approved types of signalling apparatus and telephones is issued by the Ministry of Fuel and Power.

ORE DRESSING AND COAL CLEANING.

(For details see 'The Dressing of Minerals,' by Henry Louis, from which the following is mainly abstracted.)

The operations included under the head of dressing comprise all the operations between the extraction or the crude mineral from the mine and rendering it merchantable, thus forming a link between the operations of the miner and the metallurgist. By common consent, gold milling is

generally looked upon as a dressing operation, though it is doubtful whether cyanidation should also be included.

The various operations consist of :—

Hand picking or sorting.

Comminution (coarse breaking or crushing and fine breaking or grinding).

Sizing (preliminary or rough sizing and final or fine sizing).

Classifying.

Separation by differences of specific gravity, magnetism, electric conductivity, or surface tension.

Usually a more or less complicated series of operations is required, especially in the case of complex ores, and these are usually indicated by means of flow sheets.

Let the weight of feed in any concentration process be A and its assay a per cent., the weight and assay of concentrates be B and b and of tails C and c respectively, then the

$$\text{Enrichment ratio} = \frac{b}{a} = \frac{\%\ \text{Recovery}}{\%\ \text{Concentration}}.$$

$$\%\ \text{Concentration} = \frac{100\text{B}}{\text{A}} = \frac{100(a-c)}{b-c}.$$

$$\%\ \text{Recovery} = \frac{100b(a-c)}{a(b-c)} = \frac{\text{B}100b}{\text{A}a}.\quad \text{A} = \frac{\text{B}(b-c)}{a-c}.$$

$$\%\ \text{Efficiency of concentration} = 100\frac{\text{B}}{\text{A}}\left(\frac{b}{a} - \frac{100-b}{100-a}\right) = 100\frac{a-c}{b-c}\left(\frac{b}{a} - \frac{100-b}{100-a}\right) =$$

$$100\ \frac{\%\ \text{Recovery} - \%\ \text{Concentration}}{\%\ \text{Waste in feed}}.$$

(*R. T. Hancock.*)

Sorting is performed either on floors or on picking belts or tables. An ordinary coal picking belt may be of iron plates 4 ft. wide, 50 to 100 ft. long, travelling at 40 to 65 ft. per minute. Pickers stand on both sides, 7 ft. to 15 ft. apart; each picker can treat 1–3 tons of coals per hour.

Power required to drive, 2½ to 5 h.p. Upkeep, about 50*l.* per annum.

Rubber-covered canvas belts about 32 ins. wide are also used, and will carry 25 to 50 tons per hour at speeds of 30 to 60 ft. per minute. Thin steel bands (Swedish) have also been used.

Circular tables are 3 to 4 ft. wide, 10 ft. to 18 ft. in diameter, revolving with a peripheral speed of 40 to 60 ft. per minute, and can deal with 30 to 70 tons of mineral per hour.

Comminution.—Rough breaking is done by jaw breakers, either reciprocating like those of the Blake, Dodge, etc., type, or gyrating like the Gates type. The former have been built to take pieces up to 5 ft. by 7 ft., requiring 300 h.p. to drive, and breaking 300 tons per hour. Smaller breakers usually break 1 ton per hour for a consumption of 2–3 h.p. The gyrating crusher is better suited to very large outputs, but cannot take such large pieces, and appears to be rather more economical of power.

Fine Crushing.—Rolls are used for medium and fine crushing. Let r be the radius of the particle to be broken and d the radius of the particles in the crushed product (both considered as spheres); the minimum radius of the roll should be $> 25\ (r - d)$.

For fine crushing to about ⅛-in. mesh the following table shows the general practice :—

Rolls.		Revs. per minute.	Horse-power required.	Cwts. crushed per hour.
Diam.	Face.			
10½ ins.	10½ ins.	130	3	25
12½ ,,	10½ ,,	110	4½	45
16 ,,	10½ ,,	90	6½	65
21 ,,	10½ ,,	75	9½	100
27 ,,	11 ,,	45	12½	145
37 ,,	12 ,,	45	16	200

The peripheral speed should be greater the finer the product required; for grinding to 40 mesh the speed should be 1,000 ft. per minute.

In large plants three or four sets of rolls are placed in series so as gradually to reduce the ore to the desired size. At Mount Morgan, such a plant consisted of eight sets of rolls in two series of four, thus :—

26 ins. diam.,	15 ins. face,	set	⅜ in. apart,	112 revs. per min.
26 ,,	15 ,,	,,	3/16 ,, ,,	,,
30 ,,	16 ,,	,,	1/16 ,, ,,	,,
30 ,,	16 ,,	,,	in contact	,,

The total power consumption was 100 h.p. and the crushing capacity 125 tons per 24 hours crushed to 20 mesh (0·025 in.); the wear on the steel roll-shells was 0·108 lb. of steel per ton of ore crushed.

Stamps.—The ordinary gravitation stamp is used more largely for gold milling than for other purposes, but also for crushing tinstone, etc. (though of late years in the treatment of gold ore is being largely superseded by ballmilling or tube milling and direct cyanidation). The complete stamp consists of the head, shoe, stem and tappet, the weight of which may be up to 1,500 lbs. The relative weights are about 30, 17, 40, 13. It is lifted by a cam forming the involute of a circle; if h be the lift of the stamp, r the radius of the pitch circle of the involute, and a the angular motion of the cam during the lift, $h = \frac{\pi r a}{180}$. The speed is usually about 90 drops per minute.

If n be the number of drops per minute, h the drop in inches, s the number of stamps, and w the weight of a stamp in pounds, the power required is approximately $\frac{nhsw}{300,000}$. (For a more exact formula see 'A Handbook of Gold Milling,' by H. Louis.) The average crushing capacity is 2–3 tons per 24 hours for a 1,000 lb. stamp. The Husband atmospheric stamp with two heads requires 35 h.p. to work it, and can crush 40 tons of hard tinstone per 24 hours down to 0·048 in. In modern practice on the Rand stamps are used as preliminary crushers, breaking to ½ to 1 in. and are much heavier than above given. (See *Bull. Inst. Min. Eng.*, No. 367, April 1935.)

Steam stamps are used mainly for crushing the copper-bearing rocks of the Lake Superior Mines, down to about ¼-in. mesh. One such stamp will crush 10 tons per hour, the efficiency being about 1·75 tons per i.h.p. per 24 hours.

Ball mills have been made both for dry and wet crushing. The Krupp mill is a dry-crushing mill; the No. 5 size, which is largely used, is 86 ins. diameter by 46 ins. length, makes 20–25 r.p.m., weighs 182 cwts., and a set of steel balls weighs 18 cwts. At Mount Morgan these mills require 13 h.p. and crush 22·7 tons per 24 hours to 20 mesh (0·025 in.). The balls wear at the rate of 0·725 lb. and the hunch plates at the rate of 0·681 lb. per ton of ore crushed.

Wet crushing ball mills are 60 ins. to 80 ins. in diameter and about 40 ins. long. A charge of balls for such a mill weighs about 2 tons; it is run at 25 to 30 r.p.m., takes from 25 to 35 h.p. to drive it, and will crush 25 to 100 tons down to 0·02 mesh per 24 hours; the water consumption is about 65 gallons per minute.

The tube mill is also used wet or dry, and is in effect an elongated ball mill, 3 ft. to 5 ft. in diameter and 13 ft. to 26 ft. long. A tube mill 4 ft. in diameter, 16 ft. 6 ins. long, crushed in Western Australia 38 tons of sand per 24 hours, 95 per cent. of the product being finer than 100 mesh, requiring 30 h.p. to drive it. Such mills are lined with silex, chilled iron, or hard steel bars, and are usually charged with flints. According to Mr. Davidson, the best number of r.p.m. $= 200 \div \sqrt{d}$, where d is the internal diameter in inches; the weight of flints in pounds should be $N \times 44$, where N is the internal capacity of the mill in cubic feet. Western Australian practice corresponds nearly to $N \times 60$.

The Marathon mill uses rods in place of balls; a mill 3 ft. diam., 7 ft. long, charged with 7,000 lbs. of rods ½ in. to 2 in. diam., running at 30 r.p.m., crushed 18 tons per hour with a consumption of 12·5 h.p., using 42 U.S.A. gallons of water per minute. The feed was below ¼-in. mesh and the product was ground to $\frac{1}{14}$-in. mesh, 14 per cent. being below $\frac{1}{200}$-in. mesh. The wear of the liner plates was 0·14 lb. and of the rods 0·4 lb. per ton of feed.

(*Trans. Amer. Inst. Min. Eng.*, lv., p. 678; lvii., p. 355.)

The Hardinge tube mill has the form of two cones base to base. An 8 ft. Hardinge mill requires 50 h.p. to drive it, and requires about 10 h.p. hours per ton crushed to about 48 mesh.

An 8 ft. × 22 ft. mill crushed 200–250 tons per 24 hrs. using about 56 h.p., with a charge of 5 tons of Danish pebbles wearing at the rate of 2·16 lbs. per ton of feed; with steel balls instead of pebbles 360–400 tons were crushed per 24 hours.

(*Trans. Amer. Inst. Min. Eng.*, lv., p. 678.)

The Marcy mill is a short mill (5–6 ft.) with perforated diaphragm at the discharge end, from 2 ins. to 48 mesh. An 8 ft. by 6 ft. mill driven by a 225 h.p. motor crushes 550 tons per 24 hours at 24 r.p.m., carrying 30,000 lbs. chrome steel balls.

(*C. T. Van Winkle, Trans. Amer. Inst. Min. Eng.*, vol. lix. (1918), p. 227.)

The Huntingdon mill is made in three sizes, 3 ft. 6 ins., 5 ft. and 6 ft. diam.; No. 2 is most used; it weighs about 5 tons; consumes 12 h.p.; makes 60–70 r.p.m., and will crush about 10 tons per 24 hours to about 15 mesh.

The Arrastra is 10 to 20 ft. in diameter, has usually four dragstones, 6–8 cwts. in weight. At 6 to 12 r.p.m. a charge of 1 to 3 tons is ground in 2–3 hours.

The Chilian mill has 2–3 rollers, 4 to 6 ft. in diameter, takes up to 75 h.p., and crushes up to 150 tons per 24 hours.

Sizing.—For coarse screening fixed grizzlies or 'jigging' screens are used. For medium and fine screening mostly trommels, 3 ft. to 2 ft. in diameter by 12 ft. to 3 ft. long; the peripheral velocity is usually 50 to 200 ft. per minute. The slopes may vary from 5° to 14°,

averaging about 7°. A set of seven trommels, 3 ft. diam., 5 ft. long, will size about 7 tons of ordinary crushed material per hour, and require up to 5 h.p. to drive it. For the finest work vibrating screens are used. Wedge wire is often used for the screening surface.

Classification.—This term applies to the separation into groups of equal falling particles.

It is to Stokes and Rittinger (1867) and to Pernolet that we owe our knowledge as to the limits within which it is practicable to separate minerals into classes according to size by means of screens before washing. They arrived at the conclusion that water was the best medium for the separation of the separate grains.

If we accept Rittinger's theory that it is necessary to size before washing, the theory being that a grain falling in still water attains a certain maximum velocity of fall *determined by its diameter and specific gravity* at an early period of its first second of fall.

Taking D = diameter of the grain.
δ = specific gravity of the substance.
V = maximum velocity of fall.

$$\text{then } V = \sqrt{D}\,(\delta - 1)$$

and assuming c is a coefficient whose value depends upon the resistance offered by the water,

$$\text{then } V = C\sqrt{D}\,(\delta - 1)$$

Deduced thus:—

Consider a particle of mineral of specific gravity δ and diameter D falling in water.

Then the weight in water = weight in air − weight of water displaced.

$$\text{Therefore the weight in water} = cD^3\delta - cD^3$$

(where c is a constant depending entirely on the shape of the particle, *e.g.* the contents of a sphere = diameter^3 × ·5236. Therefore c = ·5236 if the particle be an exact sphere).

Assuming the specific gravity of the body were 2 then the volume × 2 × 62·5 = the weight of the body, but 62·5 is common and therefore may be eliminated.

$$\text{Hence } cD^3\delta - cD^3 = cD^3(d - 1) \quad . \quad . \quad . \quad . \quad (1)$$

For small velocities the resistance to the motion of the particle in a fluid is proportionate to the square of the velocity.

Therefore, if the velocity of fall at any time be V

$$R \propto k_1V^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

R being the resistance and k_1 a coefficient. It is known also that R is proportionate to the are of the falling particle, hence to D^2 (diameter squared) therefore

$$R \propto k_2D^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

k_2 being a coefficient

Combining (not multiplying) (2) and (3)

$$R = kD^2V^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

k being compounded of k_1 and k_2.

The particle will fall with increasing velocity until the resistance to motion becomes equal to the downward force, when it will proceed with a uniform velocity. To find the uniform velocity equate (1) and (4) (*i.e.* equate the resistance R and the weight in water)

$$\text{Then } cD^3(\delta - 1) = kD^2V^2$$
$$V^2 = \eta D(\delta - 1)$$
$$V = \sqrt{\eta D(\delta - 1)} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where η is compounded of k and c

If we take two bodies of the same shape, but of different diameters D_1 D_2 and of different specific gravity, for these two bodies to fall together in water

$$\sqrt{\eta D_1(\delta_1 - 1)} \text{ must equal } \sqrt{\eta D_2(\delta_2 - 1)}$$

which is known as Pernolet's formulae. Dividing each side of the equation by

$$\sqrt{\eta D_2(\delta_1 - 1)} \text{ we have}$$
$$\frac{D_1}{D_2} = \frac{\delta_2 - 1}{\delta_1 - 1} \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

If we take for instance two bodies of such widely different specific gravities as say quartz (sp. gr. 2·5 approximately) and galena (sp. gr. 7·5 approximately), in order to find the relative diameters of the two particles (D_1 and D_2) which will permit of these two substances falling together in water.

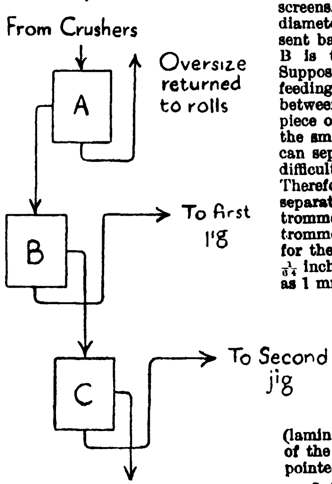
From the equation (6)

$$\frac{D_1}{D_2} = \frac{7\cdot5 - 1}{2\cdot5 - 1} = \frac{6\cdot5}{1\cdot5} = 4 \text{ nearly.}$$

So that if we have a particle of quartz four times the diameter of galena, the two particles will fall together in water. If the quartz be more than four times the diameter of the galena it will fall quicker, and if less than four times the size of the galena, slower than the galena. Hence in jigging the two substances in a washer, if we want the galena to fall first, *i.e.* reach the bottom first,

no particles of the quartz must be greater than four times the size of the smallest particles of galena.

As a practical example of the application of Pernolet's formula, let us take the case of a mixture of quartz and galena. To determine the *sieve scale* for an ore containing these two substances: Take A, B and C in accompanying figure to represent three trommels or cylindrical sieves or screens, and suppose that trommel A has holes 1 in. in diameter, any material less than 1 in. in diameter being sent back to the rolls to be recrushed. The stuff entering B is therefore something less than 1 in. in diameter. Suppose the holes in B to be x inch in diameter, the stuff feeding into the first jig or washer is therefore anything between 1 inch and x inch in diameter: *i.e.* the largest piece of quartz passing to this jig is 1 in. in diameter and the smallest piece of galena is x in. in diameter. If the jig can separate these two pieces (which constitute the most difficult case) then it will separate all the intermediate sizes. Therefore if x be made ¼ inch we shall have complete separation. By the same reasoning the size of the holes in trommel C would be made one quarter the size of those in trommel B, namely $\frac{1}{4} \times \frac{1}{4} = \frac{1}{16}$ inch. Hence the sieve scale for the ore in question would be 1 inch, ¼ inch, $\frac{1}{16}$ inch, $\frac{1}{64}$ inch (though in practice one could not go beyond ¼ inch.: as 1 mm. may be taken as the limit of successful screening).



But in regard to all these calculations it must be borne in mind that Rittinger and Stokes's law of falling bodies in water holds good only for true spheres, and requires variable constants in order that the formulæ may be made applicable to particles of other shape, and where there is presented a mixture containing such different shapes as coal (more or less cubical) and shale (laminations), or porous particles such as fusain the value of the formulæ based on Stokes's law, as Dr. Lessing has pointed out, is very much diminished.*

Spitzkasten: the first box should be 2 ft. wide for 10 cub. ft. pulp per minute, or 1–2 cwts. of solid matter; boxes may increase in geometrical progression with factor of 1·5; length of first box about 6 ft., increasing by 3 ft. for each successive box.

Trough classifiers 10 to 20 ft. long, 1 ft. wide at narrow end, will treat 100 tons per 24 hours with 100 cub. ft. of water per ton of crushed material.

Jigs.—The following represents the general modern practice:—

Diam. of particle	0·25	0·50	0·75	1·00	1·25	1·50	1·75	2·00 ins.
Length of stroke	0·4	1·2	1·55	1·8	2·0	0·15	2·3	2·4
No. of strokes per min.	200	160	145	130	125	125	125	125

CONTINENTAL JIGGING PRACTICE ON ORDINARY LEAD-ZINC ORES.

Three to five compartment jigs; screen surface 20″ × 31·5″.

Size of Particles.	Length of Stroke.	Strokes per Min.	Water per Cwt. treated.	Capacity per Sq. Ft. of Screen Area per Hour.
Inch.	Inch.		Gallons.	Cwt.
0·04	0·2	280	90	0·6
0·046 to 0·06	0·25	280	73	0·9
0·06 to 0·08	0·35	260	60	1·2
0·08 to 0·11	0·50	240	55	1·6
0·11 to 0·16	0·60	220	52	1·8
0·16 to 0·22	0·80	200	50	1·9
0·22 to 0·32	1·00	180	46	2·0
0·32 to 0·43	1·20	160	45	2·1
0·43 to 0·63	1·25	140	54	2·2
0·63 to 0·87	1·60	140	58	3·7
0·87 to 1·25	2·00	140	61	4·3

* See chapter X of Vol. 5 of 'Modern Practice in Mining,' by R. A. S. Redmayne, from which the above is extracted.

Jig screens of steel wire are liable to rapid corrosion when pyritic ores are being treated ; wire drawn from Ambrac (Cu75Ni20Zn5) has been recommended in such cases. Mechanical control has been introduced by several makers.

Round buddle; diam. 18 ft. to 25 ft.; brush makes 5–10 r.p.m. Capacity 1½–2 cub. ft. per min. carrying 28 to 56 lbs. material; time of filling 3–10 hours.

Slime table; diam. 15 ft. to 25 ft.; ¼–1 r.p.m. Pulp 1½–3 cub. ft. per min., carrying 4 to 7 lbs. per cub. ft. Clear water 1½–3 cub. ft. per min. Power required ½–1½ h.

Frue vanner; belt surface 4 ft. by 12 ft.; fall 3 ins. to 6 ins.; travel 6 ft. per min.; strokes 200 per min. Capacity 0·25 to 0·5 cub. ft. pulp per min.; 5 to 8 tons material per 24 hours. Clear water ½ of pulp. Power required ½ h.p.

Lühring vanner; belt surface 3 ft. 6 ins. by 12 ft.; travel 8–12 ft. per min.; 160–180 strokes per min. Capacity 2½–7½ tons per 24 hours; clear water 1–2 cub. ft. per min. Power required ⅓–½ h.p.

Wilfley table; 16 ft. long, 6 ft. wide, tapering to 3 ft. Will take particles up to ¼ in. 240 strokes per min., ¾ in. long. Capacity 25 to 50 tons per 24 hours. Clear water 1 to 3 cub. ft. per min. Power required 1 h.p.

Coal Cleaning.*

Coal as it comes from the mine is first screened, usually over a 2 ins. to 2¾ ins. screen ; the oversize drops on to picking belts, where shale, dirty coal, etc., are picked out. The undersize is taken to a cleaning plant (wet or dry), especially when it is required for coke-making. A jigging screen with ⅜-in. holes will screen about 1 ton per hour for each 0·75 sq. ft. of screening surface.

Coal is sometimes dry-cleaned in the Purdee spiral separator, in which it is allowed to slide down a spiral track, the clean coal falling over the outer edge, whilst the shale, etc., remains on the track.

When the separation of the clean coal from the accompanying shaly matter is easy, or when the products are not required in the highest possible state of purity, a simple form of washer may be used. Such is, e.g., the Elliott trough washer, capable of treating 5 to 10 tons of small coal per hour with a water consumption of 1,750 gallons per ton of coal washed. The Blackett washer is similar in principle, but the fixed trough with the moving dam-plates is replaced by a revolving drum within which is a spiral baffle-plate; it will treat 10 tons of small coal per hour, requiring 3,000 gallons of water in circulation per ton of coal washed. The Robinson washer works on the principle of a spitzkasten, with upward flow of water and a revolving stirrer ; it can treat 15 to 20 tons of small coal per hou .

The purity of the washed coal is usually determined by floating it in solutions of known specific gravity ; coal that floats in solutions of sp. gr. 1·35 to 1·45 is generally considered clean coal. The washing characteristics of a coal are best determined by constructing a washability curve or Henry curve. (See 'Modern Practice in Mining,' vol. v, pp. 432–437, by R. A. S. Redmayne, 1932.) The washability of coals is generally determined by plotting the proportions of the whole which float in solutions of definite specific gravities against the ash contents of these fractions as a curve, from the shape of which much valuable information as to the washing properties of the coal may be obtained.

Most plants for complete washing make use of jigs (bashes or boxes), and the coarser refuse is sometimes crushed and re-washed. See flow-sheet No. 1., p. 919.

The following details are given as explanatory of one form of washer. The unwashed coal, say under 2 ins. to 3 ins., is sized before washing. The washing is carried out in two types of boxes, viz.: (*a*) Small coal felspar boxes, which treat all the coal below ½ in.; (*b*) nut coal type of box, which treats the various sizes below 3 ins. and above ½ in. These sizes vary according to local conditions and the requirements of the market. The nut washers are 8 ft. deep, and the sieves are 5 ft. long and 3 ft. to 5 ft. wide, each washer treating 2·5 to 3·5 tons per hour per foot of width of screen. The more important data affecting the operations of nut washers are shown in the following table :—

* See 'The Preparation of Coal for the Market,' by H. Louis, 1928; and 'Preparation of Coal for the Market' in 'Modern Practice in Mining,' vol. v, by R. A. S. Redmayne, 1932.

Size of Nuts.	Depth of Layer of Shale on Sieve.	Length of Stroke.	No. of Stroke per Minute.	Mesh of Sieve.
½ in. to ¾ in.	2½ ins.	2½ ins.	80	$\frac{1}{10}$ in.
¾ " " 1 "	3 "	3 "	75	⅛ "
1 " " 2 ins.	3½ "	3½ "	70	¼ "
2 ins. " 3½ "	4½ "	4¾ "	65	$\frac{3}{10}$ "
3½ " " 5 "	5½ "	6¾ "	60	$\frac{3}{20}$ "

The Felspar Washer is built in two compartments, each 4 ft. long, making a total length of 8 ft. by 2 ft. 6 ins. wide, the box being 5 ft. deep; such a washer can wash 7 to 10 tons of small coal per hour up to ½ in. in size. The following are the leading data:—

	1st Compartment.	2nd Compartment.
Thickness of layer of felspar	2½ ins.	2½ ins.
" " shale	$\frac{4}{10}$ in.	$\frac{2}{10}$ in.
Size of pieces of felspar	$\frac{8}{10}$ in. to 1 in.	$\frac{5}{10}$ in. to $\frac{8}{10}$ in.
Length of stroke	0·55 in.	0·47 in.
No. of strokes per minute	165	165
Mesh of screen	0·55 in.	0·47 in.

A plant to wash 100 tons per hour, 50 per cent. on nut washers and 50 per cent. on felspar washers, would require four of the former, 4 ft. wide, and six of the latter. Such a plant would require in circulation about 135,000 gallons of water, and make-up water amounting to between 2,800 and 3,500 gallons per hour.

When the refuse from the nut washers contains sufficient intergrown coal to be worth treating, it is crushed, generally in a disintegrator, and the crushed material is re-washed on felspar washers.

A coal washery may require from 1·5 to 3·5 h.p. for each ton of unwashed coal treated per hour, varying with (*a*) the total output of the washery (a large washery requiring a relatively smaller h.p. per ton); (*b*) the quantity of interstratified coal, and whether the larger size is crushed and washed together with the smaller size, or whether the smaller size only is re-washed; (*c*) the type and capacity of draining and storage bunkers, and whether apparatus is installed for mixing the various products.

Messrs. Simon-Carves, Limited, build washeries on the so-called Baum principle, in which washing precedes classification. The slack below about 2 ins. is delivered to a bash having an area of about 0·8 sq. ft. per ton per hour; the washed coal then passes to a screen of about ⅜-in. mesh; the over-size is clean coal, and the under-size goes to a second bash having an area of about 1 sq. ft. per ton of coal per hour, the product being clean coal and refuse. The dirt produced in the first washing can be further crushed and washed over again if necessary. A plant to treat 100 tons of slack per hour would, therefore, require a first bash with an area of 80 sq. ft., and a second bash with an area of 100 sq. ft.; the water in circulation would be about 130,000 gallons per hour, and the make-up water required about 2,500 gallons.

The rhéolaveur consists of a series of troughs of steel, cement or wood, sometimes lined with tiles, fitted with a series of pockets with adjustable discharges and, for the coarser sizes of coal, with upward water flows. These are arranged so as to discharge clean coal, refuse or middlings for re-washing as may be required. The plant is relatively cheap, compact and efficient and can treat all sizes from 4 in. to slurry. No general figures can be given as the plant has to be specially designed for each case. It is largely used in Belgium, and altogether about 60 million tons of coal a year are treated by it.

At Grassmoor Colliery a rhéolaveur plant treats 100 tons per hour 0-1 in. in two sizes, 40 tons 0-⅜ in., 60 tons ⅜-1 in., absorbs 105 h.p., uses 140,000 gals. water in circulation.

(*A. Audry*, *Proc. So. Wales Inst. Eng.*, vol. xli., p. 567.)

(See 'The Rhéolaveur Coal-Washer in Belgium,' by H. Briggs and H. Louis, *Trans. Inst. Min. Eng.*, vol. lxxiv., 1928, p. 285; also 'The Operation of a Small-coal Washer,' by C. E. Pulford, *Trans. Inst. Min. Eng.*, vol. lxxviii., 1929, p. 47.)

The Hoyois process, used in the North of France, is not dissimilar.

In the United States small coal is being largely washed on shaking tables of the Wilfley type; such a table will treat 5 to 8 tons per hour under ⅜-in. mesh, requiring about 1 h.p.

The flotation process has been applied to washing fine coal to a very small extent in Britain but extensively in Spanish collieries.

The Chance process employs a separating cone worked with a sludge consisting of sand and water; it is said to give excellent results on Anthracite, and is used in the U.S.A. (See *Trans. Amer. Inst. Min. Met. Eng.*, vol. lxx., p. 740.)

The Lessing process uses a solution of calcium chloride as a separating medium. This process was used by Sir Henry Bessemer in 1858.

One of the principal difficulties in coal washing is presented by the fine coal (say $\frac{1}{30}$ in.), which forms slurry when suspended in water; it consists often largely of fusain, and is therefore non-coking and is often very impure. It may be collected in settling tanks, which, however, occupy much ground space. Most makers of washing plant supply an elevated conical settling tank, from which a thickened slurry can be drawn off from the bottom, whilst tolerably clean water is drawn off from the top. A still better device is the Dorr Thickener, usually between 50 and 100 ft. in diameter, though made up to 200 ft. diameter and about 10 ft. deep; the stirrer makes 4–8 revs. per hr., and 100 sq. ft. of thickener area are allowed for each 25 gals. per min. The Rheolaveur deals efficiently with all but the finest slurry. A recent device is the Kirkless Slurry Separator, which filters the slurry. It is generally considered preferable to blow out the dust or draw it off by an aspirator before the coal goes to the cleaning plant if the coal is dry enough. The dust is collected in cyclones, bag filters, etc., and is generally very suitable for dust firing, though it may need further grinding for this purpose. Some form of starch is sometimes used to 'settle' slurry.

Dry Cleaning.—The American Coal Cleaning Corporation's machine consists of a shaking table with riffles and a porous deck through which a current of air ascends on the principle of the Sutton-Steele table; this cleans dry coal and the finished product contains under 2 per cent. of extrinsic ash. Over 10,000,000 tons of coal were cleaned in this way in 1927 in the U.S.A. (See *Trans. Amer. Inst. Min. Met. Eng.*, vol. lxx., p. 758.)

The largest American plant is the McComas plant at Crane Creek Mines, W. Virginia, treating 225 tons per hour. The Berwind White Colliery Co. has at present the biggest dry cleaning plant in existence.

This table has been introduced into England by the Birtley Iron Company, Ltd. Its first form, known as the Birtley SJ Table, has been erected in a number of places, one of the first being at Wardley Colliery, Co. Durham. This plant treats coal under 2½ in. and consists of six SJ tables with dust aspirator and baghouse. It cleans to within 2 per cent of the intrinsic ash (by which is understood the ash occurring as part of the coal) leaving about 1 per cent. of free coal in the refuse. Its original capacity was 125 tons per hour, but this has subsequently been increased.

An improved form, known as the Y Table, was introduced about 1926, having double the capacity of the SJ Table, and not needing as close sizing. A still newer form known as the V separator, made in two types, the standard V and the super V, has been introduced; the former has about three-fourths of the capacity of the latter, which is as follows on average coal:

2 in. to ½ in.	60 tons per hour
½ „ „ ⅛ in.	40 „ „
Below ⅛ in.	20 „ „

The power consumption for the fan and drive of these machines averages about 0·75 h.p. per hour. The Birtley Company has constructed numerous plants in England, eight were under construction in 1930, and the total operating cost, including labour, maintenance and repairs, is given at about 2·25*d.* per ton. (See papers by K. C. Appleyard, *Trans. Inst. M.E.*, vol. lxxiii., 1927, p. 404, and World Power Fuel Conference, 1928, No. C 7.)

In 1935 there were 82 plants erected, including about 12 that represented increases in existing plant and 5 under erection.

Another table used recently in this country is the Static dry cleaner of G. Raw. It consists of a long relatively narrow table, without riffles, but with three skimmers; the bed of the table is, like the last, of perforated material, but a pulsating current instead of a continuous current is introduced into the air-chest beneath the bed. The tables have capacities of from 30 to 50 tons per hr. and the power consumption is 0·85 B.O.T. unit for a ton of coal. Costs including labour, power, maintenance, interest and depreciation, are given at 3½d. per ton (see paper by G. Raw and F. F. Ridley, World Power Fuel Conference, 1928, No. C 6). Another dry coal-cleaning table is the Kirkup table, in which the table is at rest, but receives a pulsating air current beneath its perforated bed. The Peale-Davis table is like a Birtley Y table, 40 ft. long, 14 ft. wide, with a capacity up to 250 tons per hr. This is being used at Nunnery in Yorkshire and at Bedwas in South Wales.

The dry cleaning of coal has these advantages over the wet processes:

(*a*) There is no water to be obtained, handled and clarified.

(*b*) The coal has not to be drained of water.

(*c*) There is no absorption of water by the coal.

(*d*) There being no wet coal, the difficulty of dealing with frozen coal in wagons in winter does not arise.

(*e*) The dryer the coal, other things being equal, the higher its calorific value. Coals to which the dry process is particularly applicable are :—

(*a*) A coal subject to discoloration owing to the presence of lime, if treated by water.

(*b*) Coal friable in character which under wet cleaning would absorb much water, hygroscopically and externally.

(*c*) Which are to be subjected to carbonisation, when the presence of water is an obvious disadvantage.

In some cases dry cleaning may with advantage be combined with washing, the coal below, say, $\frac{5}{8}$ in. going to a dry-cleaning plant and the coarser coal to a washery ; the washed coal after drainage is mixed with the dry-cleaned coal and the product carries a proportion of moisture suitable for many purposes.

(See also 'The Dry Cleaning of Coal,' by H. Louis, *Fuel Economy Review*, 1928.)

Dust is now occasionally removed from coal that is not naturally too wet before it is cleaned by washing or dry-cleaning ; it may be screened off by fine vibrating screens or may be taken out by an air current either on the screens or on the dry cleaning tables or else in a dust extractor ; the dust is collected by cyclones and bag-houses and is often used for dust-firing. Wedge wire is occasionally used for the fine screens. The Waring dust collector is being used successfully. (See *Min. Mag.*, Dec. 1928.)

Vibrating screens have a tightly stretched screening surface kept in rapid vibration and set at a tolerably steep angle ; the Hummer, Hayle, Overstrom, etc., screens differ mainly in the mechanism by which the vibration is produced.

For a good comparison of different methods of coal-cleaning see paper by Wheeler and Chapman in report of the Coal-Cleaning Conference at Edinburgh, July, 1927, pub. by Soc. Chem. Ind.

Mr. Norman Kemp has applied X-ray analysis to the products of the coal washery.

(*Trans. Inst. Min. Eng.*, vol. lxvii., p. 59.)

Magnetic Separation.

Maxima of magnetic susceptibility of minerals by volume in C.G.S.

Soft iron	——	400
Magnetite (pure crystallised)	Piedmont	3·12
Magnetite	Hay Tor, Devonshire	1·44
Magnetite	Altenfjord, Norway	0·27
Magnetite	Lake Champlain, U.S.A.	0·234
Magnetite (altered carbonate, impure)	Bettwys Garmon, Carnarvonshire	0·06
Red Hæmatite	?	0·00073
Red Hæmatite (crystallised)	Cumberland	0·00017
Specular Hæmatite	Nova Scotia	0·00106
" "	?	0·0005
Brown Hæmatite	?	0·00042
Brown Hæmatite (pure crystallised)	Nova Scotia	0·00011
Franklinite	New Jersey	0·0037
"	" "	0·00253
Ferrous Sulphide	Artificial	0·064
Spathic Ore	?	0·000559
Clayband	?	0·00069
Impure Carbonate Ore	Northamptonshire	0·00056
Ilmenite	India	0·00147
Monazite	Travancore	0·000069
Zircon	Ceylon	0·00000055
Pleonaste	?	0·000102
Gahnite	?	0·000054

Common non-ferrous minerals, *e.g.* mica, quartz, felspar, calcite, have susceptibilities $<$ 0·000001. 'Strongly' magnetic minerals have susceptibilities $>$ 0·001. 'Feebly' magnetic minerals between 0·001 and 0·0001.

(*After Prof. Ernest Wilson.*)

MAGNETIC SEPARATORS.

As examples of the capacity of modern magnetic separators we give the following figures :—

Dry Work.—Drum, 24 ins. diameter with 24 ins. face ; capacity 12 tons per hour ; will take lumps up to 20 lbs. in weight.

An electro-magnetic separator with six magnetic fields with maximum current on the four magnetic poles takes 12 amp. at 100 volts.

Flotation Processes.

These may be classified as follows :—

I. *Film Flotation*, utilising differences of surface tension on particles at an air-water interface—*e.g.* De Bavay, Macquisten processes.

II. *Oil Flotation*, utilising differences of surface tension in an oil-water medium where the particles are buoyed up by oil—*e.g.* First Elmore process.

III. *Adhesive Processes*, utilising differences of surface tension between particles and oil or grease, causing differential adhesion to oiled or greased surfaces—*e.g.* Murex, Cattermole, greased-plate processes.

IV. *Froth Flotation*, utilising differences of surface tension at gas-water interfaces, where the particles are buoyed up by bubbles of gas :—

(*a*) Where the bubbles are produced by chemical action—*e.g.* Potter, Delprat processes.

(*b*) Where the bubbles are produced by releasing the air dissolved in water—*e.g.* Elmore vacuum process.

(*c*) Where the bubbles are produced by mechanical means—*e.g.* Minerals Separation, Callow, Janey, K. & K. (Kraut & Kolberg), Rork, Kleinbentink, Ekof, Fagergren processes.

(See 'Froth Flotation : its Commercial Application and its Influence on Modern Concentration and Smelting Practice,' Walter Broadbridge, *Trans. Inst. M. M.*, vol. xxix., p. 205, 1920. 'A Contribution to the Study of Flotation,' H. L. Sulman, *Ibid.*, p. 44. 'The Concentration of Ores by Flotation,' H. L. Sulman, *Bull. Inst. M. M.*, Aug. 1930. 'Flotation Reagents,' B. W. Holman, *Bull. Inst. M. M.*, Nov., 1930.)

Costs of Flotation Machines.

Tons capacity for 24 hours.	First Cost per ton of daily capacity.		Operating Costs per ton daily capacity.	
	Mechanical Agitation.	Pneumatic Agitation.	Mechanical Agitation.	Pneumatic Agitation.
25	$30–50	$25–40	$0·20–0·75	—
50	25–45	12–20	—	$0·20–0·75
100	20–40	8–12	0·35	0·35
250	15–35	6– 8	0·28	—
1,000	12–25	4– 6	0·17	0·17

(*Ralston.*)

Reagents used :—

Gangue modifying :—Acids, alkalies, silicates carbonates, and other salts.

Froth producing :—Soluble oils, essential oils, cresylic acid, amyl alcohol, etc.

Froth stabilising :—Insoluble oils, petroleum, etc.

According to figures collected by the United States Bureau of Mines, a total of 50,073,450 tons of ore was treated in the United States by the flotation process in 1927. By far the greater portion, 40,881,768 tons, consisted of copper ores. The remainder was made up of complex lead-zinc, lead, zinc, copper-iron, and miscellaneous ores. A total of 220,514,373 lbs. of reagents was consumed in the treatment by flotation of all classes of ores. The bulk of this consumption was lime, of which 169,926,145 lbs. were consumed. Pine oils constituted the greater portion of the frothing reagents used, accounting for 5,064,320 of a total of 6,583,151 lbs. Appreciable amounts of cresols were also used. Of the oils used as collecting reagents, coal-tar creosotes and coal tar made up 2,655,352 of a total of 3,508,993 lbs. Other oils used as collecting reagents were wood-tar creosote, crude oils, petroleum products, blast-furnace oils, water-gas oils and tars, and miscellaneous and reconstructed oils. Ethyl xanthates and di-thio-phosphoric acids were used as collecting reagents to the extent of 3,319,639 lbs. and 1,932,996 lbs. respectively, and made up the great bulk of chemicals used for this purpose. Other chemicals used were higher xanthates (amyl and butyl), thio-carbanilide, alpha-naphthylamine, thio-ureas, and oleic acid. The acids and alkalis used in addition to the great consumption of lime were composed of sulphuric acid, hydrochloric acid, sodium carbonate, sodium bicarbonate, sodium hydroxide, barium

carbonate, and cement. Other inorganic reagents used included sodium sulphide, calcium and barium sulphides, copper sulphate, cyanides, sodium sulphite, sodium silicate, zinc sulphate, sodium dichromate, tri-sodium phosphate, aluminium sulphate, sulphur, calcium chloride, and sodium chloride. Some quantities of glue and starch were used as protective organic colloids.
(*The Chemical Age*, July 6, 1929.)

Surface Tension of Liquids against Air.

	Dynes per sq. cm.			Dynes per sq. cm.
Water (at 0° C.)	75·8	Water with 0·14 per cent. cresol	(at 23° C.)	71·5
Water („ 20° C.)	75·1	Water with 0·14 per cent. amyl alcohol	(„ 23° C.)	68·4
Phenol („ 41° C.)	37·0	Water with 0·134 per cent. saponine	(„ 23° C.)	48·7
Cresol	34·8	Water with eucalyptus 'solute'	(„ 20° C.)	44·6
Olive oil	32·0			
Petroleum	27·7			
Turpentine	25·6			

Dilute sulphuric acid reduces the contact angle and hysteresis range of that angle far more for quartz, silicates, and oxides than it does for metallic sulphides; oiling the latter greatly increases the contact angle and the hysteresis range.
(*H. L. Sulman*, 'A Contribution to the Study of Flotation,' *Trans. Inst. Min. Met.*, vol. xxix., 1920.)

The Inspiration Consolidated Copper Company's plant at Miami, Arizona, treats 15,000 tons in 24 hours. The ore contains 1¾ per cent. of copper, mainly in finely disseminated sulphides. Flotation reagents used, 1¼ lbs. per ton of ore. Recovery, 85 per cent. of total copper, 90 per cent. of copper present as sulphide; concentrates contain 20 per cent. of copper.
(*Trans. Amer. Inst. Min. Eng.*, lv., 1916, p. 576.)

Selective flotation is very largely employed for the treatment of lead-zinc ores in the Western States and in Mexico. (See 'Selective Flotation as Applied to Canadian Ores.' C. S. Parsons, *Canad. Min. Journ.*, 1927, vol. xlviii., p. 468; also 'The Trend of Flotation,' Weinig & Palmer, *Colorado Sch. of Mines Quart.*, 1926, vol. xxi., No. 2.)

Finely ground coal may be froth-floated from shale, using 1 lb. of cresol and ¼ lb. of paraffin, per ton of coaly shale; the shale contains about 65 per cent. of ash, the clean coal about 8 per cent. Potassic Xanthate is also used.

ELECTROSTATIC SEPARATION is used in some special cases—*e.g.* for concentrating rosin blende, molybdenite, and graphite (Blake-Marsden and Huff processes).

DRY SEPARATORS, using currents of air instead of water, are used for ores under exceptional conditions.

The Advisory Technical Committee in its Report to the Minister of Fuel and Power (1945) states that at present far too much hand picking of coal goes on, which is extravagant in man-power and has only been justified by the higher prices which have been obtained in the past for the large coal sorted in this way. The Committee points to the fact that in some Continental mines, despite the use of pneumatic picks for getting the coal, the whole of the product, with the exception of the largest lumps, which are hand-picked, is washed.

The Committee is of the opinion that with power loading of coal at the face (under which system a greater admixture of dirt may be expected), washing (with breaking of lumps of about 10 ins. and upwards) of the entire output should become general. At present (1939), only about 47 per cent. of the British coal output is subjected to washing.

All new coal treatment plants should, the Committee states, be appropriately equipped to treat fine coals and render them saleable. Dry fines can be separated by 'aspiration' and all fines can be cleaned by 'froth flotation.'

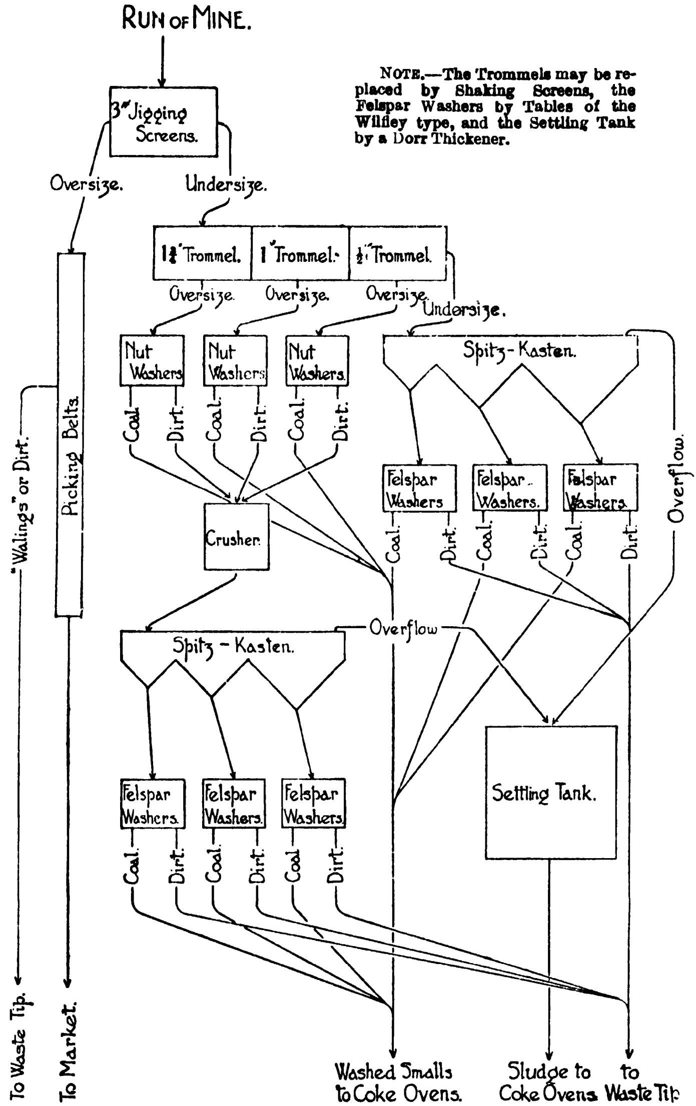
FLOW SHEETS.

The diagrams given on pp. 919–927 are specimen flow sheets, illustrating the sequence of operations in certain typical cases:—

I and II. Coking coal. III. Dry cleaning of coal. IV. Low grade magnetic iron ore. V. Quartz carrying cupriferous pyrites. VI. Lake Superior copper ore. VII. Lead ore. VIII. Tinstone containing wolfram. IX. Copper ore by Froth Flotation.

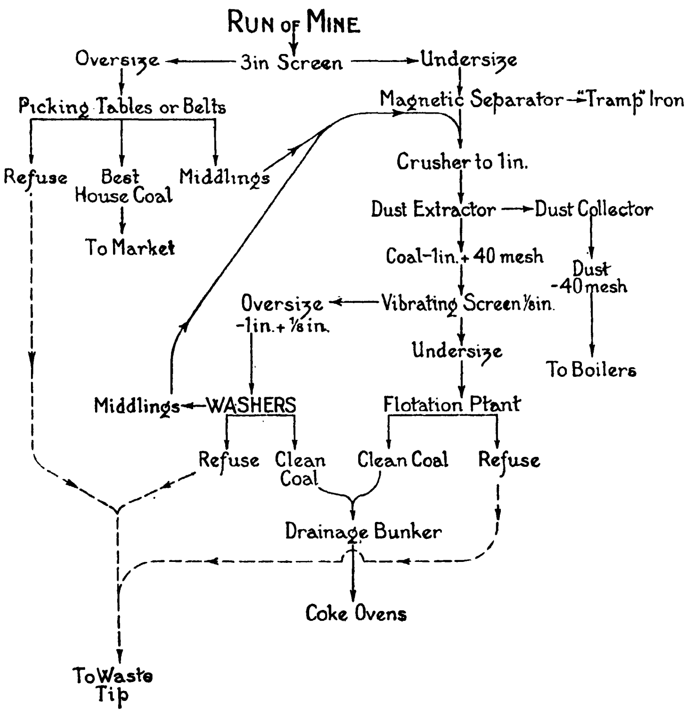
FLOW SHEET I.

NOTE.—The Trommels may be replaced by Shaking Screens, the Felspar Washers by Tables of the Wilfley type, and the Settling Tank by a Dorr Thickener.

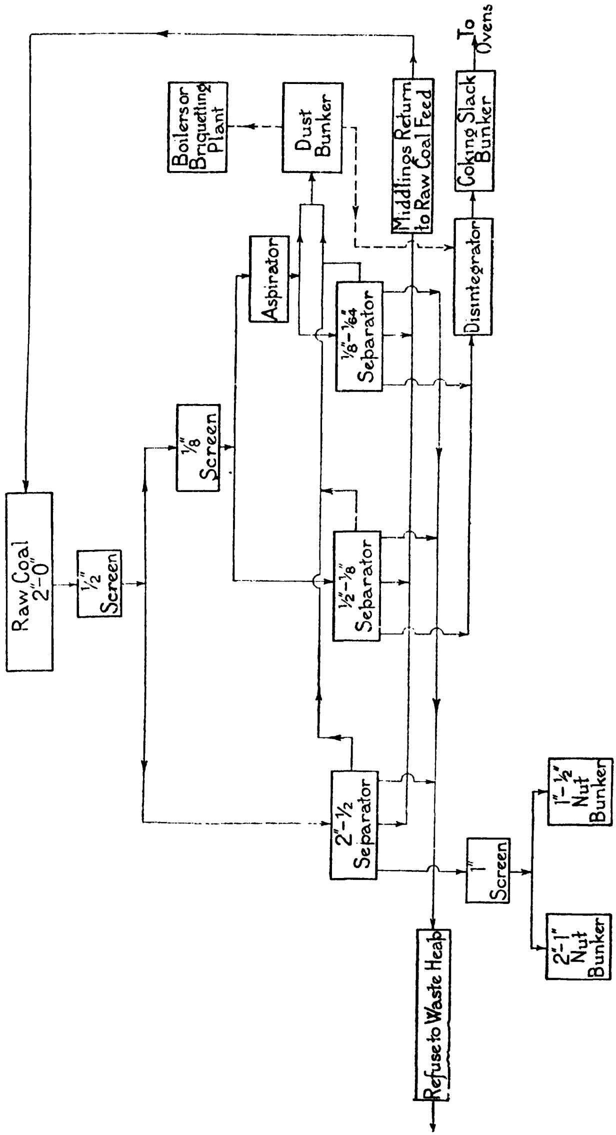


FLOW SHEET II.

(ALTERNATIVE TO SHEET I.)

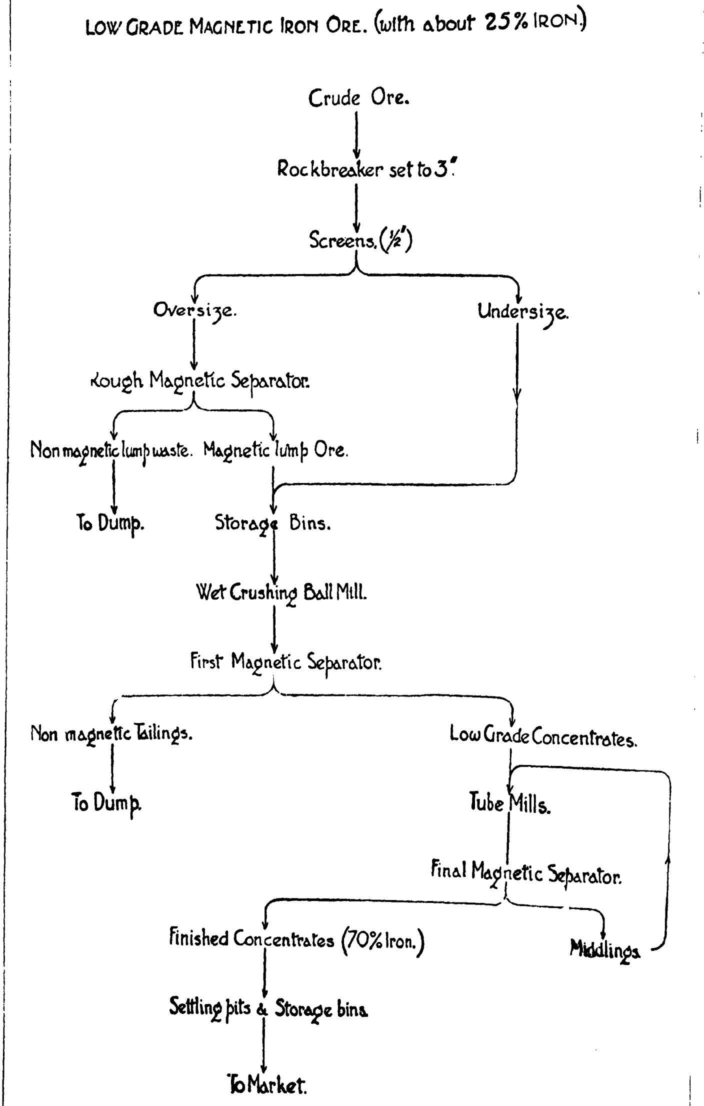


FLOW SHEET III.

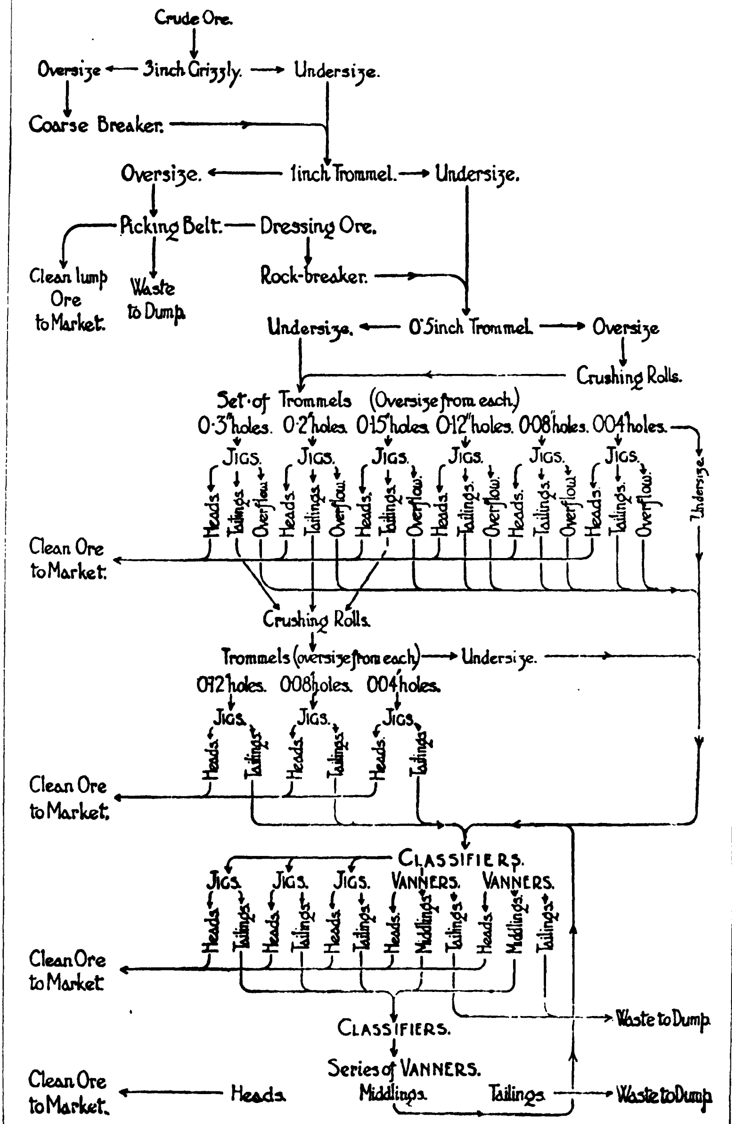


DRY CLEANING OF COAL.

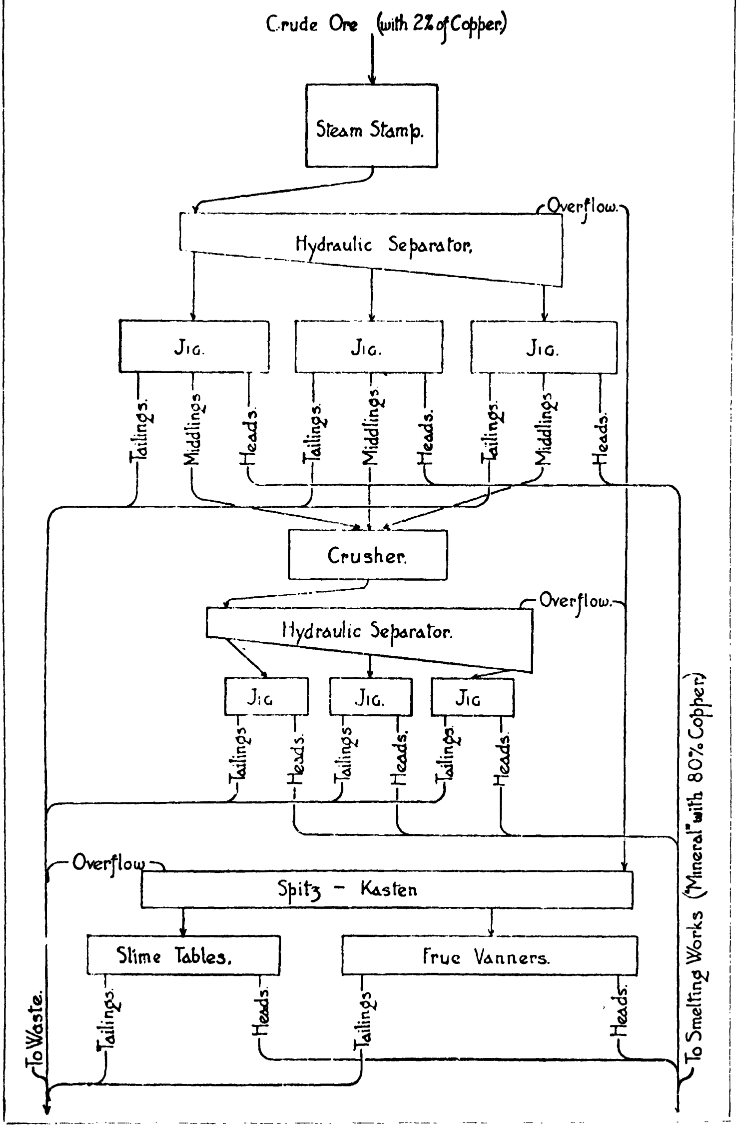
FLOW SHEET IV.



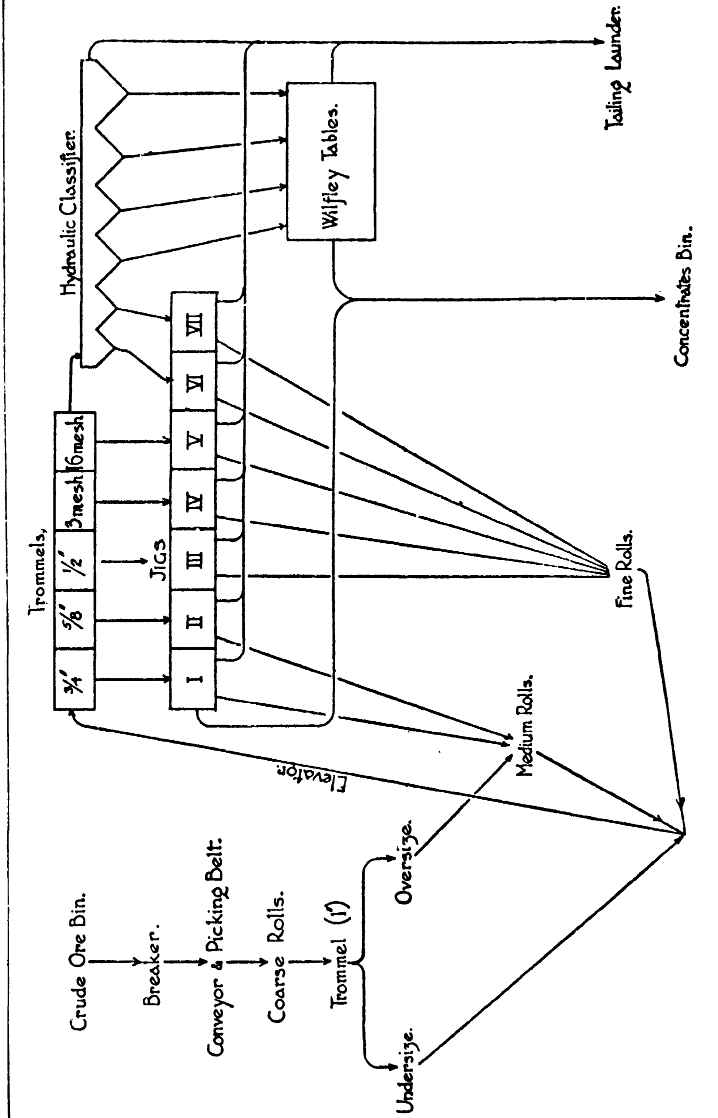
FLOW SHEET V.


Crude Ore.
Oversize
3 inch Grizzly.
Undersize.
Coarse Breaker.
Oversize.
1 inch Trommel.
Undersize.
Picking Belt.
Dressing Ore.
Clean lump Ore to Market.
Waste to Dump.
Rock-breaker.
Undersize.
0·5 inch Trommel.
Oversize
Crushing Rolls.
Set of Trommels (Oversize from each)
0·3" holes. 0·2" holes. 0·15" holes. 0·12" holes. 0·08" holes. 0·04" holes.
Jigs.
Heads
Tailings
Overflow
Undersize
Clean Ore to Market.
Crushing Rolls.
Trommels (oversize from each)
Undersize.
0·12" holes. 0·08" holes. 0·04" holes.
Jigs.
Heads
Tailings
Clean Ore to Market.
Classifiers.
Jigs. Jigs. Jigs. Vanners. Vanners.
Heads
Tailings
Middlings
Clean Ore to Market
Waste to Dump
Classifiers.
Series of Vanners.
Clean Ore to Market.
Heads
Middlings
Tailings
Waste to Dump

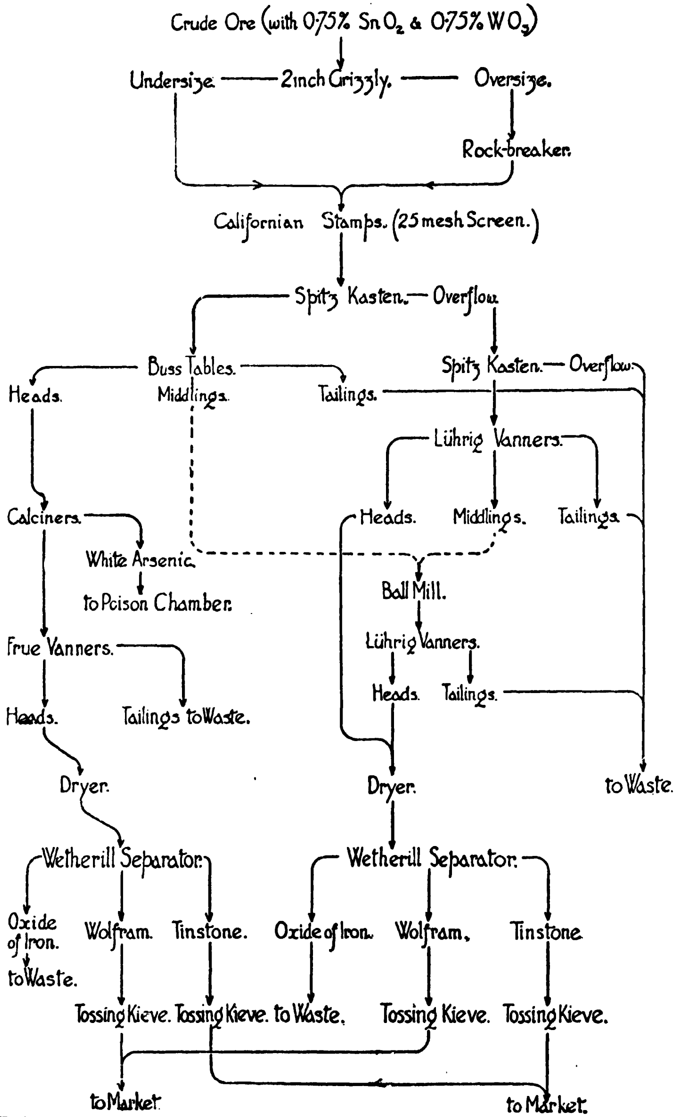
FLOW SHEET VI.



FLOW SHEET VII.

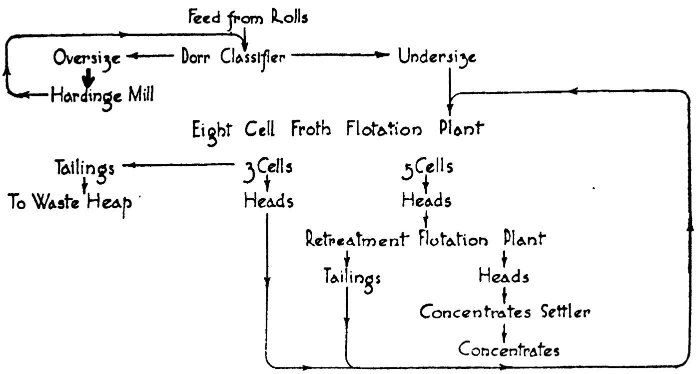


FLOW SHEET VIII.



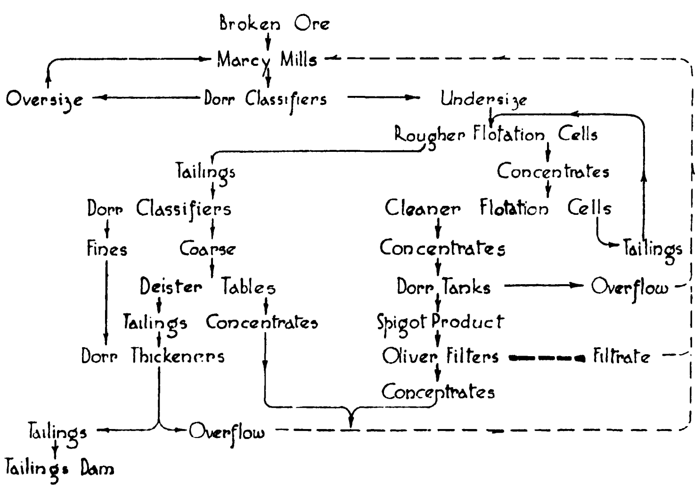
FLOW SHEET IX

Where the Frothing Agent is added in the Flotation Boxes.



FLOW SHEET IXa.

Where the Frothing Agent is added during Grinding.



GOLD MILLING.

With the improvements in the cyanide practice and also with the exhaustion of the rich free milling surface ores of various camps, which rendered such improvements necessary, have come changes in the process of treating gold ores. Instead of the simple Californian practice of crushing by 900 lb. stamps to about 40 mesh, amalgamating either in or outside the battery, concentrating the sulphides on vanners or shaking tables, chlorinating the concentrates, and sometimes treating the tailings by cyanidation, the 'all-sliming' process has come into force.

As practised on the Rand, the ore is screened through grizzlies with bars 2¾ ins. apart, the oversize going to a sorting belt, thence to a crusher. The ore passes into a feeder, thence to stamps weighing 2,000 lbs. per head, where it is crushed to pass a 10-mesh discharge screen. The pulp is classified in cones and the coarse product delivered to tube mills where it is slimed. The product of the tube mills, together with the overflow from the classifying cones, is delivered to the amalgamating tables. From the tables it is again classified, the sands being cyanided direct, and the slimes treated by air agitation, then filter-pressed or vacuum-filtered.

The most vital changes from the old practice to the new have been: the use of the heavy gravity stamp merely as a preliminary coarse crusher for the tube mills; the adoption of the tube mills for fine grinding and the employment of the tall, air-agitation tanks and of filters.

The practice is modified in each district, the modification depending largely upon the nature of the ore. In some camps silver ores are being successfully treated by fine crushing and cyaniding.

Cyanide Process.

This process may be conducted in a number of different methods, which may be classified as follows:—

I. Coarse crushing and direct cyanidation (exceptional).

II. Fine crushing with:—

- A. Simultaneous cyanidation.
- B. Subsequent cyanidation without classification.
- C. Subsequent cyanidation, preceded by classification into (*a*) sand, (*b*) slime.
  - (*a*) Sand cyanidation, generally by percolation.
  - (*b*) Slime cyanidation, performed by
    1. Decantation.
    2. Agitation (mechanical or pneumatic) and filtration.
    3. Direct cyanidation in filters.
- D. All sliming methods with simultaneous or subsequent filtration.

(*H. Louis*, '*Handbook of Metallurgy*,' I, p. 1031, 3rd edn., Schnabel and Louis.)

Filtration methods may be classified as under:—

I. Suction filters, divided into:—
- (*a*) Filters forming a thin cake, continuous acting.
- (*b*) Filters forming a thick cake, intermittent.

II. Pressure filters, divided into:—
- (*c*) Ordinary filter presses.
- (*d*) Sluicing filter presses.
- (*e*) Filtering chambers or cylinders.

III. Centrifugal filters.

(*S. J. Young*, '*Trans. Amer. Inst. M.E.*,' xlii., 1911, p. 752.)

Precipitation is performed by zinc shavings, zinc-lead shavings, zinc dust, aluminium electrolysis.

An average cyanide plant may be roughly estimated to cost complete $1,000 per ton treated per day in the Western States. An average cost of cyaniding, including crushing, is now about $1.50 per ton, but varies within very wide limits.

Stamps.

The duty of stamps depends upon the nature of the ore, the weight of the stamp, its speed and drop, the size of the aperture of discharge screen, and the height of the discharge. The following table gives the duty per stamp in various districts :—

DUTY OF STAMPS.

Mine or District.	Weight of Stamp.	Drop in Inches	Number of drops per Min.	Screen Aperture.	Discharge in Inches.	Tons Crushed per head per 24 Hrs.	Remarks.
California . .	900 lbs.	7	90	900 per sq. in.	4	3	Gold Quartz
Melones, Cal. .	1,000 "	7	104	20 mesh	—	4·41	
Tonopah Mill, U.S.A.	1,050 "	6½	104	slot ·04 in. wide	2½	5·09	Gold and Silver
Montana-Tonopah, U.S.A.	1,050 "	7	100	20 mesh	3	3·3	Gold and Silver
Tonopah Belmont, U.S.A.	1,050 "	6	104	24 mesh	—	3·7	"
Goldfield, U.S.A..	1,050 "	7	108	12 mesh	—	6	"
Komata Reefs Mine	1,030 "	6½	100	4½ holes per in.	4	5	Gold
Luipaards Vlei Rand	1,629 "	8½	98·6	200 holes per sq. in.	4	9·6	"
Simmer & Jack East	1,550 "	8	96	200 holes per sq. in.	3½	8·3	"
Waihi Grand Junction	1,110 "	7	105	5 to 10 mesh	1½	7·6	"
Oroya Brownhill.	1,100 "	7¾	108	10 mesh	2	6·48	"
Ivanhoe . .	1,192 "	7½	104	225 mesh	2	5½	"
Gt. Fingall . .	1,150 "	8	106	12 mesh	3	7	"
Mysore Gold Mine, Mysore, India	1,150 "	7·5	96	1,200, 625 and 400 per sq. in.	3	3·63	"

It is claimed that the new 2,000 lb. stamps of the Goldfields (S.A.) are capable of crushing 26 tons of ore per stamp per 24 hours by introducing an elimination screen to ' bypass ' the fine material from the rock bins direct to the tube mill and using screens with aperture of ·284 inch.

Amount of water varies on the Rand, from 5 to 10 tons of battery water to one of ore, depending upon the duty. In Western Australia the amount of water required is 4 to 8 gallons of water per stamp per minute.

Power per stamp.—The 2,000 lb. stamp requires 5 h.p. per stamp. In Nevada, for 1,250 lb. stamp 3·5 h.p. per stamp is used. At the Goldfield Con. in Nevada with 140 stamps, fine grinding and cyaniding, 1,600 h.p. is supplied to the mill or 1·73 h.p. per ton of ore milled.

COST OF EQUIPMENT OF GOLD MILLS.

The following is the comparative cost of gold mills, including all machinery and buildings :—

Mill.	Number of Stamps.	Cost of Machinery and Plant.	Cost per Stamp.
Rand Mill	200	£334,000	£1,670
New Kleinfontein	200	420,480	2,102
Wit Deep	200	346,210	1,731
Robinson Central Deep . . .	100 and Tube Mill	234,013	2,340
Great Fingall	"	193,745	1,937
El Oro	"	160,000	1,600

In Nevada, the cost of complete plants doing fine grinding and slimes filtering is 1,400*l.* a stamp.

Tube Mills

are used as secondary crushers following the stamps. On the Rand they are most efficient when fed with ore that has been crushed through a screen the apertures of which are ·27 in. or say ¼ in. They are here 25 ft. in length by 5½ ft. in diameter, and one tube mill crushes the product of 10 stamps. Each tube mill requires 100 h.p. to drive it. In New Zealand tube mills are 16 ft. by 4 ft. revolving at 29 r.p.m., and driven by 25 h.p. motor. The feed is 1½ parts of water by weight to 1 part of ore. In 22 months a tube mill ground 28,500 tons of hard quartz, 50 per cent. of which passed through a 200-mesh screen.

At Pachuca 90 h.p. is used for a 5 ft. 6 ins. by 22 ft. tube mill. Output 140 tons per day through a 60-mesh screen.

At Waihi Grand Junction tube mills 19 ft. 3 ins. long by 4 ft. 8 ins. diameter require 42 h.p. Those 16 ft. 3 ins. long by 4 ft. 3 ins. diameter require 25 h.p. They consume 2·89 lbs. of flint pebbles per ton of ore crushed at stamps. They are run at 27 r.p.m.

Chilian Mills.

Chilian mills are employed either for crushing preliminary to amalgamation, in which case the gold is partly amalgamated in the mill itself, or they are used in lieu of the tube mill as fine grinders following the stamps.

For crushing ores for direct amalgamation the so-called slow speed Chilian mills are well adapted. Machines of this type were extensively employed in Russia and Siberia. They are expensive in first cost, but the power, labour, and repair costs are very low. Furthermore they are notable for their simplicity, requiring little close or skilled attention.

The standard type of Chilian mills used in Russia is 7 ft. to 10½ ft. in diameter with 2 runners 5½ to 6½ ft. in diameter, and from 7 to 12 ins. in width. The speed is 11 r.p.m. The discharge is 8 to 10 ins. Feed is about 1 in. and discharge screen slotted. The water is 6·5 to 10·1 times the ore in weight and duty is 16 to 24 tons per 24 hours. The power required is 7 to 12 h.p.

A slow speed Chilian mill was employed in California on hard close quartzose gold ore. It ran at 8 to 10 r.p.m. It had four runners each weighing 1,200 lbs. travelling on a steel die 3 ins. thick with 5 ins. face and 7 ft. outside diameter, 4½ tons of rock in a tank above and sustained by the axles giving additional crushing weight. Twenty tons of ore per 24 hours were crushed with a 4 in. discharge, and 88·1 per cent. extraction obtained, 0·85 per cent. of the ore passed through a 100-mesh screen. One man per shift handled the mill. Cost of ore crushed 6*s.* per ton. Gasoline power was used consuming under heaviest load 2·17 gallons of distillate per hour. (*A. McLaren.*)

Rapid-running Chilian mills were used as fine grinders after the stamps in the Goldfield Consolidated mill in Nevada on fairly soft ore. There 140 stamps were in operation and six 6 ft. Chilian mills crushing from 4-mesh to 16-mesh were installed. These were operated at 32 r.p.m. Their capacity was 75 tons per 24 hours and the power required 35 h.p. each. The cost of crushing by these mills is given at 5*d.* per ton of ore milled. (*J. W. Hutchinson.*)

At Cripple Creek a 6-ft. Chilian mill running at 33 r.p.m. reduces 4 tons of ore per hour, 60 per cent. of which will pass a 150-mesh screen. The feed passed a ⅜-in. aperture.

Chilian mills are rarely used to-day.

Rolls.

Where a relatively small tonnage is to be crushed coarse, as for example when coarse concentration is employed or where the ore is porous and is cyanided coarse, it is the practice to use rolls. They are cheaper than gravity-stamps in first cost, repairs, labour and power, and are not complicated by screens.

In South Dakota the ore, a hard, close-grained Cambrian quartzite, carrying from 5 dwts. to 7 dwts. per ton, is crushed by rolls. The ore is fed to two 16- by 36-in. rolls used as cracking rolls and two others of the same size are used as finishing rolls. The final product goes through a ¼-in. mesh direct to the cyaniding vats. The rolls require 100 h.p. The milling and cyaniding cost is 2*s.* 2*d.* per ton and 14,800 tons of ore are treated monthly.

Ball Mills.

Krupp ball mills are used extensively in Kalgoorlie, where 44 are at work. Size No. 5 will crush up to 45 tons of ore per day consuming 25 h.p. No. 8 will crush 90 tons per day requiring 50 to 60 h.p. About 45 per cent. of the product from a 25-mesh screen will pass 150-mesh. It is charged with 1 or 2 tons of steel balls and revolves from 21 to 25 times per minute. They work well on either wet or dry ores. The wear and tear of liners, hunch and scoop plates cost 4·64*d.* per ton milled.

Ball mills are used extensively in connection with cyaniding.

Milling Costs.

The following are the milling costs of different stamp mills :—

Mill.	Stamps.	Tons Crushed Monthly.	Cost of Milling per ton.	Rate of Wages per day.	Remarks.
Tonopah, U.S.A.	100	15,000	12*s.*	18*s.*	Silver Gold ore
Goldfield Con., U.S.A.	140	29,500	9*s.*	18*s.*	
Ivanhoe	100	16,600	8*s.* 9*d.*	12*s.* to 14*s.*	Bromo-cyanide treatment of slimes
Oroya Brownhill	50	8,950	17*s.*	12*s.* to 14*s.*	
Gt. Boulder	—	14,800	12*s.* 4*d.*	12*s.* to 14*s.*	Dry crushing with Griffin mills. Roasting.
Melones, Cal.	100	12,000	1*s.* 1½*d.*	10*s.*	Cheap power, labour and supplies
New Unified (Rand)	—	10,000	1*s.* 4*d.*	—	Milling only
Simmer East	—	—	2*s.* 6*d.*	—	

On the Rand, cost of milling of those companies using tube mills varies from 4*s.* 10*d.* to 8*s.* 8*d.* per ton of ore.

In California, with free milling ores and cheap water power, milling costs average from 2*s.* to 4*s.* per ton. In Alaska, with large tonnages of soft, free milling ore treated daily and no cyaniding or fine grinding the costs are about 1*s.* per ton of ore.

Horse-Power per Ton of Ore Treated per Day.

Type of Mill.	Mesh.	Horse-power.
Stamps and vanners	20 to 40	0·75 to 1·0
Coarse concentration	10 to 20	0·5 to 0·8
Combination-stamp	16 to 30	1·5 to 1·75
Chloridising stamp (dry)	to 16	2·0 to 2·5
Chloridising stamp (wet)	to 40	4·0 to 4·5
Magnetic separator	—	0·25 to 0·5
Cyanidation (dry) roll-crushing	20 to 30	0·5 to 0·8
Cyanidation (wet), stamp crushing and sliming	to 80	0·75 to 1·5

(*J. A. Barr*, 'Testing for Metallurgical Processes.')

Standards Adopted by the Institution of Mining and Metallurgy.

1 ton = 2,000 lbs. (avoirdupois).
1 miner's inch = a flow of 1½ cubic ft. of water per minute.
1 sluice head = " 60 " "
1 gallon of water = 10 lbs. at 34°.

Temperatures are to be expressed in degrees centigrade.

Screen apertures and diameters of wire are equal, giving 25 per cent. screening area.

GOLD AND SILVER TABLES.

It is the practice of Continental and South American engineers to report the values of ore in grammes per metric ton. English engineers generally employ the method of quoting values in grs. and dwts., in some cases per long ton of 2,240 lbs., in others per short ton of 2,000 lbs.

TABLE for Conversion of Grammes per Metric Ton into Dwts, etc., per Long and Short Ton.

Grammes per Metric Ton.	Per Short Ton.		Per Statute Ton.		Grammes per Metric Ton.	Per Short Ton.		Per Statute Ton.	
	Dwts.	Grs.	Dwts.	Grs.		Dwts.	Grs.	Dwts.	Grs.
1	0	14	0	15·7	6	3	12	3	22
2	1	4	1	7	7	4	2	4	14
3	1	18	1	23	8	4	16	5	5
4	2	8	2	15	9	5	6	5	21
5	2	22	3	6	10	5	10	6	13

TABLE for Conversion of Dwts. per Ton into Grammes per Metric Ton; the equivalent of Dwts. per Short Ton is given in the first column, per Statute Ton in the second.

Dwts.	Grammes per Metric Ton.				Dwts.	Grammes per Metric Ton.			
	Dwts.	Grs.	Dwts.	Grs.		Dwts.	Grs.	Dwts.	Grs.
1	1	71	1	53	6	10	29	9	18
2	3	43	3	06	7	12	00	10	71
3	5	14	4	59	8	13	71	12	24
4	6	86	6	12	9	15	43	13	28
5	8	57	7	65	10	17	14	15	31

Alluvial or Placer Mining.

Deposits of auriferous or tin-bearing alluvial gravels are termed placers, alluvial drifts, or more simply alluvials. They are classified as shallow or superficial and deep-lying, sometimes called deep leads. Deep leads are accumulations of gravel from extinct or ancient rivers. They are mostly capped by igneous flows, as in California and Australia, though sometimes they may be buried under recent gravels or drift, as at Lightning Creek, B.C., and the Lena Goldfields in Siberia.

The methods of mining used to exploit deposits of this character resemble pillar and stall in some cases and long wall in others. In California the gravel beds can be attacked by means of adits which also serve as drains for the large amount of water present. When the topography of the ground permits adits being used, the cost of mining is low, as at the Hidden Treasure, California, where the costs average 4*s*. 6*d*. per cub. yd. of gravel washed. In the Deep Leads of Victoria, where the pumping charges were unusually large (eleven million gallons per day being raised from one mine), the total cost per cub. yd. was 12*s*. to 15*s*. At the Lena Goldfields the total cost was from 20*s*. to 30*s*. per cubic yard of gravel washed—i.e. when worked by the Lena Gold Fields Company, but it has been for some years under Soviet control.

Superficial Deposits.—All gravel under 60 feet in thickness may be termed shallow or superficial gravels. Surface gravels are worked by the following methods :—

Panning.—A miner can pan ½ cub. yd. to 1 cub. yd. of gravel per 10-hour shift, depending on character of ground and nearness to water; 100 pans per day is a good day's work. Gravel in place is usually estimated at 3,000 lbs. per cub. yd., and the standard sheet iron pan 16 ins. diam. at top and 10½ ins. diam. at bottom will hold 20 lbs.; 150 pans full will make a cub. yd. of dirt.

Rocker.—A man can rock ordinarily from 2 to 3 cubic yards of gravel per day, and save, in practical mining, 90 per cent. of the gold. In alluvial examinations the rocker plays a most important role, and is used in conjunction with the pan for determining the gold contents of any parcel of gravel. When used for this purpose, the design of the rocker differs slightly from that of the ordinary miner's rocker. A well-designed rocker, carefully employed, should save practically all the gold. The plans of most rockers, as published from time to time, are those of the ordinary mining rocker, which is too short, and set on too great a grade for use in saving fine gold. If much fine gold is present in the gravel, it will be advisable to use a strip of common Brussels carpet, laid along the floor of the rocker. This carpet is held in place by cleats and so can be easily removed in cleaning up.

Long Tom or short sluice can handle 3 to 10 cub. yds. of dirt per day, depending on the amount of water available and the character of the dirt.

Hand Sluicing.—Capacity of a sluice box depends upon the amount of water, grade of sluice. deposition of tailings, and character of gravel. No capacity can be given. Grade of sluice varies from 2 ins. to 21 ins. per box (12 ft.). An ordinary sluice box is made from 2-in. planks and is 12 ins. wide by 12 ft. long.

Grade and Duty of Sluices.

Grade and Duty of Sluices.—The relation of actual width to the best ratio of quantity may not be very significant, but there is a notable increase in efficiency in passing from the smallest widths to the larger, and boxes less than 1 ft. wide are to be avoided, especially if the gravel is coarse. Treating 30- to 40-mesh sand on a 1 per cent. grade (say 1¼ in. per 12 ft. box), one sluice head of water requires a channel 2 ft. wide to enable it to attain maximum duty. On a 2 per cent. grade, half a sluice head requires 2 ft. of width, the quantity of water required for maximum duty with a fixed width diminishing with increasing grade, but at what rate there are insufficient data to determine. Apparently the best width required for constant quantity of water varies directly as the grade. Coarser material requires a greater depth of water, and for 4 to 6 mesh gravel on a 2 per cent. grade, 1 ft. of width per sluice head is indicated. Variations up to 50 per cent. in the water make little difference to its duty, consequently the actual position of the optimum is not easily determined. Naturally the capacity of the sluice itself is correspondingly varied, and the mode of transportation and the saving are doubtless affected in any case.

The rate at which duty varies with slope is the resultant of so many variables that no law can be stated for all cases. In the neighbourhood of the best conditions the Utah experiments support the approximate rule that duty (water, width, and material being constant) varies as the 1·75th power of the grade. A single experiment with natural gravel on low grades by the U.S. Geol. Survey gave the 1·84th power. (*R. T. Hancock.*)

Good modern practice provides about 1 ft. of sluice-box width for every 2 cub. yds. of average material treated per hour, and 3,000 gallons of water per cub. yd. treated on a grade of 6 ins. to 12 ft.

Besides the ordinary feeding of sluice-box by means of hand shovelling, many modified methods of feeding and disposing of tailings are employed. In Russia and Siberia the favourite method in use is to transport the gravel to and from the sluice box (here greatly modified from the plain open sluice-box) by the employment of horses and small carts. This is a most expensive system and the yardage treated is small. Other variations are: the use of wheelbarrows, tramming by cars, handling gravel by means of derricks, digging gravels by means of steam shovels, and loading into cars or direct into sluice, feeding sluice by means of scrapers operated by horses or steam power (drag-line scrapers).

The methods of preventing accumulation of tailings are to deposit them in running water or to remove them by means of drags, belt or link elevators, or hydraulic elevators.

HYDRAULIC MINING.

Hydraulicing proper is the method of mining by which jets of water under high pressure are thrown against a gravel bank, thereby undermining the gravel, which falls and is carried by the current into the sluice. The necessary pressure is obtained either by natural means, *i.e.* by conveying the water from some source having a greater elevation than the working face, or by pumping directly into the supply pipe by powerful pumps. In the former case water is conveyed to the working by means of pipes, flumes or ditches. Ditches are generally most convenient and economical for conveying water for long distances, but sometimes the nature of the ground is such (either too porous or too rocky and difficult to excavate) that a flume becomes cheaper. When lumber is abundant wooden flumes are built, but in many cases, especially in the tropics, where wood rots rapidly, sheet steel flumes are preferable. Steel flume per foot, including rivets, angle iron, etc., ready for transport, costs from 4*s.* to 1*l.*; the sizes vary from a cross-section of 1·28 sq. ft. to 20-48 sq. ft. From the ditch or flume the water is fed through a strainer into a pressure box. From here, a pipe leads to the Y's, which in turn lead to the monitors or giants.

Monitors or Giants.

Monitors or Giants.—These are appliances consisting of a base for attachment to the pipe line and a long conical reducing piece, arranged with one or more joints so that it can be rotated in any direction horizontally and through a very wide angle vertically. The smaller or conical end is fitted with a nozzle butt to which different sizes of nozzles may be attached as circumstances may require. Modern giants are fitted with deflectors, attachments on the butts whereby the giant is turned or deflected by means of the force of the water itself. The high-pressure giants also have ball-bearing joints.

Nozzles from 2 ins. up to 10 ins. are used, working under heads or pressure from 100 up to 600 ft. or more.

The amount of gravel which can be washed down will depend upon its nature, upon the volume and pressure of the stream of water issuing from the nozzle, and is also dependent in no small degree upon the skill of the operator who handles the giant.

The following gives the approximate amount of gravel washed under various heads:—

Effective Head in Feet.	Size of Nozzle.	Flow in Cub. Ft. per Minute.	Approximate amount of Gravel in 24 hours in Cub. Yds.	Weight of Giant. Lbs.	Price of Double Jointed Ball Bearing Giant. £
100	2 ins.	104·88	160	120	28
200		148·32	230		
300		181·61	280		
400		209·82	320		
100	4 ins.	420·06	650	225	55
200		594·00	930		
300		727·56	1,130		
400		840·12	1,300		
100	6 ins.	944·58	1,460	335	66
200		1335·72	2,000		
300		1634·08	2,540		
400		1889·16	2,920		
100	8 ins.	1679·82	2,600	600	97
200		2375·46	3,600		
300		2909·58	4,500		
400		3239·70	5,300		
100	10 ins.	2624	4,080	690	107
200		3711	5,760		
300		4545	7,050		
400		5248	8,140		

(*Hendry.*)

Hydraulic Monitors and Pelton Wheel Jets.

J. J. Garrard (*Trans. I.M.M.* 1917) gives a table of discharges in cubic feet per minute for different sizes of nozzles and various heads derived from the formula: Delivery in cub. ft. per min. $= 3{\cdot}75\, d^2\sqrt{P}$, where $d =$ diameter of nozzle opening in ins. and $P =$ gauge pressure in lbs. per sq. in., or effective head $\times 0{\cdot}4326$. An equivalent formula given by him is $2{\cdot}468\, d^2\sqrt{h}$, where h = effective head in ft. The factor 2·468 allows for a nozzle contraction coefficient of 0·94, whereas Hendry's figures in the above table are based on the theoretical value 2·624, and make no such allowance.

Costs.

The costs per cub. yd. were at North Bloomfield 2·05*d.* per cub. yd. and at La Grange 3*d.*

In British Columbia one and a half million cub. yds. were mined and washed at a working cost of 0·97*d.*

In Northern California placer ground containing only 1·26*d.* per cub. yd. has been profitably worked. The amount of gravel washed was 1,251,399 cub. yds., yielding 6,330*l.* The profit obtained was 821*l.*

Hydraulic and other Elevators.

When a hydraulic mine possesses an insufficient slope of bed rock for moving material (the lightest grade permissible being 220 ft. to the mile) other methods of disposing of the gravel 'piped' must be employed. Where water under pressure is cheap and abundant, hydraulic elevators are employed. They are of the pipe type—either Hendry, Evans or Campbell and the Giant Elevator.

The pipe elevator operates on the principle of an injector. Water under pressure is discharged through a nozzle set within a steel jacket, creating a vacuum, and causing water and sand and gravel fed into intake to rise to a height corresponding to 10 per cent. of the head under which the water is applied.

Three sizes are made: 10-inch throat 3½–4½ inch nozzle, using 300 to 500 ins. of water; 12 inch throat 4½–5-inch nozzle, 500 to 750 ins. of water; 15-in. throat, 750 to 1,000 ins. They weigh about a ton and cost 250*l.*

In South Oregon, U.S.A., at the Weimer placer mine, two hydraulic elevators are employed to handle the gravels. One of them has a lift of 39 ft., and operating with a head of 330 ft., uses 11 cub. ft. of water per sec., besides taking all water from two No. 2 Giants. The second elevator has a lift of 9 ft., a head of 125, and uses 18 cub. ft. of water per sec., besides all the water from the first elevator. This makes a final discharge of 40 cub. ft. per sec.

Giant Elevators.

Besides undermining and cutting the gravel it has been discovered that gravel can be effectively 'driven' up a slope or incline to a raised line of sluices by means of the power of water under pressure. An elevator based on this principle is termed a 'stacker.'

In California a stacker was operated as follows: A chute was built sloping upwards at 30°. This was 4 ft. wide by 24 ft. long and constructed of 1½-inch planks. Riffles 2 ins. apart and with a drop of 2½ ins. between were laid across the bottom. These were covered by an iron grizzly. At the foot of this chute a 3-foot sump was dug in which was set up a hydraulic elevator with a 12-inch throat and worked by a 5-inch giant. This elevator, working under a 90-foot head, elevated to 16 ft. vertically the seepage, fine and water from two field-giants.

Twenty feet in front of the stacker was installed the 3-inch drive giant. Two 4-inch field-giants wash the gravel to the chute and the drive giant forces it up the stack, the fine material falling through the grizzly and down the riffles to the elevator, which raises it to the sluice-boxes.

The field-giants handled 600 cub. yds. per 24 hours, and the gravel could easily be raised to 25 ft. vertically by means of the drive giant. The cost of handling material exclusive of equipment was under 2½*d*. per cub. yd. (*S. S. Smith.*)

Ruble Elevator.

The Ruble elevator is constructed on the above principle, but it is more portable and does away with the hydraulic elevator. It is also self-contained, as the sluice-box is part of the machine.

In Idaho, U.S.A., this elevator has been tried with excellent results (*see* Porter, T.A.I.M.E., 1909, for a detailed description of this elevator). The following points are claimed to its merit: 1. It handles larger rocks with less water than other types of elevators. Handles boulders up to 1 ton in weight. 2. It dispenses with the boulder crew, either at the mine, ground, sluice-box or dump. 3. It saves the gold and the gold-saving department may be protected by lock and key. 4. The gravel is picked up in close proximity to the giants, the gold immediately extracted, and the boulders and other waste material dropped back on bedrock previously worked off, instead of being transported a long distance to the dump. It makes its own dump.

Mechanical Elevator.

In regions where water is neither cheap nor abundant, in handling gravel or tailings, resort must be had to some form of mechanical elevators. A mechanical elevator was erected some years ago on Bonanza Creek, Alaska. This was made of a frame of steel, 49 ft. high, mounted on wheels. The gravel was elevated by a bucket line of the close-connected type, 76 buckets in all, each with a capacity of 3 cub. ft. These discharged at a rate of 26 per min. directly into a sluice-box. The pit was drained by means of a 12-inch centrifugal pump, assisted by a hydraulic jet. The power was 200 h.p. in all, 35 for the bucket line and 165–175 for the pumps. The actual capacity per day was 1,500 yds. (*T. A. Rickard*). The first cost of this machine was high, and in actual running it proved costly in operation and repairs. Another machine of the same type, but employing a heavy dredge bucket, was erected in Eastern Canada.

Centrifugal Pumps.

Centrifugal Pumps as a means of elevating gravel and tailings have been extensively employed in Australia. It is claimed that with wood fuel at 7*s*. per ton of 50 cub. ft. and European rate of wages paid, to elevate tailings with pumps should not cost more than 2½*d*. per cub. yd. of solid. Thompson & Co., of Victoria, are the makers of gravel pumps suitable for this class of work. Their pumps vary in size from 3½ ins. to 16 ins. The capacity of solids per hour from 7 to 150 cub. yds. The indicated horse-power from 2½ to 583. The combined suction and delivery head from 10 feet to 100. An installation of this kind possesses the advantage of being cheaper to erect and easier to move than the belt or bucket type of elevators mentioned. However, where there is an undue amount of large stones present in the gravel, the latter type is preferable.

Pipe Lining.

Three rubber-lined pipes, each of 9 in. internal diameter and length of about 20 ft., have been in continuous operation on the discharge line from an 8-in. gravel pump at the Sungei Besi Mine, Malaya, for a period ranging from 18 months from the date of installation of the first pipe. These pipes have been subjected to a continuous flow of gravel, amounting to about 17,000 cub. yds. of solids per month, and a working pressure of about 17 lb. per sq. in. The three pipes, although differing somewhat in design, are all alike regarding the successful results of the rubber applied to them, there being little or no evidence of wear or abrasion on the rubber after this lengthy period of operation. The lining is of crepe rubber about ½ in. thick. The life of a lap-welded mild steel pipe $\frac{5}{16}$ in. thick on the same line, with frequent turning to ensure even wear, does not amount to an average of seven months. ('*Mechanical World*,' Jan. 21, 1927.)

See also 'The Use of Rubber in the Mining Industry,' by H. P. Stevens and F. B. Powell, *Trans. Inst. Min. Met.*, vol. xxxvi. (1926–27), p. 263.

Dredging for Alluvial Minerals.

(Contributed by Harold E. Fletcher, A.I.N.A., M.I.Mar.E.)

The most important form of alluvial mining is dredging, for the recovery of gold, platinum, and tin. The greater part of the platinum recovered in Siberia and Russia is by dredging, and that of the tin in the Malay States. When all conditions requisite for successful working are present, when the ground has been properly proved * and a well-designed and effective dredge installed, then this type of mining is the safest and surest of all forms of mining investment.

Careful systematic boring is necessary to get true values. Dredging properties call for certain conditions, and unless these exist the success of the enterprise suffers. The conditions are:

(1) The presence of the mineral, which should be evenly distributed throughout the pay dirt.
(2) Sufficient ground to warrant the installation of at least one dredge.
(3) Soft bedrock, free from pinnacles.
(4) Absence of large boulders.
(5) Absence of permanently frozen ground.
(6) Absence of great amount of clay.
(7) Absence of heavy floods.
(8) Surface of an even contour, not too rough and broken.
(9) The ground should retain sufficient water to float dredge (50 miner's ins.), or available water supply from a river.
(10) The economic conditions, *i.e.* labour, transport, climate, vary inversely with amount of mineral in ground.

The life of a well-designed dredge may be calculated at from twelve to fifteen years, provided it is kept in good repair. Repairs for the first three years would be normal (excepting accidents), such as the replacing of bucket pins and bushes, ladder roller spindles and bushes, screen plates, tumbler bearings, and usual power plant replacements, and later on in the life of the plant tumblers and shafts, main gears, buckets, repairs to pontoon, etc.

Broad flood plains or ancient lake gravels form more suitable dredging ground than does deposit situated in narrow rocky gulches. Frozen ground cannot be dredged without preliminary thawing. Dredging of gravels lying in rapid rivers is risky, as there is always danger of losing the dredge by sudden floods. Gravels from 10 to 80 ft. below water and 20 ft. above have been successfully dredged.

The modern dredge consists of a hull or pontoon of wood or steel, upon which is mounted the digging device, a bucket ladder capable of being raised and lowered, the washing plant comprising screen, tables or launders, the pump of the centrifugal type for water supply to screen and tables, the stacker or elevator to dispose of the tailings well astern of the dredge, and the power plant. The buckets raise the ground and discharge into the revolving perforated screen, the fines pass through the perforations on to the tables, and the stones and large material pass through the screen to the tail chute and on to the tailings stacker. The dredge can cut its own passage. The cutting depth and manœuvring lines are worked by special multi-barrelled winches, controlled by one man, the winchman.

'Suction' dredges are used at times; they are only successful in light and free ground with level bedrock. The power required has been found to exceed that of the bucket dredge of equal output. With the suction dredge it has been found difficult to control the surges of water that sweep the tables when the spoil will not draw easily. This type of dredge has not given good results on tin in the Straits of Malaya.

'Bucket' dredges are classified in two types, the New Zealand and the American; the latter has been evolved from the former to suit conditions existing in California and other parts. The principle is the same, but the American type is associated with heavier construction, larger buckets of the close-connected type instead of open, spuds (adjustable piles) instead of a headline, the raised pilot (or central control) house, and other modifications. Both types have their advocates and both their utility.

The New Zealand type was associated with open-connected buckets, moderate power, simple engine for driving all units and light construction, but to-day this type is built of fairly large capacity and heavier construction. In its smaller sizes for lighter ground its first cost is in its favour, also for prospecting work or proving a dredging property; also where difficulties of transport prevent the use of the heavier type of dredge.

Transport.—Dredges are generally shipped in pieces for re-erection, the steel pontoons being riveted and launched at site. The heaviest parts to transport are the main engine castings, the boiler shell, the tumbler castings (mostly steel), crown wheel castings, winch barrels, screen ends, etc. The 'Tumblers' weigh from 15 cwts. in the small to 15 tons in the large sizes; boilers stripped, from 3 to 10 tons; watertube boilers, the heaviest part being the steam drum, from 7 to 60 cwts. See 'Air Transportation of Gold Dredges in New Guinea,' by C. A. Banks, *Bull. Inst. Min. Met.*, August 1932, No. 335.

Hulls.—These have to be strongly designed to stand the stresses. Wood pontoons are not entirely satisfactory and generally have to be replaced during the life of the machinery; they suffer damage by ice in cold countries, and are difficult to build to stand stresses, and much heavier than steel.

* See *Alluvial Prospecting*, by C. Raeburn and H. B. Milner. 1927. Also 'The Valuation of Alluvial Deposits,' by W. R. Rumbold, *Trans. Inst. Min. Met.*, vol. xxxvii. (1927-28), p. 437.

Power.—Where wood and coal are plentiful, steam is cheapest. In Malaya the Rawang coal (8,000 B.Th.U.) is used in water-tube boilers with chain grate stokers; steam pressure 160 lbs., and 100° superheat. Where oil is available at reasonable cost, it can be burnt under boilers or used in Diesel or semi-Diesel engines, and these have proved more economical than steam.

A gold dredge with 9 cu. ft. close-connected buckets is running in South America with semi-Diesel engines, also smaller dredges in Burma, etc. Full Diesel engines require the services of a specially skilled engineer. Several of the recent tin dredges have Diesel-electric generating plant on board running electric motors. Producer-gas plant and engines are used in Burma, Malaya, Africa, etc., running on charcoal, driving direct or driving generators to supply current to the several motors. Where electric current is available and power can be transmitted to the dredge by cable at reasonable cost, and transformed on the dredge to the required voltage, it is in favour of low running costs, but it is generally accepted that it does not pay to build a special generating plant unless the ground will carry four or more dredges. Waterfall power to generate current is also being used.

The horse-power installed on gold and platinum dredges varies according to the capacity of bucket, dredging depth, etc. The distribution of power is generally as follows:

Bucket chain . . .	33% of total power.	Screen . . .	10% of total power.
Water-pump . . .	30% ,, ,,	Elevator . . .	12% ,, ,,
Winch . . .	15% ,, ,,		

On tin dredges, where larger water supply is necessary and higher water pressure, the pump power proportion is larger. On large dredges a separate winch is often fitted to deal with the bucket ladder.

Gold Dredges.

Larger types of American design are used in America, Siberia, etc., in addition to the open-connected type. Gold is saved on tables covered with cocoanut matting and expanded metal; the gold by reason of its heavy specific gravity settles in the mats, the dirt being washed down the tables overboard, the mats being taken up at periods and washed. With properly designed tables and adjustable water supply, quite fine gold can be saved. Platinum is easier to save owing to its higher specific gravity.

Tin Dredges.

The main difficulty is not in raising the ground, but in treating it efficiently; the tables require to be of large area and the water supply larger than for gold dredging. Tables or 'launders' are genearlly from 3 to 4 ft. wide and up to 80 and 100 ft. long, wood riffles are fitted, and the wash agitated gently by hand-raking or mechanical means to allow the tin to sink through the wash and settle on the bottom of the launders. Owing to the large area of the tables the pontoons are of larger dimensions than for gold dredges; tables up to 7,000 to 8,000 sq. ft. area in two tiers are in use. Owing to the difficulty of carrying such tables, some of the latest dredges have been designed with jigs and classifiers, or a combination of riffles, launders, jigs, and classifiers, which will allow of better treatment and the saving of finer tin. The use of jigs allows of better saving adjustment. With skilled attention, up to 92 per cent. of the boring values have been obtained. Clay in the ground in both gold and tin dredging presents great difficulty to the dredge designer, and to break it up and release the values it contains call for special arrangements, such as puddlers, high-water pressure, etc.

Where large boulders or heavy clay is met with a 'tray' connected bucket chain is sometimes used, the trays being castings equal to the base of the buckets and contributing in output each about 25 per cent. to 30 per cent. of the bucket capacity.

The design for tin dredges is tending towards the American type in regard to the design of the driving gear, buckets, tumblers, winches, etc. The latest dredges have close-connected buckets of cast manganese steel and nickel-chrome steel pins, cast manganese tumblers, etc.

There has been a vast amount of capital put into tin dredging in the Malay States and Siam, the number of dredging plants reaching towards 200. Some of these companies have paid 25 per cent. dividend for a number of years.

The largest English-built dredges have close-connected buckets of 12 cu. ft. capacity, a large proportion 8 and 9 cu. ft. The open-connected dredges range from 6 to 15 cu. ft. buckets. Dredging depth ranges from 30 to 80 ft., and dredges have been built to dig 120 ft.

Output varies according to bucket capacity and the rate at which the ground can be efficiently treated. The open-connected bucket is run from 14 per minute for the smaller to 12 per minute in the larger sizes. The close-connected bucket from 28 to 18 per minute.

Maximum output is calculated:

$B \times S \times 60 \div 27 =$ cu. yds. max. per hour. $S =$ number of buckets delivering per minute.
$B =$ capacity of buckets in cu. ft. $27 =$ number of feet per cubic yard.

Roughly, 1 cu. yd. is calculated $1\frac{1}{2}$ tons.

To obtain the estimated output, one-third is deducted from the maximum, but this is not always reached; buried timber and large boulders tend to decrease output. The removal of barren overburden (to reach pay dirt) and difficulties of cleaning up bedrock lessen the profitable dredging time. A further factor is time lost in cleaning up the tables or launders to recover the gold or tin. Working time varies: many dredges have averaged over 600 hours per month.

Working costs vary with difficulties of ground, cost of fuel, and labour.

Comprised in above are European and native labour, fuel, stores, spares, camp upkeep, passage money and leave pay, management expenses, depreciation of plant, insurance.

The aggregate cost of running a small dredge does not vary greatly from running a larger one and ground that is too low in value to give a return on a small dredge could be made to pay with a dredge of large output. With favourable conditions, dredges on easy ground have been run for as low as 1·5*d.* per cu. yd. treated. Large tin dredges in Malay 3·13*d.* to 4·0*d.* per cu. yd. treated.

TIN DREDGES.

Bucket Cap.	Type.	Monthly Output in cu. yds.	Working Hours per Month.	Dredging Depth.	Pontoon.	Steam. H.P.	Net Weight.
Cu. Ft.				Ft.	Ft. Ft.		Tons.
10	Open-con.	57,350	592	50	150 × 34	250	500
11	,,	89,000	550	50	150 × 40	320	700
12	,,	85,000	560	60	165 × 42	375	750
13	,,	78,000	—	60	165 × 45	380	820
15	,,	99,000	600	35	140 × 46	*400	—
5½	Close-con.	89,000	651	45	150 × 45	320	900
6	,,	Est. 90,000	600	50	155 × 45	375	—
8	,,	,, 120,000	,,	60	160 × 46	400	1,200
9	,,	,, 130,000	,,	60	170 × 52	450	1,300
12	,,	,, 150,000	,,	60	180 × 52	500	1,600

* Gas engines.

The difference in output is due to difficulties in working the ground.

MALAY TIN BASIS.

1 Kati = 1⅓ lbs. 100 Katis = 1 Picul = 133⅓ lbs.
16·8 Piculs = 1 ton. Straits Dollar = 2*s.* 4*d.*
Value of tin in ground varies from 0·33 to 2·0 Kati.
Native labour costs 1*s* 9*d.* to 2*s.* 3*d.* per day. Engineers, 4*s.* 6*d.* to 5*s.* per day.
Native coal, 24*s.* per ton. Wood, 25*s.*
Fuel oil, 90*s.* per ton. In bulk, about 82*s.*

See 'Suction-Cutter Dredging for Tin in Malaya,' G. A. More. Some Notes on Malayan Bucket Dredges,' E. J. Vallentine, *Bull. Inst. M. M.*, Dec., 1929.

NEW ZEALAND TYPE DREDGERS (GOLD).

Size of Buckets.	Estimated Capacity. In cu. yds. Monthly.	Digging Depth below Water-Line	Weight including Hull.	Price, f.o.b. London	Horse Power.
Cu. Ft.		Ft.	Tons.	£	
3	30,000	20	100	5,000	35
4	40,000	20	180	9,000	—
5	50,000	26	220	13,000	56
6	60,000	40	280	16,000	—
7	74,000	32	350	20,000	180

The Californian or American dredge is costly, very heavily built, with close-connected buckets digs with spuds, and till lately built with wooden hulls.

For properties where the transport is high, as in parts of Siberia, and where heavy repairs are difficult, a well-built New Zealand type of dredge, with ample stock of spares, is preferable

Indeed, its choice is imperative in some cases, for the high transport costs to some properties exclude the use of the ponderous American type of dredge. On the other hand, cheap transport, access to machine shops, and with heavy or stiff-cemented gravel, or large deep deposit of fair digging gravel (the latter permitting the handling of enormous daily yardages) are conditions favourable to the use of the American dredge.

CALIFORNIAN TYPE DREDGERS (GOLD)

Size of Buckets.	Estimated Capacity per month. In cu. yds.	Digging Depth below Water-Line.	Weight exclusive of Hull.	Price Erected.	Horse Power.
Cu. Ft.		Ft.	Tons.	£	
3	45,000– 60,000	36	350	22,000	180
5	90,000–110,000	36	520	30,000	220
7	110,000–150,000	35	576	34,000	300
8½	150,000–185,000	48	1,000	65,000	560*
13	250,000–280,000	40	1,100	90,000	415

* This includes 100 h.p. for pump for two monitors or giants.

See also Descriptive Section XXXIX, Part I.

Mavor & Coulson, Ltd.

SECTION XXXIX

PART II

MINERAL VALUATION.

(Revised and brought up to date by Sir Richard A. S. Redmayne, K.C.B., M.Sc., M.I.C.E., M.I.M.M., F.G.S., etc)

In all mine valuations for sale, probate or other purposes, it is necessary to ascertain the quantity and value of the product being mined, or to be mined. In the case of metalliferous mines especially, the determination of the average ore content is of the highest importance both in respect of the visible and the prospective resources, and demands the exercise of the highest qualities of expert knowledge and experience on the part of the valuer, for the prospective value depends on the probable extension of the mineral deposit in depth and laterally, in regard to which such questions as the origin and structural character of the deposit, secondary enrichment, development in neighbouring mines and depth of exhaustion enter largely into consideration.

Before the ascertainment of the value of a metalliferous mine can be made, a complete and detailed sampling of the deposit being worked must be carried out. The process is an arduous and expensive undertaking, and, in the case of an extensive mine, may well absorb many weeks before completion.

In the taking of samples the interval to be allowed between samples is of importance. If the ore is fairly regular in width and value an interval of 10 ft. will usually suffice. If spotty in value and subject to sudden changes, a lesser interval must be allowed, even as low as 3 ft. in some cases. If the vein is built up of common sulphides, such as pyrite or galene, variations are apt to be small and 20 ft. would not be an unsafe distance to take. The sample is obtained by cutting a groove across the entire width of the lode. As a sampling plan must be made it is necessary to measure as regularly as possible along the workings and mark each sampling site with coloured chalk. A sketch of each section of the working in which the sample is taken should be made, showing thereon variations in the vein (lode), the number of each sample and its position, and any other helpful information. The correct labelling of samples and the sacking and sealing of the samples, and the careful removal of them to a safe place are matters of importance.

An accurate sample represents a true cross section of the ore body. In the calculations, not the arithmetical but the geometrical mean must be arrived at. To give an example (Rickard):—

Width in Ft.	Assay Ozs. of Gold Per Ton.
4 ft. 4 in.	2·35
6 ft. 2 in.	0·45
7 ft. 5 in.	0·62
4 ft. 9 in.	0·85
2 ft. 5 in.	1·02
3 ft. 6 in.	2·40
2 ft. 5 in.	4·25
0 ft. 8 in.	5·20
1 ft. 2 in.	4·65
2 ft. 2 in.	3·21
35 ft. 0 in.	25·00

Now the arithmetical mean yields 3½ ft. of ore averaging 2½ ozs. of gold per ton, which is the wrong method of ascertainment. The correct figure is arrived at by taking the sum of the widths in each case multiplied by the ozs. per ton per ft. width and dividing this total by 35, which gives the geometrical mean of 1·71 ozs. per ton.

In carrying out the estimate of values, the term 'ore in sight' is frequently used when alluding to the ore reserves, but a better definition is that of Mr. Philip Argall, viz.:

'*Ore developed*'—which is ore absolutely without variation, ore exposed on all sides.

'*Ore being developed.*'—First class blocks with one side hidden; second class with two sides hidden; third class blocks with three sides hidden.

'*Ore expectant.*'—The prospective value of a mine, beyond or below the last visible ore, based on the fullest possible data (clearly set forth) from the mine being examined, and from the characteristics of the mining district.

In the case of coal mines and most stratified mineral deposits, owing to the more or less uniformity of the composition of the material mined, the question of sampling is of minor importance and may not even arise at all.

In considering the value of a mine as between a willing seller and a willing purchaser, or for the purposes of probate, having to estimate the amount of the annual profits which may be derivable from the mine and the period over which they will be forthcoming, one has to determine the present value of the annuities so arrived at. For which purpose various formulae are in use. In 1940 there was published by the *Iron and Coal Trades Review*, 'Formulae for use in connection with the Valuation of Mineral Properties,' for use by the valuers engaged in the work of valuing the coal royalties of Great Britain for acquisition by the State. The work of Mr. H. W. Naish, M.B.E., A.C.A., of the Coal Commission.

In Mr. Naish's valuable brochure no less than 12 formulae come under consideration, but the present writer has found the first five sufficient to meet most of his needs, namely that which deals with—

(I) Immediate Annuities.

(II) Deferred Annuities.

(2) Where a single period of deferment precedes a single uniform period of enjoyment and the remunerative rate is the same for both periods.

(3) Where a single period of deferment precedes a single uniform period of enjoyment but the remunerative rate in each period is different.

(4) Where there are varying periods of deferment and enjoyment and the remuneration rate is the same throughout.

(5) Where there are varying periods of deferment and enjoyment but the remunerative rates are different.

In all these formulae two rates of interest are employed, namely the 'accumulative,' or low, rate of interest for the recovery of the investor's capital, and the 'remunerative,' 'risk,' or high rate of interest.

Most of these formulae are based upon the same reasoning as those adopted by the present writer in 'The Ownership and Valuation of Mineral Property in the United Kingdom' (Longmans Green & Co., 1920), and used by him in his various valuations of mineral property. He has also made use of Hoskold's Tables, except in the case of deferred annuities with two rates of interest, in which case he has worked out what he believes to be the correct figures for himself.

In addition to the two works alluded to above, the interested reader is referred to Professor H. Louis' work, 'Mineral Valuation' (Charles Griffin, Ltd., 1920), 'The Theory of Finance,' by George King (C. & E. Layton, 1898), the last-named work containing a very thorough exposition of the doctrine of annuities, 'Principles of Mining Valuation, Organization and Administration,' by H. C. Hoover (London: Hill Publishing Co., Ltd., 1909), and 'Modern Mine Valuation,' by M. Howard Barnham (Chas. Griffin & Co., Ltd., 1912).

THE FORMULAE.

The 'present value' of the annuities (and in the case of a current going mine, plus the present value of the sum the machinery and plant will ultimately realise, or may be put at) is the value of the mine. The 'Purchase price' is, therefore, the sum paid down, and in order that the bargain may be an equitable one, the sum paid down should be such that the purchaser on the exhaustion of the mine or on the expiration of the term for which the lease thereof has to run, shall receive back his original outlay, together with interest per cent. agreed upon.

The formulae which the present writer has always used in valuing mining property are those in which two rates of interest are involved, namely, the interest on the purchase-money together with a return per annum on a part of the purchase money. The *rent* of the annuity in these circumstances consists of two portions—the *remunerative* or high rate of interest (sometimes termed the 'risk rate') and the *accumulative* or low rate of interest—for we create, as it were, a sinking fund to be set aside and separately invested and accumulated at a low rate of interest so as to liquidate the debt or capital and extinguish it suddenly at the end of the period; or we may regard each portion of capital in the successive payments of the annuity to be at once applied towards liquidating the debt, which will then gradually decrease until it finally vanishes, in which case as the debt is being paid off, a less and less proportion of the annuity will be required for interest, and a greater and greater proportion will be available to refund the capital. The two ways of regarding the transaction are the same. The sinking fund may be invested in securities until it amounts to the debt, or it may be invested in the security of the debt itself.

The sinking fund is therefore of the nature of an annuity which must accumulate at interest until it amounts to the original capital.

When it comes to considering the problem of a *deferred* annuity, under conditions in which two rates of interest are involved, authorities are at variance: thus the method adopted by some eminent valuers in the past was to find the value of an immediate annuity at two rates, and discount the value so obtained during the period of deferment at the higher rate, whereas it is not according to the conditions of the problem to use one rate only during the period of deferment.

For the clear mathematical reasoning by which Mr. King arrives at his formula, which is widely accepted as the correct one to use, readers are referred to pp. 38 and 39 of his excellent treatise referred to above. But the doctrine on which his formula is based may briefly be stated thus:

(1) A speculator is throughout the period of waiting or deference, only entitled to interest at a high risk rate on his original invested capital.

(2) During deference the latent interest (or hypothetical dividend) should accumulate at a low rate of compound interest.

The income during a period of enjoyment comprises:

(3) A high risk rate of interest on the original capital only

(4) A low rate of interest upon the accumulation;

(5) The exact extra annual sum required a sinking fund (put out at a low rate of interest) to replace the capital and accumulation at the figure at which it stood at the end of the deference immediately before entering on the annuity.

Mineral properties which fall to be valued may be divided into three different classes, viz.:

(1) Terminal annuities with immediate entrance.

(2) Terminal annuities with entrance deferred.

(3) Lump sums due after a period of deferment.

Valuations of all these rights may be governed by Mr. King's stipulations. Various formulae for the solution of the problems have been invented, some of which are very complicated. Those devised by Mr. Naish are, in my opinion, the simplest and are reproduced below. But for his examples and proofs the reader is referred to his book, in which are given a series of tables also.

I. Immediate Annuities.

$$\text{YP} = \frac{a}{\dfrac{20}{20-x} + \dfrac{a\text{R}}{100}}$$

where YP = Years purchase.

a = The amount to which £1 per annum accumulates during the period of enjoyment of the annuity at r per cent.

R = The remuneration (or high or risk) rate of interest.

r = The low or 'gilt-edged' rate of interest (*i.e.* after deduction of income tax).

x = The rate of income tax in shillings in the £.

II. Deferred Annuities.

(*a*) Where a single period of deferment precedes a single uniform period of enjoyment and the remunerative rate of interest is the same for both periods.

$$\text{YP} = \frac{a}{\dfrac{20}{20-x} + \dfrac{\text{AR}}{100}}$$

here A = The amount to which £1 per annum accumulates during the period covered by the whole transaction (*i.e.* deferment + enjoyment) at r per cent.

(*b*) Where a single period of deferment proceded a single uniform period of enjoyment but the remunerative rate in each period is different.

$$\frac{a}{\dfrac{20}{20-x} + \dfrac{a\text{R}}{100} + \dfrac{bc\text{P}}{100}}$$

here b = The amount to which £1 per annum accumulates during the period of deferment at r per cent.

c = The amount to which £1 accumulates during the period of enjoyment of the annuity at r per cent.

P = The remunerative rate of interest during the period of deferment.

(*c*) Where there are varying periods of deferment and enjoyment and the remunerative rate is the same throughout.

$$\frac{az}{\dfrac{20}{20-x}+\dfrac{AR}{100}}$$

Here the numerator = the sum of the results obtained by multiplying each stage of income by aZ.

d = The period of deferment before a given stage of income.
e = The period of enjoyment of a given stage of income.
t = The period covered by the whole transaction.
z = The amount to which £1 accumulates at r per cent. in $t-(d+e)$ years.

(*d*) Where there are varying periods of deference and enjoyment but the remunerative rates are different.

$$\frac{az}{\dfrac{20}{20-x}+\dfrac{a'R}{100}Z'}$$

here a' = The amount to which £1 per annum accumulates in e' years at r per cent.
d' = The period of deferment before a given remunerative rate rules.
e' = The period during which a given remunerative rate rules.
z' = The amount to which £1 accumulates at r per cent. in $t-(d'+e')$ years.

As this formula has reference to a somewhat involved situation, the following example of a valuation of a coal property may be given in exemplification.

Example.

In this case the coal area, which will not be taken over until the commencement of the 15th year (period of deferment is therefore 15 years), the ' certain ' or ' dead ' rent ruled as follows :—

at the end of the 15th year is £500
" " " " " 16th " " £650
" " " " " 17th " " £750
" " " " " 18th " " £900
" " " " " 19th " " £1,500

and by way of royalty rent—

at the end of the 20th year is £2,500.
and thereafter for 80 years is £3,666 per annum.

The period of revenue of certain rent is 5 years, and 15 + 5 = 20.

The period of revenue from royalty (in which the certain rent merges) is 80, and 80 + 20 = 100 the total period of the transaction. Then the values of z and z' are respectively as follows :—

$$\begin{aligned}
&\text{Value of } z = t-(d+e)\\
&\text{and of } z' = t-(d'+e')\\
(1)\ & 100-(15+1) = 84 = z\\
& 100-(15+5) = 80 = z'\\
(2)\ & 100-(16+1) = 83 = z\\
& 100-(15+5) = 80 = z'\\
(3)\ & 100-(17+1) = 82 = z\\
& 100-(15+5) = 80 = z'\\
(4)\ & 100-(18+1) = 81 = z\\
& 100-(15+5) = 80 = z'\\
(5)\ & 100-(19+1) = 80 = z\\
& 100-(15+5) = 80 = z'\\
(6)\ & 100-(18+1) = 80 = z\\
& 100-(20+80) = 0 = z'\\
(7)\ & 100-(20+79) = 1 = z\\
& 100-(20+80) = 0 = z'
\end{aligned}$$

Now, owing to the rent derivable from the property during the first 5 years being in the nature of a ' certain ' rent, we may take the remunerative or risk rate at 8 per cent., and for the remaining period of 80 years duration, at 10 per cent. That is to say, the 8 per cent. will rule over the period of 15 + 5 = 20 years, and the 10 per cent. over the period 20 + 80 years = 100 years.

ANNUITIES TO BE VALUED.

£500	per annum for	1 year	deferred	15 years	at 8 per cent.
£650	" " "	1 "	"	16 "	
£750	" " "	1 "	"	17 "	
£900	" " "	1 "	"	18 "	
£1,500	" " "	1 "	"	19 "	
£2,500	" " "	1 "	"	20 "	at 10 per cent.
£3,666	" " "	79 years	"	21 "	

Valuation by the formula 5.

$$\frac{\left\{\begin{array}{l}(500 \times 1 \times 11{\cdot}976) + (650 \times 1 \times 11{\cdot}627) + (750 \times 1 \times 11{\cdot}288) + (900 \times 1 \times 1095) + \\ (1500 \times 1 \times 10{\cdot}96) + (1500 \times 1 \times 10{\cdot}640) + (2500 \times 1 \times 10{\cdot}330) + (3666 \times 0 \times 311{\cdot}03)\end{array}\right\}}{1.333 + \frac{26{\cdot}87 \times 8}{100} \times 10{\cdot}64 + \frac{321{\cdot}36 \times 10 \times 1}{100}}$$

$$= \frac{1213896153}{55{\cdot}502} = £21{,}871$$

Deducting for—

Mineral rights duty .	. 3·75 per cent.	
Royalty welfare levy	. 3·75 ,, ,,	
Management costs .	. 5·00 ,, ,,	
	12·50 ,,	2,734
Net valuation		£19,137

If the annuities valued as above had been those derivable by way of estimated profits from the working of a colliery instead of by way of rental, the process followed in the valuation would have been the same except that the risk rates one would have taken would have been higher, being determined according to the extent of the risk attending the exploitation of the mineral deposit and the disposal of the product mined. In all cases, however, the amount of the accumulative or 'gilt edged' interest would be the same. In the example above this is taken at 3 per cent.

In the valuation of the colliery as a current going concern, inasmuch as such plant and machinery, as would be the property of the lessee and not the lessor (in present circumstances the lessor of all coal mines in the United Kingdom is the State), would not be disposable until the termination of the lease, the present value would be a lump sum representing the existing value of the same, discounted at say 5 per cent. per annum, for the unexpired term of the lease. Consumable stocks (*e.g.* timber, oil, grease, canvas, etc.), are covered by the working cost and would not be subject to valuation.

Mr. Naish in his valuable treatise recites certain cases requiring special treatment, for the manner of the elucidation of which the reader is referred to the work in question, but the formulae which have been reproduced above meet the majority of the cases with which the valuer of mineral properties will have to deal.

SECTION XL

EXPLOSIVES

Contributed by Arthur Marshall, A.C.G.I., F.I.C., etc., author of 'Explosives, their History, Manufacture, Properties and Tests,' 3 vols., 'A Short Account of Explosives,' 'A Dictionary of Explosives')

(Pp. 949–953)

SECTION XL

EXPLOSIVES

(Contributed by Arthur Marshall, A.C.G.I., F.I.C., etc., author of 'Explosives, their History, Manufacture, Properties and Tests,' 3 vols., 'A Short Account of Explosives,' 'A Dictionary of Explosives.')

Blasting Explosives.

These are the explosives with which the engineer is most concerned. There are a number of varieties available, and they differ from one another in physical properties, in power and in violence. As regards physical properties some are powders, more or less coarse, like gunpowder, and some are plastic, like blasting gelatine, but many are intermediate between these. Sometimes they are made into hard cylindrical pellets, but more often they are in the form of cartridges wrapped in paper. Many of them contain hygroscopic salts, such as ammonium nitrate or sodium nitrate, and for these the paper is made damp-proof.

The power of an explosive may be measured by firing a cartridge of known weight in a lead block and measuring the enlargement of the bore-hole (Trauzl test). Another method is to fire a charge of the explosive out of a gun at a ballistic pendulum, suspended immediately in front of the muzzle and measuring the swing imparted to the pendulum. These tests must, of course, be carried out under standard conditions if the results are to be comparable with those obtained by other experimenters.

The violence of an explosive depends upon the rate at which it detonates. A column of explosive when detonated at one end develops a wave of detonation which travels with a velocity of some thousands of metres per second. This varies somewhat with the diameter of the column and the degree of confinement and other conditions, but is a fairly definite quality. It increases as a rule with the density of the explosive, but some explosives, if compressed too much, become difficult to explode, and so there may be only partial explosion or a total missfire. The properties of a few typical explosives are given at the end of this Section.

Power is always a desirable quality in an explosive, for the more powerful it is the less is one obliged to use. The violence, however, should be appropriate to the use to which the explosive is to be put. If the velocity of detonation be very high the explosive has a shattering effect, therefore when it is desired not to break up the rock or other material too much, it is advisable to use a comparatively mild explosive.

Black Powder.

Black powder or gunpowder is a mixture of saltpetre, charcoal and sulphur, in about the proportions 75 : 15 : 10. It is much less violent than the more modern blasting explosives. Considerable quantities are still used, especially in quarries and open workings, but the amount is falling. It has less power than modern explosives and is also objectionable because it generates a large amount of smoke and poisonous fumes. With gunpowder, even more than with high explosives, it is essential that the charge be well confined by tamping the bore-hole with clay or sand, otherwise it will produce little effect. It is best applied in 'sprung' bore-holes, that is to say the bottom of the hole is enlarged into a chamber by firing one or more small preliminary charges before the final charge of gunpowder is inserted. Black powder may be used advantageously in obtaining block stone and monumental stone, also in blasting limestone for kilning, because high explosives develop shatter cracks in it with the result that the kiln becomes choked with powder.

Black powder made with sodium nitrate (Chile saltpetre) instead of potassium nitrate is cheaper and more powerful, but is even slower and takes up moisture from the air more. It should not be used in rock that is much fissured.

Dynamites.

Until the latter part of the nineteenth century gunpowder was practically the only explosive manufactured or used for all purposes; as a propellant in firearms, for blasting and as a filling for shell. Then Alfred Nobel succeeded in making comparatively safe explosives, consisting

largely of nitroglycerine. These are generally called dynamites, but the jelly-like material made by mixing the liquid nitroglycerine with about 8 per cent. of collodion cotton (a variety of nitro-cellulose) is known as blasting gelatine. Gelignite, which is a variety of explosive much used in Great Britain, consists of 50 to 63 per cent. of nitroglycerine, thickened with collodion cotton and mixed with potassium or sodium nitrate and wood meal. Gelatine dynamite is a similar mixture but richer in nitroglycerine, of which it contains 70 to 77 per cent. Ammon-Gelignite contains a comparatively small proportion of nitroglycerine, but also has some ammonium nitrate. In Great Britain the term dynamite is generally applied to a mixture of about 75 per cent. of nitroglycerine absorbed in 25 per cent. of kieselguhr. Although this is not so powerful as the gelatinised explosives that have just been mentioned, it has the advantage that it develops its full violence when only slightly confined.

In the United States a range of dynamites is made containing different proportions of nitroglycerine from 5 or 10 per cent. upwards. The *straight dynamites* contain also wood pulp and sodium nitrate together with small quantities of other substances. For the weaker grades some of the wood pulp is often replaced partly by flour and sulphur. In the *gelatin dynamites* the nitroglycerine is gelatinised by the addition of a little collodion cotton. This prevents the liquid nitroglycerine exuding and consequently renders the explosives safer, especially in damp places. The *ammonia dynamites* contain a considerable proportion of ammonium nitrate. *Ammonia gelatins* contain ammonium nitrate and gelatinised nitroglycerine. The 'grade' of a straight dynamite represents the percentage of nitroglycerine that it contains; in the other varieties the grade is supposed to give the strength in terms of straight dynamite. Thus 40 per cent. straight dynamite contains about 40 per cent. of nitrogylcerine, and 40 per cent. gelatin dynamite or 40 per cent. ammonia dynamite is supposed to be of the same power, but actually is generally rather weaker.

Low Freezing Nitroglycerine.

A grave objection to explosives containing ordinary nitroglycerine is that the substance is liable to freeze to a crystalline solid which has a melting point of 13° C. (55° F.). When frozen they are less sensitive to detonation but more so to blows; in the semi-frozen state they are still more sensitive to blows and consequently very dangerous. In this condition they have caused many accidents. Frozen cartridges should be thawed before use by placing them in a special thaw-pan with a water jacket containing warm water. They should not be thawed by placing them near a fire; this may appear obvious, but many fatal accidents have been caused in this way.

By the addition of various substances to nitroglycerine the freezing point can be reduced so much that it will no longer become really hard even under severe winter conditions. In America nitrotoluene was at one time used extensively for this purpose, and in Germany nitrochlorohydrin, but now the tendency is to use dinitro-glycol. This is a substance very much like nitroglycerine and an equally powerful explosive, and it can be manufactured at a comparable price. All the nitroglycerine explosives made now in the United Kingdom are low-freezing, and a prefix, such as 'Polar' or 'Antifrost,' is added to the name to show the fact, as 'Polar Ammon Gelignite.' Fatal accidents due to freezing have consequently ceased.

Ammonium Nitrate Explosives.

Explosives of which the main constituent is ammonium nitrate are used more and more and they are comparatively safe to handle. Ammonium nitrate alone can be detonated under extreme conditions, as was shown by the appalling disaster at Oppau in Germany in 1921, but to secure an explosive that will not give rise to the danger of missfires it is necessary to add not only a proportion of combustible matter to combine with the excess of oxygen in the nitrate, but also a small quantity of some fairly sensitive explosive such as nitroglycerine or trinitrotoluene.

Coal Mine Explosives.

In all the principal countries that possess coal mines there are stringent regulations as to the explosives that may be used in them. The object of course is to prevent the use of those that have too much tendency to ignite mixtures of fire-damp and air or coal-dust and air. In England these explosives are tested in a special testing gallery at Buxton, where they are fired from a steel cannon into a large cylinder containing a mixture of methane and air, and also into a mixture of coal-dust and air. The most severe conditions that may occur in a coal mine are thus imitated. If no ignitions occur the explosive is placed on the 'Permitted List,' and it is called a 'Permitted Explosive.' There are similar testing stations in the United States, Germany, France and Belgium. In the United States the testing station is conducted by the Bureau of Mines under the Federal Government. As the regulation of the mines comes under the individual states the Bureau has no direct power and can only make recommendations, and the approved mixtures are consequently called 'Permissible Explosives.'

Nearly all the special explosives used in coal mines contain a considerable proportion of ammonium nitrate, which gives a comparatively cool flame with little tendency to cause ignitions. This tendency is further reduced by the addition of some 10 or 25 per cent. of a 'cooling agents,' such as sodium chloride or ammonium chloride or oxalate. This, of course, also diminishes the strength of the explosive.

Up to the end of 1931 there was a sort of modified gunpowder called 'Bobbinite' on the Permitted List, but this has now been discontinued. As it had only a low rate of explosion it gave a greater proportion of large coal. It was, however, considered to have a slightly greater tendency to ignite fire-damp than the other permitted explosives, and so, after a controversy lasting many years, it has been removed from the list. Millions of pounds of gunpowder are still used in coal mines, but only in comparatively safe localities.

Sheathed Cartridges.

The safety is increased considerably by enclosing each cartridge in a sheath containing some substance which tends to extinguish the flame. Various materials have been found effective, but as the result of investigations by the Safety in Mines Research Board a layer ⅛ in. thick of sodium bicarbonate round each cartridge is specially recommended, and such sheathed cartridges are being used extensively in Great Britain. It is found that the sheath causes no loss of power; it has a cushioning effect on the explosion and therefore increases the proportion of lump coal.

Low Density Explosives.

The proportion of lump coal is also increased by using explosives of very low density containing a bulky saw-dust or plant fibre instead of wood meal. The bulk density can thus be reduced as low as 0·7.

Chlorate Explosives.

There are some blasting explosives which contain neither nitroglycerine nor ammonium nitrate, but mostly they are of minor importance commercially. Many attempts have been made to produce satisfactory explosives containing chlorate or perchlorate as the oxygen carrier, but they suffer under two serious disadvantages: they are rather expensive and are more sensitive than other commercial explosives to blows and friction. Cheddite, one of the most successful, contains chlorate and a nitro-compound together with 5 per cent. of castor oil, which reduces the sensitiveness. In Germany a mixture of 90 parts potassium chlorate and 10 parts mineral oil is used in the potash mines and is called Miedziankit or Choratit 3.

Liquid Oxygen Explosives.

A powerful explosive is obtained by absorbing liquid oxygen in a porous combustible material, such as lamp black or wood meal. It can be fired directly by means of safety fuse, but is more reliable if an electric detonator be used. The materials are cheap, but there are a number of disadvantages: the cartridge must be fired before too much of the liquid oxygen has evaporated, and the explosive is rather sensitive. Consequently its use requires a higher degree of skill on the part of the shot-firer, and of course there must be a supply of liquid oxygen at hand. On the other hand, missfires are not dangerous because the oxygen all evaporates off, leaving only inert matter. On the Continent this explosive is used on a fairly large scale, especially in iron ore mines of Lorraine; in Great Britain, although attempts have been made to introduce it, the consumption has fallen heavily since 1929. They are really only suitable for use in porous rocks which allow the oxygen to escape as it evaporates.

Cardox.

Liquid carbon dioxide has also come into use for blasting purposes, especially for coal. The 'Cardox' cartridge consists of a hollow steel cylinder, 2 or 3 inches in diameter, containing a few pounds of liquid carbon dioxide and a heater composed of a mild but hot explosive, which can be fired electrically. When this is done the heater converts the liquid into gas, the pressure shears a steel closing disc in the end of the cylinder, the gas escapes and breaks down the coal. The cylinder is, of course, placed in a bore-hole and firmly secured in position.

Hydrox.

This is a somewhat similar device, and is also not strictly speaking an explosive. A mixture of sodium nitrite and ammonium chloride in a container of steel tubing is inserted into the bore hole and is fired by a powder fuse, whereupon it decomposes with the evolution of gas. Like Cardox it is finding increasing favour for bringing down coal.

Procedure in Blasting.

As regards the drilling of the bore-hole, see Section XL. (p. 886, *et seq.*). The hole is cleaned out carefully and the cartridges of explosive are inserted one after the other and pressed home with a wooden tamping rod. Into the last cartridge, that nearest the mouth of the hole, is inserted the detonator. If an electric detonator be used the two wire leads are arranged to

emerge from the hole. If the detonator be an ordinary one it is before insertion provided with a piece of safety fuse of the required length. One end of this is pushed into the copper detonator shell which is crimped on to the fuse. *Safety fuse* is a cord of jute or other fabric suitably waterproofed and provided with a fine core of gunpowder, which burns at the rate of 2 feet a minute.

The free portion of the bore-hole is now filled with a tamping of clay or sand, or preferably a 3 : 1 mixture of sand and clay, pressed gently but firmly home, and the shot is then ready for firing when everyone has retired to a safe distance.

The leads from electric detonators are connected by means of a considerable length of insulated wire with a source of current, which may be either a main or a hand-operated machine called an 'exploder.' A number of shots can be fired simultaneously with electric detonators.

Simultaneous shots can also be fired by means of a *detonating fuse*, otherwise called 'Cordeau,' which consists of a core of a detonating explosive in a suitable envelope. The most usual variety is composed of trinitrotoluene in a lead tube, and has a velocity of detonation of about 5,000 metres per second. Flexible cordeau is also made by enclosing a core of penthrite or fulminate in a waterproofed textile cover.

Demolitions.

For blowing up buildings and machinery the procedure is much the same as for obtaining minerals out of the earth, but in some cases it is not feasible to bore a hole to take the explosive. It must then be attached to the outside, and the most violent explosives should be used, such as dynamite, as these lose less of their effect under these conditions. The charge should be placed in intimate contact with the object to be blasted, and should be covered over with a good layer of clay to confine it as far as possible.

Steel plates can be cut in this way with a string of cartridges of Arctic Dynamite, and a ship can be divided in two. The same explosive is used for increasing the flow from oil wells and artesian water wells; a considerable charge is enclosed in an iron container, lowered to the bottom of the well and there fired. To break up heavy castings a hole is bored and a chamber made at the bottom by firing a small charge without stemming and then the main charge is fired with stemming. For this purpose ammon gelignite may be used; it is also recommended for breaking up brickwork. For submarine work and for deepening rivers the gelatinised nitroglycerine explosives are used, as they are less affected if a small amount of water gets through the waterproofing. The harder the rock the more nitroglycerine the explosive should contain. The manufacturer of the explosives will give advice and help in carrying out any of these operations, or similar ones.

For military demolitions ordinary commercial explosives are sometimes used, but as they are rather dangerous under the strenuous conditions of active service, special ones are issued. In the British Army blocks of wet compressed guncotton are used, fired by a primer of dry guncotton, which in turn is set off by a detonator fired either electrically or by safety fuse.

Shell Fillings, etc.

Shells can only be charged with very insensitive explosives, otherwise they would explode in the bore of the gun when it is fired. The two substances that are most used are trinitrotoluene (trotyl or T.N.T.) and picric acid (lyddite). It is usually mixed with a considerable proportion of ammonium nitrate: Amatol 80/20 is a mixture of 80 parts ammonium nitrate and 20 of T.N.T., and similarly Amatol 40/60 consists of 40 per cent. ammonium nitrate and 60 per cent. T.N.T. Picric acid is still used in the British Navy. It has the disadvantage that it forms dangerously sensitive picrates with lead, iron, and some other substances. Although it is insensitive enough to stand the shock of discharge, shells charged with it will not as a rule penetrate modern armour plate, but detonate before they have got through it.

Shrapnel shells are filled with bullets with a small charge of gunpowder to eject them before the shell reaches its objective.

A shell is caused to explode by means of a fuse fixed either into the nose or the base. There are two varieties: time and percussion fuses. Time fuses act at a determined time after the discharge of the gun, and are generally fitted to shrapnel shell; percussion fuses act when the shell strikes its object.

Grenades, aerial bombs, naval mines and torpedo warheads are filled with the same explosives as shells or similar ones.

Detonators.

Fulminate of mercury is no longer the only substance used for filling detonators, as lead azide has replaced it to a considerable extent. There are also composite detonators containing a main charge of T.N.T. or tetryl primed with fulminate or azide. In this country two sorts are made: a fulminate detonator loaded into a tube of drawn copper 6 mm. in diameter, and a tetryl-azide detonator in an aluminium tube. The latter may not be used in dangerous coal mines. Of the various sizes mentioned on p. 863, Nos. 6 and 7 are those generally used. In an electric detonator the electric fuse fits into the tube just above the explosive. Both high and low tension electric detonators are in general use, but the latter are preferable for multiple shot-firing.

Propellants.

Black powder is still used to a small extent in *shot-guns* on account of its cheapness and reliability. Its disadvantages are the smoke it gives off, the serious fouling of the barrel and the heavy recoil. The smokeless shot-gun powders consist essentially of nitrocellulose partially or entirely gelatinised by the use of a solvent, such as ether-alcohol, which is afterwards dried out. By suitable manipulation the powder is made into porous grains which are so bulky that a charge occupied about the same space in the cartridge case as the much greater weight of black powder which gives the same muzzle velocity.

The 'powders' for use in *rifled fire-arms* are not porous. They are made in the form of flakes, rods, cords, tubes or strips. The longer the barrel, the greater the time that the powder has to burn before the projectile leaves the muzzle, and consequently the thicker are the flakes, tubes, etc. There are two main classes of military powders: those consisting almost entirely of nitrocellulose, called 'nitrocellulose powders,' and those containing also a considerable proportion of nitroglycerine, called 'nitroglycerine powders.' The British service powder, cordite, is of the latter type. Most of the foreign powers use nitrocellulose powders, but the Germans have both.

Precautions.

Explosives are made to explode, and consequently should be treated with due respect. Every endeavour is made to prevent them from being unduly dangerous, but they can never be safe. Their use and storage is regulated by the rules drawn up by H.M. Chief Inspector of Explosives, which may be studied in the 'Guide to the Explosives Act,' published by the Stationery Office. An explosive must be stored in properly constructed magazines, for which a licence must be obtained. Detonators should not be kept in the same room with other explosives. Magazines must not be repaired until all the explosives have been removed. The transport of explosives is also subject to official regulations.

Explosives containing nitroglycerine and nitrocellulose are liable to deteriorate on keeping, especially in hot climates, and consequently should be inspected and tested from time to time.

When fired explosives almost always generate some carbon-monoxide gas, even though they contain sufficient oxygen for complete combustion. As this gas is poisonous underground workings should be well ventilated, and the site of an explosion should not be approached too soon.

Properties of Typical Explosives.

Explosive.	Density. (Water = 1.)	Power. c.c./10 g.	Velocity of Detonation. Metres/sec.
Blasting gelatine	1·55	520	5500
Gelatine dynamite	1·6	415	5000
Gelignite (60 per cent. nitroglycerine)	1·6	340	5000
Dynamite (kieselguhr)	1·24	335	5000
American dynamites—			
Straight, 40 per cent.	1·32	309	4880
" 60 "	1·30	365	5700
Ammon 40 "	1·28	275	3780
" 60 "	1·28	347	5700
Gelatin 40 "	1·56	254	2700–4900
" 60 "	1·50	333	2930–5610
Coal mine explosives (about)	1·1	230	2500
Black powder	1·04	187	280–400
Guncotton	1·1	375	6000
Trinitrotoluene (T.N.T.)	1·57	300	6950
Picric acid (cast)	1·63	300	7250
Amatol, 40/60	1·55	350	6470
" 80/20	1·4	380	4900

SECTION XLI

RADIOCOMMUNICATION

(Pp. 957–973.)

(Compiled by G. Parr, M.I.E.E.)

SECTION XLI

RADIOCOMMUNICATION.

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GENERAL.

The term Radiocommunication covers all methods of transmitting intelligence from point to point by means of electromagnetic radiation through space, and thus includes not only broadcasting of speech and music but signalling from ships and aircraft, transmission of identification or warning signals from beacons, direction-finding signals, and television. Receivers intended for long distance reception of Morse signals or speech are referred to as 'communication receivers' in distinction to the 'broadcast receivers' used for listening to programmes regularly transmitted by the broadcasting stations of the world.

The regulations under which radiocommunication is carried out are drawn up by an International Radio Convention but international broadcasting policy, such as the allocation of wavelengths, power of stations, and interference problems is determined by an International Broadcasting Union.

The wavelength bands at present allocated for broadcasting purposes are approximately :—

1,000–2,000 m.	300–150 kc/s	Long waves
800–200 m.	1,500–375 kc/s	Medium waves
50–13 m.	6–2·3 Mc/s	Short waves

Other wavebands have been allocated as follows :—

Broadcasting : 13, 16, 19, 24, 31, 41, 49 m.

Amateur radio : 144–145 Mc/s ; 1215–1300 Mc/s ;
5,650–5,850 Mc/s ; 10,000–10,500 Mc/s.

The definition of radiation in terms of its wavelength is being gradually displaced by the use of the frequency to denote the classification, and the term 'short-wave' is more accurately 'high-frequency.'

The classification of radio waves in terms of frequency is given by B.S.I. 204—1943, as :—

Very low frequency (audio frequency)	Below 30 kc/s
Low frequency	30–300 kc/s
Medium frequency	300–3,000 kc/s
High frequency	3,000–30,000 kc/s
Very high frequency	30,000–300,000 kc/s
Ultra-high frequency	300,000 kc/s–3,000 Mc/s
Super-frequency	3,000 Mc/s–30,000 Mc/s

The use of higher frequencies for radiocommunication has several advantages, chiefly in the greater transmitter efficiency obtainable and in the ability to direct the radiation within a narrow beam towards the desired reception point. A further advantage is that a greater number of stations, each having a frequency bandwidth of the full range of speech and music, can be accommodated within a given high frequency band without interference.

The disadvantages of the higher frequencies, particularly those corresponding to a wavelength below 10 m., are that the absorption along the earth's surface is considerable and unwanted reflection or refraction takes place. Communication on very high frequencies is usually limited to stations within horizon distance, although no definite rule can be established.

WAVELENGTH-FREQUENCY CONVERSION TABLE.

Formulæ: $\text{Wavelength in metres} = \frac{300,000}{\text{frequency in kc/s}}$.

$\text{Wavelength of oscillatory circuit} = 1,884\sqrt{LC}$

where L and C are in μH and μF respectively.

Frequency. kc/s.	Wavelength. m.	LC.	Frequency. kc/s.	Wavelength. m.	LC.
100	3,000	2·5330	550	545·45	0·0837
150	2,000	1·1258	600	500	0·0703
200	1,500	0·6332	650	461·54	0·0599
250	1,200	0·4054	700	428·57	0·0514
300	1,000	0·2814	750	400	0·0450
350	857·15	0·2067	800	375	0·0395
400	750	0·1583	850	352·94	0·0350
450	666·66	0·1250	900	333·33	0·0312
500	600	0·1013	1,000	300	0·0253

To extend this table for use with multiples or sub-multiples of the quantities, the following factors should be used:—

Range of Frequency.	Wavelength Factor.	LC Factor.
10–100 c/s	10,000	100,000,000
100–1,000 c/s	1,000	1,000,000
1–10 kc/s	100	10,000
10–100 kc/s	10	100
1–10 Mc/s	10^{-1}	10^{-2}
10–100 Mc/s	10^{-2}	10^{-4}
100–1,000 Mc/s	10^{-3}	10^{-6}

Example.—To find the wavelength corresponding to 15 Mc/s:

The wavelength in the table is given as 2,000 for 150 kc/s. For 15 Mc/s divide by 100, giving 20 m. as the wavelength. The LC product is 1·1258 × 100, or 112·58.

FREQUENCY MODULATION.

The usual method of transmitting speech or music in radiocommunication is by varying the amplitude of a radio frequency carrier wave, the amount of variation being proportional to the intensity of the sound. This method is termed *amplitude modulation.*

An alternative method, first suggested by Round in 1921 (1), and developed by Armstrong (2), is to maintain the amplitude of the carrier wave constant and vary its frequency in proportion to the frequency of the modulating signal. In this system, known as *frequency modulation*, two changes in the carrier wave take place: the frequency of the modulating signal is caused to vary the carrier at a corresponding frequency, and the amplitude of the modulating signal controls the frequency deviation of the carrier about a pre-determined value. For example, a 1,000 c/s note of low intensity will vary the carrier frequency of 1,000 kc/s from 1,000·1 to 999·9 at a rate of 1,000 c/s.

Four important advantages are given by frequency modulation over the more usual amplitude modulation: greater signal-to-noise ratio, lower transmitter power for a given output from the receiver, less distortion of the modulating voltage, and greater freedom from interference between adjacent stations. The disadvantages are that reception must be confined to ultra-short wave 'direct path' transmission if distortion of some of the frequency components and the distortion introduced by selective fading is to be avoided.

The improved signal-to-noise ratio, which is a feature of the system, is still further increased by the use of 'pre-emphasis'—the increase in amplification of the higher frequencies at the transmitter. This requires corresponding 'de-emphasis' to correct the tonal balance at the receiver.

The successive increases in signal-to-noise ratio given by various factors in the transmission can be tabled as follows:

Increase due to phase modulation . . .	2·9 times
„ due to wider modulation pass band .	25 „
„ by pre-emphasis and de-emphasis .	5·5 „
„ by more efficient transmission . .	2 „

giving an overall improvement of 800 times.

Phase-modulation.

In the method of transmission known as *phase-modulation* the amplitude of the carrier remains constant and its phase angle with regard to its unmodulated condition is varied in accordance with the frequency of the modulating signal. The magnitude of the phase change is determined by the intensity of the signal.

The mathematical expression for a frequency-modulated wave (assuming sinusoidal form) is

$$E \sin(2\pi f_c t - M \cos 2\pi f_m t)$$

where f_c is the carrier frequency, f_m the modulating frequency, and M is the ratio of frequency deviation to the modulating frequency, termed the Modulation Index.

For phase-modulation, the expression becomes:

$$E \sin(2\pi f_c t + \phi M \sin 2\pi f_m t)$$

the phase angle covered being ϕM, where ϕ is a constant and M is proportional to the amplitude of f_m.

Receivers for F.M.

A diagrammatic layout for a frequency-modulation receiver is shown in Fig. 1. The circuit differs from an amplitude-modulated receiver in the inclusion of a 'limiter' and frequency-amplitude conversion stages. The dipole aerial is connected to one or more radio frequency

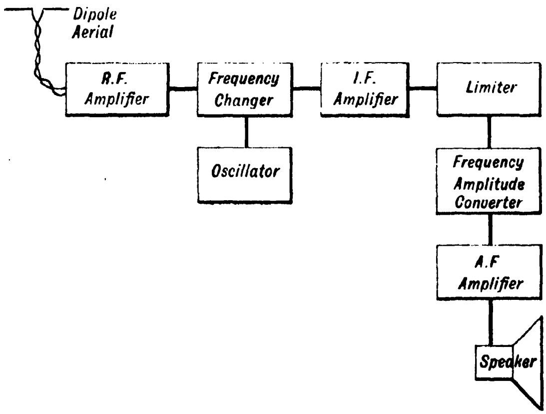

FIG. 1.—Schematic diagram of receiver for frequency-modulated signals.

amplifying stages, followed by a frequency changer, as in the conventional superheterodyne. The output from the intermediate frequency amplifiers is fed to the limiter, which has the function of reducing any amplitude modulation of the carrier due to noise, interference, or variation in the frequency response of the pass band. A common form of limiter is a saturated amplifier stage, whose amplification factor is inversely proportional to the amplitude of the input signal.

A common form of frequency-amplitude converter employs a frequency discriminator similar to that used for automatic frequency correction in superheterodyne receivers. This translates the I.F. carrier frequency into a D.C. bias voltage which is zero at the correct carrier setting and becomes increasingly positive or negative for frequencies above or below this value. A modulated

carrier wave applied to such a circuit will therefore give an output voltage which will vary with the A.C. component of the carrier wave. The audio frequency amplifier which follows the frequency converter is on conventional lines but is designed to take advantage of the wider audio frequency range of the transmitted signal.

Experimental Transmissions.

A large number of experimental frequency-modulated transmitters have been in operation in America for over three years, and the service is also being established as an adjunct to the ultra-short wave television transmissions. In this country an experimental station at Wrotham is under construction and will be operating in 1949. The wavelength is 3·3 metres and the power has been provisionally stated to be 25 kW.

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1. H. J. Round, *Radio Review*, vol. ii., p. 220 (1921).
2. E. H. Armstrong, *Proc. I.R.E.*, p. 689, May 1936.
3. B. van der Pol, *Proc. I.R.E.*, p. 1194, July 1930.
4. G. W. O. Howe, *Wireless Engineer*, p. 547, November 1939. See also ' Frequency Modulation.' Monograph, by K. R. Sturley (Morgan Bros. (Publishers), Ltd.).

BRITISH TELEVISION.

The present television service was inaugurated in 1937 and, according to official statement, will remain as standard for many years. The service area of the London station (Alexandra Palace) within which good reception can generally be relied on is bounded by a circle of approximately 25 m. radius, passing through Tring, Slough, Woking, Redhill, Sevenoaks, Tilbury, Pitsea, Waltham, and Hitchin. Freak reception has been reported from widely scattered areas, including Jersey, Isle of Wight, Bristol, the Midlands, and the Continent.

Details of the System.

The vision signal is transmitted on a frequency of 45 Mc/s and the accompanying sound on a frequency of 41·5 Mc/s. The band width of the vision transmitter is 3 Mc/s, and the frequency response of the sound transmitter is flat from 30 to 10,000 c/s. The radiation is vertically polarised. The peak output of the vision transmitter (' full white '—see on) is 17 kW, and that of the sound transmitter 12 kW.

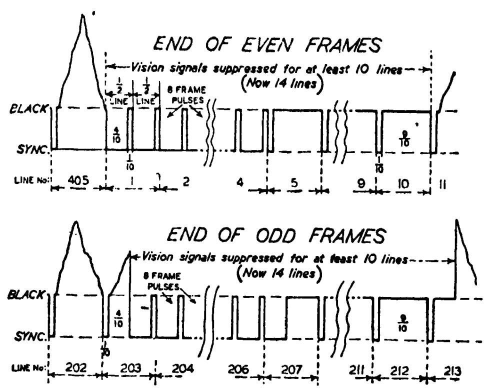

FIG. 2.—Waveform of vision signal from the London station, showing difference between odd and even frames.

A characteristic of the British system is that a ' positive ' signal is radiated, *i.e.* the amplitude of the carrier wave is proportional to the picture brightness at any instant, and maximum carrier corresponds to full white tone in the picture.

In order to provide for a synchronising signal which can be readily separated from the vision signal, the level of the carrier wave corresponding to black in the picture is set at 30 per cent. of maximum amplitude. At the end of each line signal transmitted the carrier amplitude falls below this datum (see Fig. 2), giving a synchronising pulse which can be readily separated from the vision

signal by a form of amplitude filter circuit in the receiver. Such a pulse also serves to 'black out' the beam of the cathode ray tube so that its return path on the screen at the end of each line and frame is not visible.

Interlacing.

The picture is scanned with 405 lines, 25 times per second. In order to minimise flicker a repetition rate of 50 per second is desirable, and this can be achieved in effect without increase in the band width of the signal by the system known as interlacing.* The picture is scanned with half the total number of lines ($202\frac{1}{2}$) in $\frac{1}{50}$th sec. to form one 'frame,' a gap being left between each line equal to the thickness of a line. At the completion of this scan the spot returns to the beginning of its travel and traces a second set of $202\frac{1}{2}$ lines in the spaces left between the first set. The eye thus receives an impression of a complete picture repeated 50 times per second, as it does not appreciate the interval between the scanning of adjacent lines.

The diagram of Fig. 2 shows the waveform of the vision signal at the end of the even and odd frames, the displacement of the spot by half a line being accomplished by a variation in the synchronising impulses.

Programmes.

Programmes are transmitted every day from 3 to 4 P.M. and 8.30 to 10.0 P.M. A test film is also transmitted each morning from 11 A.M. to noon for the setting-up and demonstration of receivers.

The programmes include variety, concerts, full-length plays from the studio and outside theatres, illustrated talks and interviews, and films adapted for television. The B.B.C. has its own newsreel unit.

Outside events are covered by two mobile transmitting units which can cover distances up to 30 m. from Alexandra Palace. In addition, a special cable has been laid in London with tapping points at suitable intervals to enable direct transmission of events of national importance from the mobile unit to the transmitting station.

Miscellaneous Data.

The staff of the television station numbers about 300 on the programme side and 200 engineers and technicians. The Head of the Television Service is Norman Collins, and the Programme Director is Cecil McGivern. D. C. Birkinshaw, M.B.E., is the Superintendent Engineer, and the Studio Engineer-in-Charge is H. Baker.

The total number of television licences held at the end of 1948 was approximately 75,000, the cost of a combined radio reception and television licence being £2. The production of television receivers amounted to a monthly rate of 6,430 in 1948, compared with 2,300 in 1947. It is expected that a total of 200,000 receivers will be issued by the end of 1949.

Extension of Service.

By the end of 1949 it is expected that a television station will be in operation at Sutton Coldfield, Birmingham, to serve the Midlands area. This station will be connected to the London television station by means of a series of relay links at Harrow Weald, Dunstable, Blackdown Hill, and Rowley Regis, through which the programmes will be transmitted on ultra-short waves, and also by coaxial cable.

The transmitter at Sutton Coldfield will operate on a frequency of 61·75 Mc/s for vision and 58·25 Mc/s for sound. The power of the vision transmitter will be 35 kW., which should provide adequate signal strength over a radius of 50 m. from the transmitter.

It will also be possible to transmit programmes in both directions, and the relative merits of the ultra-short wave link and the cable will be studied to determine the most economical method of relaying programmes to other parts of the country. The mast at Sutton Coldfield will be 800 ft. in height, which is the maximum so far attained by the B.B.C. service.

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'The Marconi—E.M.I. Television System,' by Blumlein, Browne *et al.* *J.I.E.E.*, vol. lxxxiii., No. 504 (1938).

'The London Television Service,' by Macnamara and Birkinshaw, *J.I.E.E.*, *ibid.*

RADIO VALVES.

Base Connexions.

Since the war the use of radio valves fitted with an 'octal' base has become almost universal, and indirectly-heated valves operating from 4 volt A.C. are being replaced by 6·3 volt valves.

* Ballard, Brit. Pat. 420391 (R.C.A.).

The octal based valves originated in America and bear American type numbers, but equivalents are obtainable from the majority of British manufacturers.

A similar base to the octal base is fitted to Mazda valves, but the dimensions and the connexions to the pins differ. The points of difference are shown in Fig. 3 and in Table below. Care should be taken not to force an American octal base into a Mazda octal socket. The table gives the convention for the pin connexions in each type of base.

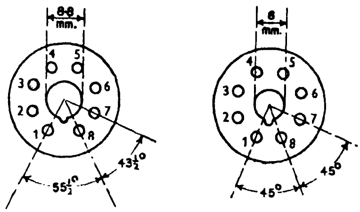

FIG. 3.—Illustrating the principal points of difference between the Mazda Octal-type base (left) and the American Octal Base. Note the spacing of the pins.

CONNEXIONS TO PINS OF VALVE BASES.

Pin No.	American Octal Base.	Mazda Type Octal Base.
1	In metal valves connected to the shell, in glass bulb valves, usually blank, or to internal shielding	Always filament or heater.
2	Always heater or filament . .	Cathode of indirectly heated (I.H.) valves, except in some diodes.
3	Varies, but usually anode . .	Anode or diode anode except in ½-wave A.C./D.C. rectifier when it is the cathode.
4	Varies	Varies. Always screen in pentodes.
5	Varies	Varies.
6	Usually a grid when present. Often omitted	Usually metallising, but a control grid in double triodes and pentodes.
7	Always heater or filament . .	Varies.
8	Usually cathode	Always heater or filament.
Top cap	Usually control grid . . .	Always control grid, but G_4 in octodes.

Pins are numbered clockwise with No. 1 to the left of the key when the base is held towards the viewer, key downwards.

In multi-grid valves, the grids are designated G_1, G_2, G_3, etc., in order outwards from the cathode.

Miniature Valves.

Developed during the war for use in portable and lightweight apparatus, miniature valves with all-glass bases are now fitted to many types of receivers. Typical dimensions of an all-glass based valve are 2 ins. overall length and ¾ in. diameter. The British type reference numbers are B.7.G and B.8.A., indicating the type of base used.

The pins in the base are of small diameter and comparatively fragile in comparison with the more robust split pin in the moulded base, and special precautions must be taken in fitting the valves into their sockets. Valveholders should be fitted with ' floating ' contacts to avoid bending the pins on insertion, and it is desirable to use a form of jig inserted into the socket when soldering the connecting wires in place.

Code of Practice.

The British Radio Valve Manufacturers' Association* have issued a booklet giving guidance to the designers of equipment employing electronic valves so that optimum performance and life may be secured. Some of the more important points are summarised below :—

1. The heater voltage should not vary more than 7 per cent. from the rated value. In certain rectifiers this permissible variation is much less.

2. Heaters of mains-operated valves should not be connected in series unless specifically designed for the purpose.

3. Valves should normally be operated base downwards, but if mounted horizontally, the plane of the filament should be vertical.

4. Mercury vapour rectifiers should never be operated in any other position than the vertical.

5. The potential difference between heater and cathode should not exceed 150 volt, except where specified as permissible.

6. The published ratings should be closely observed. The first maximum rating reached should be the limiting factor in the performance.

7. The layout should give adequate ventilation to ensure safe bulb temperature. This particularly applies to 'miniature' valves and valves in screening cans.

8. Valves should never be operated without a D.C. connexion between each electrode and the cathode. The resistance between grid and cathode may be $3 \cdot 5$ megohms in small valves with self-bias, but 1 megohm is recommended. The resistance for power valves (10 W dissipation should not exceed $0 \cdot 5$ megohm.

9. Valves should not be operated with the cathode current cut off for long periods, when the heater is still connected to the supply.

Microphony.

Microphony in a valve is the sound produced by small rhythmic variations in the output, caused by variations in the electrode spacing when mechanical vibrations reach them from an external source.

The effects of microphony can be minimised by the following precautions :—

1. The loudspeaker should be insulated from the chassis by rubber bushes, and should not be too close to the valves in the early stages of the receiver.

2. The chassis should be mounted on rubber bushes and not fixed rigidly to the cabinet.

3. Valves which are followed by the highest degree of amplification should be mounted in resilient holders with short flexible connexions to the socket contacts.

4. Valves can be shielded from direct impact of sound waves from the loudspeaker by interposing other components.

Mercury Vapour Rectifiers.

These need special care in mounting and putting into commission. If the rectifier has been stored or shaken in transit, the mercury should be volatilised from the cathode surface by running the cathode at operating temperature for at least 15 minutes before applying the H.T. voltage. The filament and anode should never be switched on simultaneously except in very small rectifiers on low loads.

The filament end of the rectifier should be kept as cool as possible, and power rectifiers should have a forced draught cooling if adequate ventilation is not practicable.

Flashover is frequently caused by excessive temperature, and a safety fuse should be inserted in the anode lead as a precaution.

FIXED RESISTORS.

Colour Code.

The values of resistance are marked on fixed resistors by means of a colour code, based on the colours of the spectrum and including black, brown, and white. The colours are applied in one

* 16 Jermyn Street, London, S.W. 1.

of the three ways shown in Fig. 4. In addition a fourth colour denoting the tolerance is sometimes added. If no fourth colour is present, the tolerance is 20 per cent.

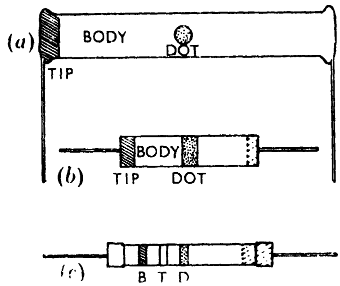

FIG. 4.—Three types of resistor, showing markings.

TABLE OF COLOUR CODE.

Body. (1st Significant figure).	Tip. (2nd Significant figure).	Colour.	Dot. (Number of zeros to follow).
0	0	Black*	0·0
1	1	Brown	0
2	2	Red	00
3	3	Orange	000
4	4	Yellow	0000
5	5	Green	00000
6	6	Blue	000000
7	7	Violet	
8	8	Grey	
9	9	White	

A convenient method of memorising the code is to associate the 'dot' colour with the following:—

Black	Tens	Yellow	Hundred thousands
Brown	Hundreds	Green	Megohms
Red	Thousands	Blue	Tens of megohms
Orange	Ten thousands		

The 'body' and 'tip' colours are then read in that order, mentally inserting a decimal point between the figures. Example:

Green—black . . . . . . 5 · 0
Brown—red . . . . . . 1 · 2

and the value of the dot is then read according to the above table.

Example: Green body, black tip, orange dot:
5 · 0 ten thousands.
or 50,000 ohms.

Note: Red body, green tip, green dot is read as
2 · 5 megohms.
or 2½ megohms, and not 25 megohms.

* Values below 10Ω are indicated by two black dots.

Preferred Values.

To simplify the stocks of resistors and the values specified in design practice, the Radio Component Manufacturers' Federation* has agreed on a series of preferred values of resistance which should be used in normal circumstances for radio receivers and electronic apparatus. Special values can still be supplied to requirements, but their use is deprecated. The table of preferred values is governed by the tolerance required, and the columns in the Table conform to the specified tolerances of 5 per cent., 10 per cent., and 20 per cent.

PREFERRED VALUES OF RESISTORS.

Tolerance.			Tolerance.			Tolerance.		
± 20 %	± 10 %	± 5 %	± 20 %	± 10 %	± 5 %	± 20 %	± 10 %	± 5 %
10	10	10	330	330	330	10,000	10,000	10,000
		11			360			11,000
	12	12		390	390		12,000	12,000
		13			430			13,000
15	15	15	470	470	470	15,000	15,000	15,000
		16			510			16,000
	18	18		560	560		18,000	18,000
		20			620			20,000
22	22	22	680	680	680	22,000	22,000	22,000
		24			750			24,000
	27	27		820	820		27,000	27,000
		30			910			30,000
33	33	33				33,000	33,000	33,000
		36	1,000	1,000	1,000			36,000
	39	39			1,100		39,000	39,000
		43		1,200	1,200			43,000
47	47	47			1,300	47,000	47,000	47,000
		51	1,500	1,500	1,500			51,000
	56	56			1,600		56,000	56,000
		62		1,800	1,800			62,000
68	68	68			2,000	68,000	68,000	68,000
		75	2,200	2,200	2,200			75,000
	82	82			2,400		82,000	82,000
		91		2,700	2,700			91,000
					3,000	100,000	100,000	100,000
100	100	100	3,300	3,300	3,300			
		110			3,600			
	120	120		39,00	3,900			
		130			4,300			
150	150	150	4,700	4,700	4,700	Higher values are in the same proportion.		
		160			5,100			
	180	180		5,600	5,600			
		200			6,200			
220	220	220	6,800	6,800	6,800			
		240			7,500			
	270	270		8,200	8,200			
		300			9,100			

FIXED CAPACITORS.†

Colour Code.

The colour code for fixed capacitors is based on the same series of colours as in the resistor code. Fig. 5 shows the arrangement of colours in two types of capacitor. In addition to the value markings there are colours for a complete range of tolerances from 1–10 per cent. and for D.C. voltage ratings. If only three colours are shown, it is assumed that the tolerance is 20 per cent. and the D.C. voltage rating is 500 volts.

* 22 Surrey Street, Strand, W.C. 2.

† It is now recommended that the term 'capacitor' be used in place of the original term condenser.'

The values are expressed in micromicrofarads (picofarads). A third significant figure is sometimes found in American practice, two of the colours being on one side of the capacitor and two on the other.

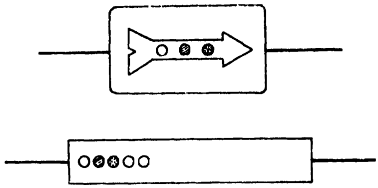

FIG. 5.—Two types of fixed capacitor, showing arrangement of colour dots.

BRITISH COLOUR CODE FOR FIXED CAPACITORS.

Colour.	Significance of First Three Colours.	Fourth Colour (Tolerance Per Cent.)	Fifth Colour (Voltage Rating).
Black	0		
Brown	1	1	100
Red	2	2	200
Orange	3	3	300
Yellow	4	4	400
Green	5	5	500
Blue	6	6	600
Violet	7	7	700
Grey	8	8	800
White	9	10	1,000

Note.—The American code is similar to the above, but has white 900 volt rating, gold 1,000 volt, copper 1,600 volt, and silver 2,000 volt.

Leakage.

A common cause of trouble in amplifier circuits is the use of capacitors which have a leakage current in excess of normal. It should also be appreciated that capacitors do not have an indefinite life in storage, and the use of old capacitors is a false economy owing to the risk of breakdown and faulty insulation.

The following table is based on the British Standard-Fixed Capacitors (B.S. 1082—1942). Group 1 includes all capacitors which are connected in a circuit in such a way that there is a risk

INSULATION RESISTANCE OF FIXED CAPACITIES (BRITISH).

Group.	D.C. Test Voltage.	Time of Application Min.	Rated Capacity μF.	Min. Insulation between	
				Terminals. $M\Omega \times \mu F$.	Capacitor and case $M\Omega$.
(1)	500	1	>1 ≯1	1,000 1,000	100 300
(2) Other than electrolytic	300	1	>1 ≯1	300 300*	30 100
Electrolytic with D.C. rated voltage	Rated voltage	3	All	7	100
Electrolytic with D.C. rated voltage (wet type)	Rated voltage	3	All	3	100

* Megohms only.

of shock if the capacitor develops a fault. Group 2 includes other capacitors to which no risk of shock is attached in event of failure.

The classification is for test purposes, and does not indicate the use to which the capacitor should be put.

Electrolytic Capacitors.

Electrolytic capacitors are frequently assembled in blocks in a common outer cover, the connecting leads being brought out at the ends. The values of capacity are also marked on the case in plain figures, but the connecting leads are colour coded to distinguish them. The arrangement of the internal connexions is identified as follows :—

+ A common positive connexion.	± Series connexion.
− A common negative connexion.	& Independent sections.

The spectrum order is also applied to the coloured leads, which then denote the capacities in descending order of magnitude :—

Positive connexions : red, yellow, green, blue, violet.

Negative connexions : black, brown, grey.

The centre tap of a voltage doubler capacitor is coloured white.

Examples.—(1) Two independent capacitors of $4\mu F$ and $2\mu F$ would be marked '4 & 2' and their leads would be coloured red and black for the $4\mu F$ section and yellow and brown for the $2\mu F$ section.

(2) Three capacitors of 8 ; 8 and $4\mu F$ with a common negative lead would have their leads coloured red, red, yellow and black.

Mains Transformers.

A number of transformers supplied for use in power rectifier units have coloured wires to indicate the windings to which they refer. The conventional code is shown in the drawing below.

Heater windings are usually of 18 or 16 s.w.g., while the rectifier heater winding may be 20 s.w.g. Some high tension transformers for use in cathode-ray tube equipment and those for use with half-wave or metal rectifiers have their secondary leads coloured red and red-and-yellow.

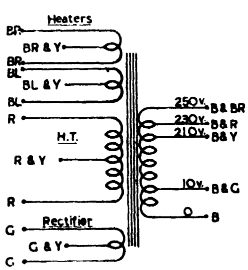

FIG. 6.—Colour code for transformer leads. BR = brown, BL = blue, B = black, Y = yellow, R = red, G = Green. Where two colours are indicated they are interwoven in the insulating sleeving.

If there is an electrostatic screen between primary and secondary windings it may be permanently connected to the core inside the transformer cover, or may be brought out to a flexible bare copper lead.

American power transformers are similarly colour coded but there may be variations in the rectifier filament winding (yellow and yellow-and-blue instead of green and green-and-yellow) and in the heater winding, which may be green and green-and-yellow. A voltmeter should always be used in doubtful cases in addition to checking the wire size.

In some American receivers the intermediate frequency transformers have their leads coloured according to the following code :—

H.T. wire . . .	Red	Grid wire . .	Green
Anode wire . . .	Blue	Earth wire . .	Black

This code can also apply to audio frequency transformers.

BRITISH COLOUR CODE FOR CARTRIDGE FUSES.

Colour.	Fusing Current.	Colour.	Fusing Current.
Black	60 mA.	Dark Blue	1·0 A.
Grey	100 mA.	Light blue	1·5 A.
Red	150 mA.	Violet	2·0 A.
Brown	250 mA.	White	3·0 A.
Yellow	500 mA.	Black and white	5·0 A.
Green	750 mA.		

BRITISH COLOUR CODE FOR WANDER PLUGS.

Red	H.T. + max.	Black	Common negative
Yellow	H.T. + 2	Brown	G.B. – max.
Green	H.T. + 3	Grey	G.B. – 2
Blue	H.T. + 4	White	G.B. – 3
Pink	L.T. +		

Note.—H.T. + 2 denotes second highest possible value, and so on.

MISCELLANEOUS DATA.

Attenuators.

An attenuator is a network of resistances or impedances designed to produce a known reduction in amplitude of a signal applied to its input terminals. For accurate working the impedance of the input source Z_1 and the impedance of the output load Z_2 should be matched. Attenuator networks are generally of three types: T-section, π-section, and bridged-T section (see Fig. 7).

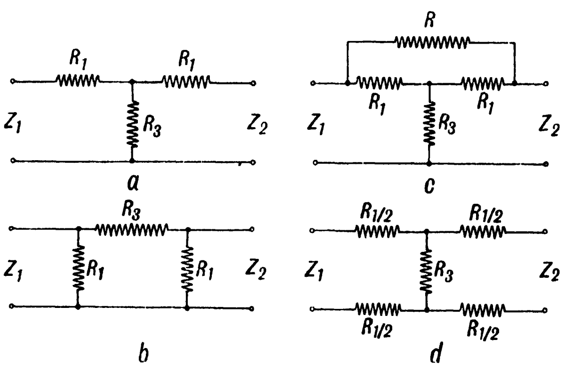

FIG. 7.—Types of attenuator circuit. (*a*) T-section, (*b*) π-section, and (*c*) bridged T-section. (*d*) is a balanced T-section to which the table refers.

The following table gives the attenuation in *db* for a symmetrical T attenuator for $Z_1 = Z_2 = 500$ ohms (Fig. 7*d*).

db.	R_1	R_2
1·0	28·8	4,330
2·0	57·3	2,152
3·0	85·5	1,419
5·0	140·1	822
10·0	259·7	351·4
20·0	409·1	101·0
30·0	469·3	31·65
40·0	490·1	10·00
50·0	496·8	3·162
60·0	499·0	1·000

Tone Control.

The majority of audio-frequency amplifiers require to be fitted with a form of tone control, either to suit the taste of the individual listener or, in the reproduction of gramophone records, to compensate for the deficiencies in the recording characteristic.

A variety of tone control circuits have been designed, based on the properties of resistance-capacitance or resistance-inductance combinations, and two basic forms are shown in Fig. 8.

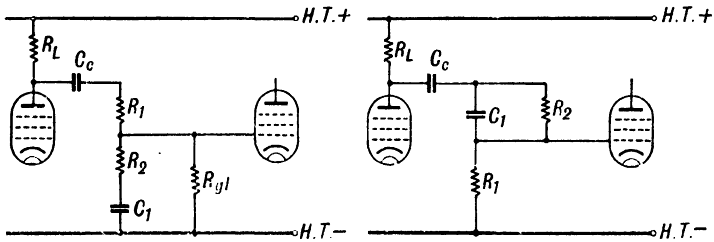

FIG. 8.

Details of Frequency Response.	*Component Values.*	*Details of Frequency Response.*	*Component Values.*
Bass lift, beginning at 300 c/s, rising 12 db at 50 c/s	R_L = 50,000Ω C_c = 0·01μF R_1 = 250,000Ω R_2 = 50,000Ω C_1 = 0·015μF Rgl = 1MΩ	Bass cut, beginning at 300 c/s, falling 11 db at 50 c/s	R_L = 50,000Ω C_c omitted C_1 = 0·001μF R_2 omitted R_1 = 1MΩ
Top cut, beginning at 2,000 c/s, falling 12 db at 10,000 c/s	R_L 50,000Ω C_c = 0·01μF R_1 = 250,000Ω R_2 omitted C_1 = 0·002μF Rgl = 1MΩ	Top lift, beginning at 2,000 c/s, rising 12 db at 10,000 c/s	R_L = 50,000Ω C_c = 0·02μF C_1 = 100μμF R_2 = 0·5MΩ R_1 = 0·1MΩ

Inductance.

Formulæ for the inductance of single layer coils are only strictly accurate for D.C. or low frequency A.C., as no method of allowing for uneven current distribution of current at high frequencies is available. The most widely used formula for single layer coils is that due to Professor Nagaoka :—

$L_s = 0{\cdot}00987\, D^2 n^2 l K$ microhenries.

where D = diameter of coil between wire centres (cm.)
n = turns per cm.
l = length of coil in cm. = $n \times$ total number of turns
L_s = inductance in μH. of a 'current sheet' inductance, *i.e.* a coil of infinitely thin tape with infinitely thin separation between turns
K = a factor depending on the ratio of D : l.

TABLE OF K FACTORS.

D/l.	K.	D/l.	K.
1·0	0·69	0·02	0·99
2·0	0·53	0·03	0·98
3·0	0·43	0·04	0·98
4·0	0·37	0·05	0·97
10·0	0·20	0·1	0·96

A closely wound coil with thin insulation approaches the ideal sufficiently accurately for the above formula to be used. If, however, the turns are spaced apart, a correction (due to Rosa) can be applied as follows :—

The reduction in inductance $\Delta L = 0{\cdot}006283\ Dnl(A + B)$. μH. where D, n, and l have the same significance and A and B are factors depending on the spacing and dimensions of the coil.

Rosa's Correction Factors.

$A = f(d/p)$ $B = f(D/l)$.

where d = wire diameter (cm.).
p = pitch in cm.

Ratio D/l	B.	Ratio d/p.	A.
0·02	0·11	0·1	− 1·55
0·03	0·17	0·15	− 1·35
0·04	0·2	0·2	− 1·05
0·05	0·22	0·25	− 0·83
0·1	0·27	0·3	− 0·65
0·5	0·32	0·35	− 0·5
1·0	0·33	0·4	− 0·36
2·0	0·33	0·5	− 0·14
3·0	0·33	0·6	+ 0·05
4·0	0·33	0·8	+ 0·34
10·0	0·34	1·0	+ 0·55

The value of ΔL calculated from this formula is then subtracted from the value found from Nagaoka's.

TABLE OF TURNS PER CM. LENGTH.

S.W.G.	Diam. mm.	D.W.S.	Covering. E. and S.S.	Enamel.
18	1·219	7·7	7·5	7·8
20	0·914	10·1	9·7	10·2
22	0·711	12·7	12·1	12·9
24	0·559	15·7	15·3	16·2
26	0·457	19·2	18·7	19·9
28	0·376	23	22	24
30	0·315	26	26	29
32	0·274	30	30	33
34	0·234	34	35	39
40	0·122	54	58	72

Effect of Screening Cans.

The effect of a screening can on the inductance can be calculated from the following table, which assumes that the top of the can is a distance equal to the diameter of the coil from the top

of the coil former. D_s is the diameter of the can and D the diameter of the coil. There is negligible difference between cans of square or circular cross-section.

REDUCTION IN INDUCTANCE DUE TO CANS.

Ratio of D_s : D.	Per Cent. Reduction.
10	0·13
8	0·25
6	0·6
5	1·0
3	4·5
2	15·0

Self-capacity of Short Solenoids.

Palermo gives :
$$C_s = \frac{\pi D}{3\cdot 6 \cosh^{-1}(p/d)}\ pF.$$
when l is not greater than D, and the symbols have the significance previously given.

REFERENCES.

'Calculation of Inductance and Capacity,' by W. H. Nottage. 'Design of Single-layer Coils,' by A. I. Forbes-Simpson, *Electronic Engineering*, November 1947, p. 353.

'Inductance of Single Layer Solenoids,' *Electronic Engineering*, Data Sheet, October 1941, p. 447.

Amplification.

The stage gain of a valve at audio frequencies is given by the following formulæ :—

1. At low frequencies $$G = \frac{G_m}{\sqrt{1 + \frac{1}{\omega^2 C^2 R_t}}}.$$

2. At medium frequencies $$G_m = \frac{\mu R}{R + R_a}.$$

3. At high frequencies $$G_h = \frac{G_m}{\sqrt{1 + \omega^2 C^2{}_1 r^2}}$$

where :

G_m = Gain at medium frequency.

R = Effective resistance of load resistor and grid leak in parallel ($R . R_g/(R + R_g)$).

R_t = Sum of grid leak resistance and effective resistance of load and internal valve resistance in parallel ($R_g + R . R_a/(R + R_a)$).

R_a = Internal (anode) resistance of valve.

r = Effective resistance of R and R_a in parallel.
$$= \frac{R.R_a}{R + R_a}.$$

C = Coupling capacitance in farads.

C_1 = Total shunt capacitance (including strays) in farads.

μ = Magnification factor of valve.

Typical Operating Conditions of Amplifiers.

The following table (from R.C.A. data) gives the amplification that can be obtained from a single stage, using the valve type specified with the circuit component values shown. The symbols used are explained in Fig. 9.

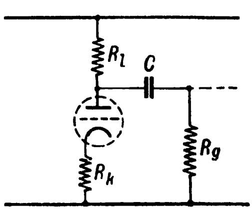

FIG. 9.

Valve Type.	H.T. Volts.	Load Resistance. R_l.	Grid Resistance. R_g.*	Cathode Resistance. R_k (Ohms).	Coupling Capacitor. C (μF).	Voltage Gain.
			TRIODES.			
6B6.G . .	180	0·1 M.	0·25 M.	2,900	0·015	36
6N7.GT .	180	0·1 M.	0·5 M.	1,950	0·007	22
6SL.7 . .	300	0·1 M.	0·22 M.	1,800	0·014	38
			PENTODES.			
6J7.G . .	180	0·1 M.[1]	0·25 M.	750	0·01	69
	180	0·25 M.[2]	0·5 M.	1,600	0·005	118
6SF7 . .	300	0·22 M.[3]	0·47 M.	1,300	0·005	88

Screen grid resistances: (1) 0·5 M.; (2) 1·18 M.; (3) 0·94 M.

* This refers to the succeeding stage.

The figure for voltage gain refers to an output voltage of 5 volts. M = megohms.

Classification of Amplifiers.

An amplifying valve can be used under varying conditions of operation, depending on the portion of the anode-current/grid voltage characteristic selected. The most usual method is to apply negative bias to the grid so that the anode current when no signal is applied is the value given by the mid-point of the straight portion of the characteristic curve. The signal then causes the anode current to vary linearly (within the limits of the straight portion) with the change in grid voltage. This method of operation, in which the anode current flows at all times during the cycle of changes, is called *Class A* operation.

Other methods of operating are:

Class AB. The initial bias is increased so that the anode current is cut off for part of the cycle when the grid is strongly negative.

Class B. The valve is initially biased to the point at which the anode current is cut off. The current then flows only when the grid potential is less negative than its initial value.

Class C. The grid is biased beyond the ' cut-off ' point, and thus the anode current only flows for a portion of the operating cycle.

Cathode Follower.

This is a form of circuit in which part of the output load resistance is included in the cathode circuit of the valve, the input being applied between the grid and earth in the usual way. A signal voltage applied to the grid will appear in the same phase across the cathode resistance, and the cathode potential may thus be considered to follow that of the grid—hence the name.

Although the impedance of the input is high, the output impedance of the cathode follower is low (being equal to $1/g_m$, the mutual conductance of the valve), and this enables the valve to be matched to a low impedance output without the use of a transformer. The voltage gain from the cathode is less than unity, and the valve can be used only for obtaining power gain. Other advantages of this circuit are: the input and output circuits have a common earth connexion; the frequency and phase response are good; the input capacitance is reduced.

Bel.—A unit used in comparing two amounts of power P_1 and P_2, defined as the logarithm of their ratio. The commonly used sub-multiple is the *decibel* ($\frac{1}{10}$th bel).

$$N_{db} = 10 \log_{10} \frac{P_1}{P_2}.$$

It is important to note that the decibel is not an absolute measure of power, but can only be legitimately used when the conditions are adequately specified.

The decibel is also used to express a ratio of two voltages or currents, in which case:

$$N_{db} = 20 \log_{10} \frac{V_1}{V_2}.$$

Matching Loudspeaker to Valve.

In audio frequency circuits it is of great importance that the impedance of the loudspeaker be matched to the optimum load impedance of the power valve which supplies it. This is done by

means of an output transformer with its primary and secondary windings tapped to give a number of different ratios.

The correct ratio is found by referring to the maker's data on the output valve for the optimum load impedance and relating the speech coil impedance of the loudspeaker to the transformer ratio from the chart (Fig. 10).

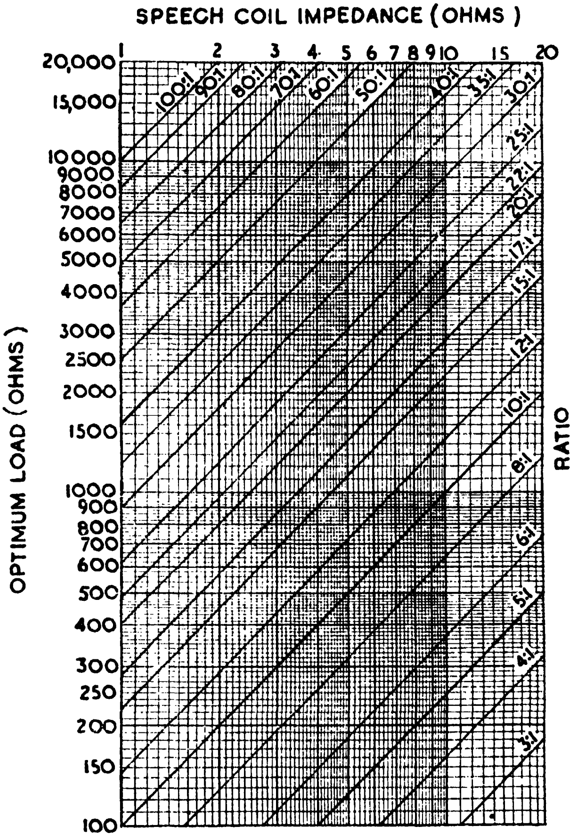

FIG. 10.—Chart for determining optimum transformer ratio between output valve and loudspeaker.

By courtesy of the Edison Swan Co.

SECTION XLII

PAINTS—VARNISHES—ENAMELS—LACQUERS, ETC.

(Contributed by Geo. H. Howse.)

(Pp. 977–997.)

SECTION XLII

PAINTS—VARNISHES—ENAMELS—LACQUERS, ETC.

(Contributed by Geo. H. Howse.)

Purchasing.

As the making of paints, varnishes, japans, enamels and lacquers is a highly skilled industry dealing with a great variety of raw materials and finished products for use in widely different trades and processes it is essential that, lacking a comprehensive knowledge of the trade, a buyer should be explicit in making up his enquiries and orders. During the last fifteen years standardisation of products by specifications has advanced considerably, and for general purposes it is possible to quote standard specifications for the tints of the paints, etc., and for all the major ingredients, *e.g.* oils, pigments, varnishes, driers, solvents, etc., as well as for the finished products. The increase in the use of spraying, stoving and baking, dipping, etc., processes adapted for the use of enamels, japans, cellulose, lacquers and synthetic-resin enamels, working under a wide range of drying times and conditions of 'finish' emphasises the necessity of placing the supplier in possession of every possible item of information about the materials and processes required. Tests of materials should be carefully controlled and the results reduced to a properly arranged schedule of economic values against the price. For many special requirements in the protection and decoration of plant, products and buildings, etc., there are proprietary products of firms of high reputation that satisfy the purposes where ordinary paints, etc., would certainly fail. It is poor economy to buy paints, etc., on their price alone as the covering power (cost per square foot) and the durability or service and cost of application are governed by factors to which the price is not sufficient guide. Reference to the tables and lists of specifications, etc., given later will be found useful in comparing the values of coating compositions, etc. With the highly-skilled technical knowledge and research information at the disposal of the modern manufacturer of 'finishes' there is rarely any difficulty in securing products that are completely satisfactory in service and cost.

It is vitally important to bear in mind that the success or failure of paint, enamel, lacquer, or other coating composition depends as much upon the painter or the operator applying it and the conditions of application and drying as upon the paint, etc., itself, and the paint manufacturer should not be held to blame for every difficulty that arises in finishing processes until it has been proved beyond any doubt whatever that it is his material which is at fault. It will be found in practice, however, that the technical and service departments of the larger firms are always ready to co-operate in any painting problem, to reduce the possibilities of failure to the minimum.

Storage of Paints, etc.

Keep paints, varnishes, enamels, etc., in a cool dry store, away from flame or naked lights. See that all packages are securely closed when not in use to avoid loss of material, skins and contamination. Materials to which the Petroleum Acts, the Celluloid and Cellulose Solutions Regulations apply must be kept in fireproof, isolated and locked stores under responsible control and licence of the local Authorities. Buy paints in the most suitable and economical packages, *i.e.* common paints (red and white lead, zinc red oxides), oils, etc., in ½-cwt. drums; general-purpose paints, enamels, varnishes, cellulose and synthetic enamels in regular consumption in 1-gallon cans. Paints, etc., in occasional use should be bought in ½- or ¼-gallon cans. These smaller packages add little, if anything, to the cost and save much time and material in storage and handling.

Marking of Lead Paints.—Under the Lead Paints Act it is the duty of the *purchaser* to *instruct* the supplier to mark the packages to comply with the Acts. The supplier is not bound to do so without instructions.

Paste Paints should be kept covered with raw linseed oil (not water).

Distempers should be kept in a cool dry place (covered with petrifying liquid when necessary). Avoid exposure to frost or heat (above 80° F.).

Tints should always be checked before use. For large uniform surfaces, order sufficient of the colours required at one time or mix the contents of the several containers in one large vessel to ensure consistent shade.

Strain all paints, etc., taken from partly used containers to remove skins, etc., using metal gauzes wherever possible. In dip tanks and continuously operated plants regular clarification or straining of the paint by mechanical means is essential if finishes are to be obtained of the highest quality.

Thinners should be obtained from the maker of the material or to comply with the necessary specifications. Follow instructions carefully because many products, *e.g.* stoving japans, cellulose, lacquers, synthetic enamels, spirit varnishes, coach varnishes, are ruined by the addition of even small quantities of incorrect materials.

Brushes should be thoroughly cleaned in the proper solvents or thinners after use and hung up to dry or kept in the proper liquids between service. Never keep brushes in water; more painting troubles can be traced to moisture than anything else, and every care should be taken to avoid it. Old-fashioned painters still adhere to the practice when using white lead, but the results condemn the practice with other materials.

Distemper brushes should be thoroughly washed in hot soapy water and dried in a cool place hanging free in a current of air.

Flat rubber-set brushes of good thickness w'th medium length bristles give the best service.

Varnish brushes should be kept in thin varnish in a brush-saver between service.

Painting.

General Precautions.

All surfaces are to be cleaned free of dirt, rust, grease, scale and cracked or perished paint or tar. Perished paints are to be stripped off down to the bare surface by the use of blow lamp or liquid stripper. Blow lamp is best for plain surfaces, stripper for intricate detail or mouldings and glazed work—a combination of the two for painted plaster and very old enamelled work. The bare wood surface is to be made smooth, knotted with best shellac varnish, primed, stopped and filled before applying further paint. Bare steel should be scraped and scratch-brushed and primed with either special steel primer or red lead, etc.

Priming Paint for Wood.

Red lead or white lead with silicate fillers in refined linseed oil and high grade liquid driers and turpentine make good primers. Special paints of high quality are made by all good paint makers. Avoid water paints, glue and size compositions and common quick drying paints or mixtures of smudge residues as primers for outside exposure. Allow any paints to dry thoroughly before the next coats are applied. Aluminium powder mixed with high grade shellac varnish for knotting or with highly elastic waterproof oil varnish for general services provide excellent primers on American or Canadian pine woods where other kinds of priming paints prove unsatisfactory. On damp and unseasoned timber or under very moist conditions ethyl cellulose base primer lacquers are very valuable.

Priming for Steel.

Red lead, orange lead, white lead, blue lead, zinc dust, iron oxide and silicates, lead, zinc and barium chromates and other rust inhibitive pigments are good for steel in combination with refined and polymerised linseed oils, water-resisting varnishes and synthetic-resin media. The correct preparation of the oils and varnishes and the proper incorporation of the pigments with them to the right consistency is very important. Proprietary paints are based on long research and experience, and undoubtedly give the best results in marine, chemical-laden or tropical conditions. Bitumen solutions, particularly coal tar compounds or graphite, *i.e.* carbon black, vegetable black, lamp black and plumbago paints, should not be used alone as priming paints on steel. It is vitally important that all rust scale, dirt, grease and moisture be removed from the surface of the metal, and the loose rust scratch-brushed or treated with rust remover. Paints should be applied during the drier months of the year, *i.e.* June to September inclusive.

Spray Painting.

Spray Painting has very largely displaced brush application, especially in industrial finishing and the protection and decoration of large buildings and structures. Many types of spray pistol are available and the correct one should be selected for the paint or lacquer, etc., to be applied as the spreading power, viscosity, atomisation and drying of the paints, etc. are affected by the design and controls of the 'gun.'

For outdoor use, especially on steel surfaces, particular care has to be used in applying ordinary linseed oil paints as moisture condensed in the atomising air causes rapid corrosion under the paint surface. Special types of paints are made to minimise this fault.

The best working pressure lies between 45 and 60 lb., and for large surfaces a pressure-controlled paint container to feed the pistol gives quickest results. The spraying plants of the best known engineers are highly efficient and effect great economies in paint application. Paints containing lead oxide exceeding 5 per cent. of the pigment cannot be sprayed by women or young persons

and special regulations govern their use by men. As efficient leadless (maximum 5 per cent. lead oxide in the pigment) paints can be obtained in every required tint or finish, however, there is no obstacle to their use for every possible painting operation. Many novelty and special finishes owe their success to the use of spray plant in conjunction with conveyor drying and baking ovens.

Spraying Plant must be kept clean and free from dried paint or lacquer dust. Booths and fume pipes should be regularly scraped with brass scrapers and the clean bare metal coated with motor grease or soft soap. If newspaper is pasted on to the grease and coated with used motor oil the spray paint and dust will be easily stripped off when necessary. Walls and floors should be regularly freed from dust and dirt. Coat the walls with oil or grease and varnish or paint the floors to facilitate cleaning.

Pressure tanks should be blown out at the end of each shift, keep air lines and filters free of oil and water. Pistols should be kept thoroughly clean, allow to soak occasionally in strong solvent. Worn thimbles and needles should not be used.

Ventilation should be adequate—a good working standard is 5 changes per hour in the working room or space, and 75 cu. ft. per minute in the exhaust outlet of each booth. Remember that dust is drawn (from adjacent floors, windows, doors and cracks or crevices in roofs) into the spray booths by the exhaust draught.

Therefore to prevent spoiled work and unfair complaints about the paints, the access of dust must be prevented by every possible precaution and the removal of litter, etc. All lights and switches should be gasproof and totally enclosed. Stores for the paints, etc., should comply with the factory regulations applicable. Fire extinguishers and sand should be kept at hand in all stores and spraying rooms. Consult the suppliers of paints, etc., for information about the regulations which must be complied with in the plant and premises. Exits should be adequate and kept free from obstructions.

Stainers.

Stainers (tinting colours) should be purchased of good strong quality in semi-liquid form, packed in ½-gallon or ¼-gallon cans to ensure best and most economical results. The use of cheap paste stainers has nothing to recommend it.

High purity ochres, umbers, siennas, Venetian reds, Vandyke browns, yellow, brown, red and maroon oxides of iron, drop blacks, ultramarine, Prussian and Brunswick blues, chrome yellows and greens, lake yellows, greens, scarlets, maroons and other tints should be of the finest possible grinding, and lake colours should be permanent 'fast' shades. Do not expect 'fast' brilliant colours to be low in price and high in staining strength as well. Buy your staining colours as pure as you possibly can, use and store them carefully, and the results will be richer tints and better work at lowest possible costs.

Leadless Paints.

For many painting purposes high percentage lead pigments cannot be used, *e.g.* in sulphur-laden atmospheres, Food and Ordnance factories, Industrial processes employing women sprayers, interior decoration, heat-resisting finishes, and for certain colours. Therefore other suitable pigments have to be used. These are sometimes more expensive than lead pigments, *e.g.* titanium and antimony white oxides, cadmium yellow and selenium red sulphides, so that they cannot be called substitutes. Other pigments may be cheaper than white lead, *e.g.* lithopone, zinc oxide and reduced titanium oxide compounds, but they serve a sphere of usefulness for which white lead would be unsuitable in any case and too expensive. It is also a fact that there are many brilliant pigments which have greater permanence than any lead pigments, *e.g.* Prussian and ultramarine blues, zinc, cadmium and barium yellows, iron oxide yellows, cadmium selenium, orange and red pigments, Azo lake yellows, reds, oranges, scarlet, carmine and maroon colours, chromium and lead or zinc chromate. Greens and lake blues are also very permanent, and the addition of lead pigments to them is a detriment rather than an improvement. The range of iron oxides produced in such rich tints of yellows, browns, Venetian, Turkey and Indian reds and blacks are themselves far more permanent than lead pigments, and they do not need the addition of lead to brighten their colours or improve the covering power and durability. The technical research and advances made by the colour-making industry during the past thirty years have resulted in the production of new pigments which, in conjunction with the media resulting from intense research by the paint trade during the same time, now provide paints of greater durability under extreme conditions than could ever be provided by 100 per cent. lead-linseed-oil paints, and furthermore, the range of colours available is much wider than if lead pigments only were to be used. The old fetish of the superiority of white lead paint is steadily passing away and the spread of more knowledge of paint and painting materials is enhancing the confidence that the leadless paints can be used for exterior exposure. The use of leadless pigments for interior work has been absolutely essential for many years, and lead paints under such conditions are at a great disadvantage.

Paints for Aircraft.

The paints required have to be prepared under the most careful technical control, skilled testing and inspection to specifications which are among the most rigid in the paint, etc., industry. For obvious reasons the details cannot be referred to. The Departments concerned must be

referred to. The Departments concerned can supply the necessary information to bona-fide users. The principal paint manufacturers are usually familiar with the details of the specifications required.

Under civil conditions the B.S.I. Specifications for aircraft materials and components comprise the following :—

Paints and Dopes.

2D8.	Nitro-cellulose syrup.
4D100.	Air Ministry cellulose acetate dope.
D102.	Nitro-cellulose dope.
2D101.	Doping schemes.
2D103.	Nitro dope coverings and identification colours.
D105.	Pigmented nitro-cellulose dope.
4X2.	Oil and petrol resisting battleship grey paint.
3X4.	White dope-resisting paint.
3X6.	Varnish for external woodwork.
3X7.	Varnish for internal woodwork.
2X8.	Undercoating propeller varnish.
3X9.	Bituminous paint.
2X11.	Transparent woodfiller for propellers.
2X12.	Finishing propeller varnish.
2X14.	Priming varnish.
X16.	Undercoating grey paint.
2X17.	Seaplane varnish.
X18.	Shellac varnish.
X19.	Acid-resisting paint.

The ingredients of these paints and dopes are covered by a long list of 'D' specifications including solvents, plasticisers, pigments, etc. Full details of these can be obtained from the British Standards Institution lists. The Air Ministry Specifications comprise :—

DTD56B.	Stoving enamels.
DTD62B.	Pigmented oil varnish.
DTD63A.	Cellulose enamels.
DTD83A.	Seaplane dopes.
DTD103.	Air screw lacquers.
DTD145 & 157.	Identification colours.
DTD226.	Paint remover.
DTD235.	Stoving enamels.
DTD260A & 314.	Pigmented oil varnishes.
DTD308.	Cellulose finishes.

All these paints and enamels have been perfected after long research and experimental work and are produced under rigid technical control, continual testing and inspection. In addition to the above-mentioned paints there are very large numbers of special paints, enamels and compositions prepared and approved for special purposes, details of which, of course, are not available. It can be said without hesitation that the standard of paints and compositions and the skill and knowledge required to successfully produce them represent the highest standards in paint manufacture for large scale production, and that firms on the approved lists of suppliers can be relied upon for the unfailing quality of their products.

HEAT-RESISTING AND FIREPROOF PAINTS.

The composition of these paints varies according to the heat and conditions to which they are exposed, *e.g.* paints for smokestacks differ widely in composition from paints for treating woodwork which may be exposed to flame. For many years combinations of water glass with pigments which may be of the reactive type, combined with alkali borates, have been used effectively on wood and fibre constructions. Similarly, pigments produced at high temperatures, *e.g.* Indian reds, vegetable blacks, titanium whites, incorporated in media consisting of heat-treated oils and synthetic resins which polymerise on exposure to heat, together with combinations of bitumens of high melting points, are used for metal work exposed to high temperatures. The pigments are usually inert and refractory and comprise asbestos, kaolin, barytes, oxides of zinc and titanium with heat-resisting staining colours, *e.g.* ultramarine blues, vegetable blacks, cadmium yellows, selenium reds, chrome greens, aluminium powder, etc. In the event of fire the fire-retarding nature of the paint is more important than its colour or covering power, and the incorporation of ammonia compounds in the paints is often useful. For fabrics, canvas, hessian and cotton goods special dressing solutions containing borax, boracic acid, salammoniac, alum, alkali phosphates are employed. Solutions of cellulose ethers and cellulose acetates, chlorinated rubber and chlorinated resin compounds are now available in clear and coloured finishes which can also be applied to woodwork. These compositions, when properly compounded, do not burst into flame even under the worst conditions and are definite fire retardents.

Bituminous Solutions and Paints.

There are many 'bituminous' materials from which these solutions and paints are made and include coal tar and water gas pitches, petroleum pitches, natural bitumens and asphaltums, rosin and wood pitches, stearine pitches, etc. Solutions of these in solvents, *e.g.* coal tar naphtha and white spirit, with or without drying oils and gum resins constitute the 'Bitumen Solutions' of commerce, and they may be pigmented with oxides of iron, lead and zinc colours, chrome greens and fillers or extenders as may be required. Specially refined and decolourised bitumens or pitches which are a deep brown colour or the specially extracted resins from certain coal tar products may be used in the production of so-called light bitumen paints, *e.g.* white, cream, grey, etc. Actually, a completely decolourised or white bitumen does not exist, the palest of them being straw colour which darken on exposure to light. Therefore the palest colours (white, light greys, cream, etc.) cannot be made in genuine bitumen solutions. Such light colour bitumen paints, and indeed many of the darker colours, usually consist of a quick-setting elastic oil-varnish medium which includes varying proportions of bitumen varnishes with metallic oxide and silicate pigments. The natural black or dark bitumens and asphaltums are remarkable for their resistance to acids, alkalis and moisture, especially when properly compounded for application to wood or cast iron. The purity of the bitumen, its origin and its treatment, all combine to secure these durable properties. The covering power of bitumen solutions may reach 1,200 sq. ft. per gallon in the unpigmented types and 900–1,000 sq. ft. per gallon in pigmented qualities. They are usually cheap, easily applied by unskilled labour, and dry rapidly in any climate and by reason of their peculiar properties are ideal for many conditions, especially underground. The best grades contain 60–65 per cent. of plastic bitumen. Special types of bitumen solutions are prepared having great resistance to heat and are therefore suitable for industrial plants, boilers and pipelines. Long experience has shown that bitumen paints and solutions do not give the best results when applied direct as primers to steel surfaces, and that there is a small optimum amount of bitumen which may be incorporated in a paint which is to give service equal to that of the best quality oil varnish paints. Good priming paint made with metallic oxides, chromates and tough drying oil-varnish media, followed by the bitumen paints or solutions, are much better than bitumen paints alone. In any case bitumen paints have a much shorter durability than good quality oil paints, and should be looked upon as more or less temporary in their protection. Bituminous emulsions are prepared by the incorporation of special types of plastic bitumens with water in the presence of emulsifying and stabilising agents, with the addition of the necessary pigments and extenders to produce the colours required. They can be made in glossy or flat finishes, in rough or smooth textured surfaces, and for camouflage purposes are being used in enormous quantities, affording a cheap, fairly durable and easily applied substitute for linseed oil paints. For the treatment of cement surfaces they should be superseded by a special primer if the cement is new, but on old weathered brickwork, cement, plaster or asbestos sheets, two coats are usually ample, whilst on galvanised or painted sheets, steelwork, glass, wood and tarred roads, fabric, vegetation, etc., one coat is sufficient.

Red Lead Paints.

The advantages of the 'non-setting' types of red lead for making paints for steelwork are now universally appreciated.

Red lead complying with B.S.I. Specification 217A, containing about 25 per cent. lead peroxide, is largely used, although superior results are obtained by using the type B.S.S. No. 315 (217C/1936) containing about 32 per cent. lead peroxide. The old type of red lead which, owing to its rapid setting properties had to be used in conjunction with white lead to keep it workable for a reasonable length of time, is now largely confined to the preparation of red lead jointing pastes. It contains about 15 per cent. lead peroxide.

The only B.S.I. specification covering red lead liquid paint (ready mixed) is contained in B.S.I. Specification No. 153, and is given as follows:—

Red lead, B.S.S. No. 217A	33 lbs.
Boiled linseed oil	$\frac{2}{3}$ gallon.
Raw linseed oil	$\frac{1}{3}$,,

This material is very thin and does not give as good results as a mixture of 80 per cent. red lead to B.S.I. Specification No. 217A or C with 20 per cent. of raw or boiled linseed oil, with or without the addition of small quantities of turpentine or turpentine substitute. The covering power of red lead paints to the above specification is about 50 sq. ft. to the lb. and owing to the very high gravity (often 30–32 lb. per gallon) the use of 'Highly Dispersed' red lead is gradually gaining favour. These highly dispersed red leads are very fine in texture, of very high covering power, and produce paints weighing about 15 lb. per gallon containing up to 60 per cent. of oil which have the same covering power as the older types of red lead weighing 30–32 lb. per gallon. There is, therefore, a considerable saving in the weight of red lead used per ton of steelwork, and the anti-corrosive qualities of the highly dispersed red lead paint are superior to those of the older types. These new highly dispersed red leads are less prone to set or settle than non-setting types of red lead, and their greater use in the future is to be expected. Careful preparation and testing of the linseed oils for making red lead paints are very important, very low or

high acid value and over-boiling must be avoided. The old custom of melting a little *tallow in the oil* before mixing with the lead has much to recommend it.

A new specification, B.S. 1011 (with a War Emergency Revision), was issued in 1942, and covers two types: Type 1, for genuine red lead paint permits a maximum of 4 per cent. of extender to prevent caking; and Type 2 permits of extender not exceeding 15 per cent. Particular care should be exercised in selecting the kinds of extender used for these paints, or they defeat the object for which they are incorporated if of unsuitable quality.

White Lead Paints.

Genuine dry (basic hydrated carbonate) white lead should comply with B.S.I. Specification 239—1936, paste white lead should comply with B.S.I. Specification 241—1935, and ready mixed white lead with B.S.I. Specification 261—1936. Under a legal decision 'genuine white lead' may consist of white basic sulphate of lead. When ordering, the correct description should be specified to avoid confusion. In actual durability there is little difference, in fact on steelwork the basic sulphate gives superior results. It does not, however, permit of the liberties in use and making-up that do not affect the basic carbonate. The consistency of white lead pastes may be varied considerably to suit various markets.

Basic carbonate lead white mixed paint weighs about 28 lb. per gallon and covers 50–60 sq. ft. per lb. (1,500 sq. ft. per gallon or 6,000 sq. ft. (700 sq. yds.) per cwt.) on smooth hard-primed work. Lower figures must be taken for priming on wood and higher figures for tinted paints, *e.g.* grey paint covers 75 sq. ft. per lb. The average life of a white lead linseed oil paint (exterior, in town atmospheres) is about three years. White lead paint should be purchased in ready-mixed condition wherever possible. The heavy paste entails much labour in breaking down and results are not consistent.

A War Emergency Specification, B.S. 929, was issued in 1942 and amended in 1943, and includes Tinted White Lead Base Primer (929/P.1), White Lead Undercoating Paint (929/U.1), Tinted White Lead Undercoating Paint (929/U.4) Finishing White Lead Paint (929/F.1), Tinted Finishing White Lead Paint (929/F.5). Full details of composition for each type are given in the Specification.

It has been found that paints containing very low percentages of white lead in the light tints fail to pass the corrosion tests set out in this War-time Emergency Specification, and a minimum of 20 per cent. is desirable (as suggested in B.S.S. 1057/1942) for substitute paints for exterior finishing.

Gloss Paints and Enamels.

The combination of a finely divided pigment and varnish results in a coating with a glossy finish. The composition may vary from the cheapest Venetian red with the commonest 'oak' varnish used for painting gas pipes to the superbly ground and finished royal purple pigment in the highest grade enamel body varnish. Between the two extremes lie the hundreds of types necessary for the painting of houses, ships, transport vehicles, machinery, engines, steel-work, etc., each type requiring selected pigments and varnishes for its purpose.

Gloss paints and enamels effect a saving of time in application and give increased durability over finishes obtained by varnishing over flat colours and permit of more brilliant and attractive colours to be used. The covering power of gloss paints is very high—many cover 1,200 sq. ft. per gallon—the average being about 950 sq. ft. per gallon. They weigh from 12½ to 18 lb. per gallon according to tint and dry in 6 to 14 hours, covering solid at one coat. The best types are very durable, being superior to varnished work. They are made in lead base and leadless types. ENAMELS are heavy bodied, more elastic, slower drying types of gloss paints for interior use for very hard wear.

In the manufacture of the varnish media, high-grade copal gum resins, *e.g.* congo, animi, kauri, with refined and processed linseed and wood oils and the usual solvents and turpentine, or synthetic resins of the alkyd or phenolic types of special elasticity and durability give the best results for exterior work.

The production and processing of these materials require considerable skill and experience to yield highest grade products. Post-war conditions have restricted supplies of wood oils and certain synthetic resins of the phenolic types, but considerable research and practical work has been done to make use of other drying oils, *e.g.* processed castor oil. The resulting finishes are proving very satisfactory and have marked a new epoch in paint technology.

The same post-war shortage has also restricted the use of linseed oils and a number of pigments in paints, and therefore alternative specifications have been issued by the B.S.I. to cover other drying oils and pigments for wartime emergency use. Fuller details are given under 'Specifications.'

In 1943 specifications were issued for a number of paints, including Finishing Oil-Gloss (B.S.S. 929), Types F.1–F.14 inclusive, and B.S. 1056, Schedule 2, for specially hygienic finishes for hospitals, dairies, food factories, explosive factories, decontamination centres.

Owing to the control and restriction upon the use of certain pigments, oils, etc., it was not possible during war-time to produce gloss enamels of the high quality available prior to 1940, and the technique and methods of manufacture of these gloss paints and enamels had to undergo drastic changes.

Fuller information about these changes can be obtained by reference to B.S. Specifications 925, 926, 927, 928, 929, 1056 and 1057.

Paints for Woodwork.

The moisture content of building timber varies from 2 to 35 per cent. by weight. Successful paint for woodwork must therefore be capable of adhering to a surface that is continually changing in shape and porosity. The efficiency of white lead in the painting of wood is due to its capacity of retaining and discharging a considerable quantity of moisture without materially affecting the elasticity of the paint film. Zinc oxide on the other hand has no capacity for moisture and scales and flakes off badly as the film of paint is dissolved from behind. Titanium oxide has such intense surface energy that it passes through the binding medium and chalks badly. Lithopone forms an emulsion and is washed away eventually. Oxides of iron are the most inert coloured pigments, and when used in combination with thoroughly waterproof drying media give the most durable results. Lead chrome yellows, red and orange leads, lead base greens and blues are fairly stable, but the effect of the lead pigments on the drying oils reduces the durability. Lamp black pigments are durable if the drying is not forced, zinc base or lead base tinted whites (creams, greys, etc.) are very satisfactory if not reduced by the use of extenders in excess.

The addition of 5 per cent. or 10 per cent. of asbestine and up to 25 per cent. of silica or barytes is not detrimental to a pigment of high opacity and fine division.

Priming paints for wood should be made with double boiled (bodied) linseed oil and turpentine —be easily brushable and dry to a firm hard half gloss surface in 16 hours.

Undercoats should be eggshell matt, be tough and firm (not brittle) in 12 hours.

Finishing gloss paints or varnish paints, enamels or varnishes should dry dust-free in 4 to 6 hours, and be tough and elastic in 24 hours. Tests for varnishes are given in B.S.I. Specification Nos. 256 and 257—1936.

All the precautions with regard to buying, storage and application of paints given in the early paragraphs of this section should be observed in dealing with paints for woodwork. Every maker of repute *should* provide satisfactory paints for wood.

Painting Generally.

As very few of the old type of skilled craftsman painters who knew how to mix their paints properly for the work in hand are left in the trade, the application of paints, varnishes and enamels is now largely in the hands of semi-skilled labour or of men specialising in the use of 'proprietary' paints.

This is an era of ready-mixed paints, whether for brush, spray or dip application, problems of which have been solved by many paint manufacturers, each with a 'proprietary' material of its own for every particular purpose.

The semi-skilled painter, working under 'contract' conditions, demands easy-flowing, fast-drying, smooth finishes that require little rubbing down, have good opacity, and set tough and firm under almost all conditions. Under such conditions painting is lower in cost and factories, industrial plant, public buildings, houses, hospitals and institutions are kept in a better, more attractive and durable condition at a lower annual expense.

Conditions, however, can and do arise when ignorance or curiosity cause painters and those in charge of painting jobs to ignore certain basic principles of paint application and to meddle with 'proprietary' materials with which they are supplied, particularly if the contract price has been cut to very fine limits.

If, therefore, complaints are received about the behaviour of the productions of well-known firms, investigation should be very thorough. The condition of the wood or metal surfaces that have been treated, the weather conditions during application, the methods of application, and, above all, the treatment of the materials themselves, particularly as regard the addition of turps, spirits, boiled oil, paraffin, linseed oil, or the mixing of old 'overs,' and the care and maintenance of the brushes, spray guns, etc., employed, should be very thoroughly dealt with.

The Service Department of the modern paint manufacturer is a very efficient organisation, has a thorough knowledge of paint and painting problems, and can be relied upon not only to assist in obtaining the required finish and durability of its firm's materials, but also in counteracting any attempts at imposition or evasion in the obligations of the painting contractors.

Steelwork Paints.

This section of the paint industry has been the subject of the most intense research and trial during the last twenty-five years. The whole question of the cause of corrosion, its development and prevention have been the close study of technical and engineering bodies all over the world. The tests have proved the necessity of providing not only pigments of the required colour and opacity, but the inclusion of substances capable of preventing iron passing into solution in water and of converting the active iron compounds (hydroxides) into inert material within the binding medium used to form the film. As it is impossible to secure either a completely dry and rust-free iron or steel surface in general constructional steelwork practice, the importance of securing effective compositions for the protection of the steel can be understood. Thorough cleaning and scratch-brushing followed by treatment with paints containing iron oxides of high purity, red leads, lead chromates, zinc chromates, orange lead, basic white leads, blue leads, in oil-varnish vehicles, has been found effective. The most successful depended on the waterproof and elastic properties of their mediums. Linseed oil alone is not the best vehicle. Inert pigments such as graphite (plumbago), micaceous iron ore (Graphax), either alone or in combination with iron oxides, white lead, blue lead, zinc-lead oxide or aluminium powder have proved successful over satisfactory priming coats. The introduction of spraying plants has added to the problems of the steel paint chemist who has reinforced his media with synthetic resins to solve the difficulty. The value of properly made primers and protective coats has been realised, and there is not the cheese-paring of prices that was prevalent years ago. The best types of steel paints will ensure certainty of results under conditions which would be fatal to the old style red or white lead, linseed oil or oxide paints that were the stand-by of all engineers. The pre-treatment of the steel by chemical de-rusting compounds is now a recognised process in steel painting. The use of baked primers and finishes on portable parts in the sheet metal industry demanded such metal-cleaning material, and the method has extended to constructional work on a large scale. The next step was to include rust-converting compounds in the paint itself, and this is now the basis of a well-known patent. The life of steelwork paints of the best quality can be averaged at five years. Highly efficient paints and gloss enamel for every type of steelwork can be procured from the established firms, but in spite of many years work and research no universally satisfactory specification has yet been issued for gloss paints for steelwork.

Specifications of Paints, etc.

Nearly a thousand specifications for paints, pigments, oils, lacquers, varnishes, enamels, compositions, colour tints, etc., are issued by the British Standards Institution, the Government departments, railways, shipping companies, corporations, engineers and architects, the British Colour Council and professional bodies.

The majority of paint requirements are covered by specifications, but many important materials are not yet covered. Reliance can be placed on the products of well-established firms for the special materials, and information about standard specifications can be usually obtained freely from them. The B.S.I. specifications most frequently required by engineering contractors for paints are as follows:—

B.S.S. No. 197.	Black oil pastes, Grade 1, Grade 2.
B.S.S. No. 217.	Red lead, dry, for paints, Types A, B and C.
B.S.S. No. 239.	White pigments for paints. Genuine basic carbonate of lead.
B.S.S. No. 254.	Zinc oxide, Types 1 and 2.
B.S.S. No. 296.	Lithopone.
B.S.S. No. 338.	Antimony oxide.
B.S.S. No. 392.	Titanium dioxide.
B.S.S. No. 636.	Titanium white.
B.S.S. No. 637.	Basic sulphate of lead.
B.S.S. No. 241.	Genuine white oil pastes for paints.
B.S.S. No. 273.	Zinc oxide oil paste.
B.S.S. No. 297.	Lithopone oil paste.
B.S.S. No. 242.	Linseed oil, refined.
B.S.S. No. 243.	Linseed oil, raw.
B.S.S. No. 244.	Turpentine, Type 1.
B.S.S. No. 245.	White spirit.
B.S.S. No. 290.	Turpentine, Type 2.
B.S.S. No. 256.	Interior oil varnish, Types 1, 2 and 3.
B.S.S. No. 257.	Exterior oil varnish, Types 1, 2 and 3.
B.S.S. No. 258.	Flatting or rubbing oil varnish, Types 1 and 2.
B.S.S. No. 274.	Extra hard drying varnish, Types 1 and 2.
B.S.S. No. 259.	Boiled linseed oil.
B.S.S. No. 261.	White lead ready-mixed paint.
B.S.S. No. 262.	Tinted ready-mixed white lead paints, Types 1 and 2.
B.S.S. No. 277.	Zinc oxide base white ready-mixed paint.
B.S.S. No. 278.	Zinc oxide base tinted ready-mixed paints, Types 1 and 2, Grade L and H.
B.S.S. No. 293.	Green ready-mixed paints, lead chromate base, Types 1, 2, 3 and 4.
B.S.S. No. 294.	Black ready-mixed paints, Types 1, 2 and 3.

B.S.S. No. 295. Red oxide of iron ready-mixed paints, Grade L and H.
B.S.S. No. 371. Purple brown red oxide of iron ready-mixed paints.
B.S.S. No. 381. Colours for ready-mixed paints.
B.S.S. No. 391. Tung oil for paints.
B.S.S. No. 654. Perilla oil for paints.
B.S.S. No. 298. Red oxide of iron oil paste, Class 1. Natural or mixed oxides.
B.S.S. No. 299. Red oxide of iron oil paste, Class 2. Oxide of iron base.
B.S.S. No. 303. Brunswick green for paints (pure and reduced).
B.S.S. No. 304. Brunswick green oil paste for paints (pure and reduced).
B.S.S. No. 311. Varnish gold size.
B.S.S. No. 337. Ochre for paints.
B.S.S. No. 331. Paste driers.
B.S.S. No. 332. Liquid driers.
B.S.S. No. 388. Aluminium (powder and paste) for paints.
B.S.S. Nos. 390 and 393. Oil pastes. General colours.
B.S.S. No. 544. Linseed oil putty, Types 1 and 2.
B.S.S. No. 3011. Paints. Identification for engine-room piping.
B.S.S. No. 954. Shellac.
B.S.S. No. 284. Black carbon pigments.
B.S.S. No. 285. Bone black.
B.S.S. No. 286. Vegetable black.
B.S.S. No. 287. Lamp black.
B.S.S. No. 288. Mineral black.
B.S.S. No. 272. Natural red oxides of iron.
B.S.S. No. 305. Manufactured red oxides of iron.
B.S.S. No. 695. Blended oxides of iron.
B.S.S. No. 306. Black oxide of iron.
B.S.S. No. 339. Purple oxides of iron.
B.S.S. No. 851. Yellow oxides of iron.
B.S.S. No. 370. Venetian red.
B.S.S. No. 312. Sienna.
B.S.S. No. 313. Umber.
B.S.S. No. 319. Vandyke brown.
B.S.S. No. 282. Lead chrome yellow.
B.S.S. No. 389. Zinc chrome yellow.
B.S.S. No. 318. Chrome oxide greens.
B.S.S. No. 383. Prussian blue.
B.S.S. No. 314. Ultramarine blue.
B.S.S. No. 333. Lake red pigments.
B.S.S. No. 320. Mercury vermillion.
B.S.S. No. 987. Camouflage paints.
B.S.S. No. 1011. Red lead paints.
B.S.S. No. 1033. Priming paints lead-base (Types 1, 2, 3, 4 and 5).
B.S.S. No. 1053. Water paints and distempers.
B.S.S. No. 1056. Painting of buildings in wartime.
B.S.S. No. 1057. Substitute paints for exterior finishing.

War Emergency Paints.

B.S.S. No. 925. Paint oils.
B.S.S. No. 926. Extenders for paints.
B.S.S. No. 927. Alternatives for lead and zinc chrome yellows.
B.S.S. No. 928. Alternatives for chrome greens.
B.S.S. No. 929. Ready-mixed paints, priming, undercoating and finishing.

In addition to the above specifications there are, of course, many others which cover the ingredients, *e.g.* Prussian Blue No. 283; red lakes, No. 333; vegetable black, No. 286; red oxide of iron, dry, Nos. 272, 305 and 694. The index of the B.S.I. Handbook should be consulted for fuller details.

VARNISHES (OIL AND SPIRIT).

Of all the painting materials used by the engineer, varnish can be the most troublesome.

For decorative and protective purposes the B.S.I. specifications cover three interior types, B.S.S. 256—1936, types 1, 2, 3; and three exterior types, B.S.S. 257—1936, types 1, 2, 3. The latter may be used for other purposes, *e.g.* transport vehicles, engines, machinery, etc., in conjunction with flatting varnish B.S.S. 258—1936, and extra hard drying varnish B.S.S. 274—1936. For special purposes it is best to buy correctly prepared products which comprise heat-resisting, oil, petrol, and benzol-proof; alkali and acid-proof, suds-proof, gas-proof, di-electric, anti-rust and moisture-proof varnishes, baking varnishes, etc., cements and gold sizes (B.S.S. 311—1936).

The gums and resins may be of natural origin or produced synthetically or a combination of the two in conjunction with drying oils, solvents and plasticisers and chemical compounds.

Synthetic and artificial resins have reached a high state of perfection, ensure reliable results and effect great economies in production times. The insulating varnishes made from the phenol-

cresyl formaldehydes and the glycerine-phthallic anhydride resins and their compounds are used in enormous quantities in the engineering trades.

Spirit varnishes are made from seed lac, shellac, garnet lac, bleached and refined lacs, natural and synthetic resins and plasticisers and balsams with the addition of rosin, fatty oils (castor oil), drying oils, etc. These are dissolved in methylated spirits, butyl and amyl alcohol, naphtha, benzol, acetone and other organic solvents with dyes and pigments. Combinations of these mixtures can also be made with nitro-cellulose, celluloid and other cellulose compounds to produce special types of lacquers.

'Hot' lacquers contain methylated spirits as the main solvent, whilst 'cold' lacquers are made by substituting higher boiling solvents for part of the methylated spirit to ensure freedom from moisture absorption ('water-blush') during drying.

DISTEMPERS, WATER PAINTS, EMULSION PAINTS.

Aqueous solutions of glue, casein, Irish moss, water-soluble gums and resins, with soaps made from drying oils, drying oils and gum resins in the form of stable emulsions are the bases of these paints.

The cheapest consist of glue-size and whiting made in jelly form and treated with preservatives to prevent decomposition. The best contain little, if any, glue or casein, consisting mainly of lithopone, earth and lake colours, drying oils and resins in an aqueous emulsion.

Modern types have great covering power and resistance to wear; are very stable and consistent in quality. The decorative effects are of a very high order, especially when applied by spraying.

CELLULOID AND CELLULOSE SOLUTIONS REGULATIONS.

Under 'Cellulose Solutions, S.R. & O. 990 (1934)' and 'Cellulose Regulations 980 (1925)' the use, storage and manipulation of celluloid in solid and liquid form and of cellulose solutions in clear or pigmented condition are placed under strict control and inspection by the Home Office Factory Department.

The storage of the raw materials and finished solutions or enamels, the processes of production, the application of the enamels and lacquers to industrial materials in factories or workshops and the maintenance and repair of the buildings and plant in which the production, use, manipulation or application of these solutions and enamels are carried on, including the provision and maintenance of the necessary exits from the spraying, painting, mixing and storage places and rooms and the ventilation, heating, lighting and cleaning of such places are all governed by these Statutory Orders and Regulations. A copy of the Regulations must be kept posted in the proper places and reference should be made to the Factory Inspector and to the Memorandum on the use of Cellulose Solutions issued by the Home Office. Suppliers and manufacturers of cellulose products will assist with the necessary information where possible to ensure that the users will conform to the Regulations laid down. It must also be borne in mind that as cellulose lacquers usually contain petroleum mixtures having flash points below 73° F. the storage, use and transport of them is governed by the Petroleum Acts and that the local Petroleum Inspector has the right of entry and examination under the powers conferred upon him by statute. The Petroleum Acts and the Regulations issued thereon should therefore be consulted as well as the Cellulose Regulations and Factory Acts. The painting of buildings (including the internal decoration of cinemas, restaurants, etc.), is expressly excluded from the operations of the Cellulose Solution S.R. & O. 990 (1934).

LEAD PAINT REGULATIONS.

The following Rules and Orders and Regulations are at present in force under the Factory Acts and Lead Paints Act:—

Paints and Colours, Manufacture of, 945
Lead compounds, Manufacture of, 979, 996, 1713, 1714.
Painting of Buildings, 995 & S.R. & O. 847.
Painting of Vehicles, 994.

These regulate the rubbing down of dry paint films, the spraying of paints, the use of overalls, washing appliances, medical examinations and employment of women and young persons, dust extraction, etc., and should be consulted for details as failure to comply with them involve penalties on employers and employees.

PAINT ECONOMY MEMORANDUM.

The Interim Economy Memorandum issued by the Ministry of Works in September 1946, has now been reviewed in the light of present day conditions. Whilst raw material shortages are not so acute as in 1946, linseed oil supplies involve heavy expenditure of hard currency and this makes it necessary to continue to economise, wherever possible, in its use. The Schedule in the previous Economy Memorandum has been withdrawn and the following recommendations issued in its place:

1. Paints containing linseed oil should not be used in maintenance work for repainting surfaces which, by cleaning or other reasonable treatment, can be restored to a condition appropriate to the purpose for which the room or building is about to be used.

2. Paints, or other treatments containing linseed oil should not be used on exterior surfaces, previously unpainted, of stone, brick, clay-block, concrete cement rendering or stucco.

3. Paints, or other treatments, containing linseed oil should not be used when re-painting surfaces previously treated with cement paints, silicate paints, bituminous paints or wood preservatives.

4. Bituminous or tar paints should be used wherever practicable on exterior iron and steel such as gates, railings, outbuildings, etc.

5. Wood preservatives, in place of paint, should be used wherever practicable on exterior woodwork such as weatherboards, bargeboards, gutterboards, outbuildings, etc.

6. Encouragement should be given to the use of stain and varnish, in place of paint, wherever practicable on interior woodwork such as doors, cupboards, architraves, picture rails, skirtings and staircases.

7. Cement paints, or silicate paints should be used where suitable, according to the type and purpose of the building, on interior surfaces of brick, concrete, cement rendering, asbestos-cement, lime plaster and insulating board.

Linseed and other Drying Oils.

Linseed oil has been used as a paint medium for thousands of years. Its preparation and uses have been the objects of generations of research.

Intensive study of its technology during recent years has resulted in improved paints, varnishes and enamels. The raw linseed oil of commerce, containing mucilage, foots, water, free acid and albuminous materials, is not fit for the preparation of high quality paints, etc., until it has undergone prolonged tanking to settle out some impurities and render it mature, or has been subjected to one of the several refining processes (cracking by acid or alkali, de-sliming by siliceous earths, neutralising by glycerine) or treatment by heat alone. Clarification by filter-pressing, centrifuging or tanking followed by washing and neutralising result in 'refined' oil. Linseed oil is extracted from seed produced in Western India (Bombay oil), Eastern India (Calcutta oil), Argentina (Plate oil), Finland and Esthonia (Baltic oil). Baltic oil of the 1913 type was considered the acme of quality for best quality varnishes and enamels, but the quality depreciated badly following the Russian revolution and has not yet regained its former reputation. Calcutta oil is the most widely used for varnish, enamel and high grade paints. Plate oil is mainly used for cheap paints and white enamels as it bleaches easily.

Linseed oils are subjected to heat treatments to produce 'Boiled' and 'Double Boiled' linseeds on combination with rosinates, linoleates, naphthenates, oxides, acetates and hydrates of lead, cobalt, zinc, magnesium and manganese.

Polymerised linseed oils (stand oils or litho oils) are produced by heating the specially prepared oils at 550–600° F. for 4–24 hrs. or for 20–60 mins. at 700° F. in special plants. These oils are thick and heavy in body, dry slowly but impart durability and elasticity to varnishes, etc. Linseed oils can be thickened by heating and blowing with air or oxygen or chemical treatment to yield oil of special interest in spraying, plastic, linoleum-wet-on-wet, and wall finishes. The linseed oils are used in many forms either alone or in conjunction with Chinese wood oil (Tung oil, B.S.S. 391—1929), soya bean oil, fish oils, perilla oil, castor oil, oiticica oil, and natural (fossil) resins, synthetic and artificial resins, pitches, driers, turpentine and white spirits or other solvents to produce varnishes, japans, enamels and gloss paints for hundreds of purposes.

The specifications for linseed oils are :—

Raw linseed oil . . .	**B.S.S. 243—1926.**
Refined linseed oil . .	**B.S.S. 242—1926.**
Boiled " " . .	**B.S.S. 259—1926.**

To economise in the use of linseed oil during war-time, its use was rigidly controlled for even essential war purposes and prohibited for many 'civilian' uses. Tung oil, Perilla oil, Oiticica oils and treated castor oils were reserved for highest priority work only. Mixtures of linseed oils with other drying oils, *e.g.* fish oils, vegetable oils (except those referred to above), aromatic petroleum extracts (liquids and solids) were provided for in Government Specifications, *e.g.* B.S. 925/1011/1056/1057. Linseed oil was also partly or wholly replaced in oil-bound water-paints, and bituminous-emulsion camouflage paints even for Government work.

To meet the demand for drying oils to replace Tung, Oiticica and Perilla oils necessary for durability and resistance to solvents, chemicals, etc., in war finishes very thorough investigations into methods of processing linseed and other oils were undertaken and new types were successfully produced for use with imported natural and home produced synthetic resins.

Turpentine, White Spirits, etc.

Best American (spirits of) turpentine is distilled from the gummy exudation of pine trees, leaving rosin behind in the stills. It should comply with B.S.S. 244, Type 1, and occasionally turpentine of Indian, Swedish, French and Spanish, Greek or Russian origin may pass this specification. In general these other turpentines may comply more with the specification for the extracted wood turpentine (B.S.S. 290—1929 (Type 2). For painting purposes, polishes, etc., all are suitable, but for varnishes and veterinary purposes, best American turps (B.S.S. 244—1926 (Type 1) should be asked for.

White Spirits (Turpentine Substitutes).—For diluting paints made with linseed oil, white spirit of good quality is frequently mentioned in B.S.I. and other specifications. It is not advisable to use white spirit alone in paints containing varnish, stand oil, boiled linseed oil, japans or gold size, as the solvent power is poor. Turpentine should be added to the white spirit to ensure good mixing. There are several types of white spirit, one of them, B.S.S. 245—1926 (Type 1), being generally satisfactory for ordinary work. White spirit of good quality should leave no stain on clean filter paper and no residual odour after 1 hour's drying at 60° F., besides complying with other details of the specification. For painting indoors it is always advisable to use best turps whenever possible.

Solvent Naphtha (*Coal Tar Naphtha*) is largely used for thinning bitumen solutions and paints, rubber solutions and compositions. Two types are in general use, viz. Low Flash (90/160), B.S.S. 479A, and High Flash (90/190), B.B.S. 479B. These solvents are important as they are essential for the correct working of many painting compositions. Owing to the changes taking place in the treatment of coal for gas-making, crude oil for oil distillation and the economics of natural bitumen production, the character of bituminous solutions and paints is undergoing important changes.

Cellulose, Lacquers and Enamels.

Cellulose Lacquers and Enamels.—Treatment of cellulose (cotton, paper, wood fibre, vegetable matter) by acids (nitro-sulphuric and acetic) yields cellulose nitrates and acetates which are compounded with camphor, resins, plasticisers, gums, oils, solvents and diluents, pigments and dyes to produce cellulose lacquers and enamels for a great variety of purposes. Types differ for exterior (building and transport vehicle) and interior (building, hardware, fancy goods, machinery, furniture and domestic articles). Application is usually by spraying and great speed of finishing is possible. Modern types show exceptional resistance to sunlight, hard weather, abrasion, chemical action, heat and moisture. An immense number of types is available in the most brilliant and durable of colours. The types of 'cotton' are high, medium and low viscosity in 'ester' and 'spirit' soluble grades, and these are dissolved in combinations of organic solvents (alcohols, esters, ketones, ethers) with aromatic hydrocarbon diluents (toluol, xylol, benzol) and aliphatic hydrocarbons (ligroin, petroleum spirit, etc.). Synthetic resins and plasticisers are used in the most satisfactory types of lacquers to secure adhesion and toughness combined with durability.

The cellulose ethers (ethyl, methyl and benzyl, amyl and propyl celluloses) have special uses in industrial finishes. Methyl cellulose is the basis of modern water-bound plastic and texture paints, distempers and sizes.

The majority of the solvents, diluents, plasticisers, etc., used in the manufacture of cellulose lacquers together with some of the pigments are covered by B.S.I., 'X' or D.T.D. specifications. Numbers can be obtained from the standard index.

Synthetic Resin Varnishes and Enamels.

From the research and practice devoted to the phenol-formaldehyde spirit-soluble 'true' synthetic resins in the early years of the twentieth century have sprung legions of synthetic resins available to paint, varnish, lacquer and enamel users. There are three main types: phenol-aldehyde, cresol-aldehyde and xylenol-aldehyde (oil and spirit soluble), glycerine phthallic-anhydride (glyptal-oil-modified and spirit soluble), urea-formaldehyde (spirit soluble), coumarone, petroleum, vinyl and acrylate resins. In the original types 'mixed phenols' (carbolic acid) was used in conjunction with formaldehyde. Progress in technical knowledge has added considerably to the types available and the variations in the resins and their combinations that can be produced. The phenolic oil-soluble types yield varnishes of extraordinary resistance to sunlight, heat, hard weather, sea-water acid, alkali, oil, petrol and abrasion. The spirit-soluble types yield lacquers of exceptional di-electric strength, couparone and petroleum resins being unaffected by dilute acids and alkalis, have special applications in industrial and building coating compositions. Vinyl and acrylate resins being water white of perfect transparency, and unaffected by ultra-violet rays, have special technical uses. 'Modifications,' usually by adding rosin and resinates, are frequent and cause many complications. From the 'glyptals' are produced interior and exterior varnishes and enamels of especial durability and gloss for quick air-drying and baking purposes whilst the 'urea' resins yield enamels and lacquers of exceptionally pale colour, polymerising in very short times (15 min.) at 200–300° F. to very hard, tough and heat-resisting films. The combinations of all three types have special uses and provide a great variety of coatings for insulating, decorative, chemical proof and hard wearing service. The addition of the proper oil-soluble types of synthetic resins to linseed oil paints, varnishes and enamels ensures improved durability, higher gloss, greater protective power and durability and increased resistance against moisture and sunlight with the addition of speedier drying properties.

Stoving Japans, Enamels and Varnishes.

Natural, synthetic or artificial resins combined with drying oils, plasticisers, solvents, pitches, asphaltums, etc., yield coating compositions that can be baked or stoved at 75–500° F. in from 5–200 mins. after application by spraying, brushing or dipping. The composition of the japan or enamel is adjusted to the method and speed of production used, and the price available. The

work should be cleaned free of grease, oil and rust by the use of solvent, boiling caustic or de-rusting solutions, painted, drained, baked and cooled. One-coat work is general, two- or three-coat is the best. Box ovens are being replaced by 'conveyor' or 'tunnel' ovens through which the work is passed by chain conveyor systems. Heat is applied by either closed steam pipes, Bunsen gas burners, gas, coke or oil-fired 'plenum' (hot air) systems, infra-red (gas elements or electric lamps) and electric radiant elements. Steam, gas, electric and oil fired 'plenum' systems are all successful in their special spheres. Short exposure at a high temperature is possible with the quick polymerising (urea-glyptal) types of resin vehicles, white enamels being successfully baked at 600° F. in 3 mins. to hard durable finishes in modern infra-red stoves. Temperatures of 500° F. for 15 mins. are usual for black hardware japans. Baking types of cellulose enamels and synthetic resin enamels are used in vast quantities in the finishing of motor cars, hardware, toys, cycle and motor parts and accessories, etc. Food cans are treated with stoving finishings that resist bending, abrasion, weak organic acids, alkalis, sulphur compounds, sugar, superheated steam and weak alcohol and salt solutions. 'True' synthetic resins are expensive materials which provide distinct advantages over ordinary stoving enamels in drying time, finish, gloss, colour and resistance to modern conditions of hard wear, weather, etc. Unfortunately the temptation to substitute cheaper and less efficient materials at slightly less than the 'true' synthetic prices has resulted in dissatisfaction and suspicion of 'synthetics' generally. Careful tests will prove the economy and value of the highest grades.

Paint Shades and Colour Tints.

About eighty standard tints are recorded in the B.S.I. 381C specification. In addition the colour standard catalogues of the British Colour Council and Colour Users Association should be consulted for others.

These standard colour tints are a great help to both maker and user as colour names are to some extent brought into line for the first time.

Insulating Varnishes.

Quick air-drying types are made with spirit-soluble resins of high grade, *e.g.* refined shellacs, fossil resins, natural, synthetic and polymerised resins, pitches, cellulose acetate, with castor oil, Venice turps, synthetic plasticisers and the usual range of alcohols, ketones and hydrocarbons. These are also referred to sometimes as 'anti-sulphuric' enamels and are used for accumulators and battery room fittings. These varnishes are often lightly baked and can be applied to metal, fabric, cotton covering, paper, glass, vulcanite, etc.

Slow air-drying or baking types are prepared from 'phenol'-formaldehyde bases, glyptals or 'urea' resins either alone or combined. Natural fossil resins are chemically treated and combined with linseed, wood, perilla and fish oils or the oil-soluble synthetic resin varnishes to produce pale insulating varnishes.

Bitumens, asphaltums, pitches, etc., are added with or without waxes to produce the black insulating and impregnating types.

The vacuum-immersion type of application is essential for best results. Baking times vary from ½ hr. at 250° F. to 12 hrs. at 120° F. according to the varnish used. The B.S.I. Specifications are No. 514 (baking bitumen type) and No. 119 for clear baking oil varnish, for the varnished paper and tubes B.S.S. 316 and for varnished cloth or tape B.S.S. 419.

Chemical Resistant Enamels.

Natural resins, gums and oils are readily attacked by many chemicals, etc., and need reinforcement by other materials for improvement to resist even weak chemicals. The processes involved in the manufacture of synthetic and artificial resins pointed out the directions in which more satisfactory results could be found and the combination of these resins, etc., with polymerised oils (wood oil, castor oil, perilla oil) waxes, rubber and bitumens has yielded a long range of products giving satisfactory results. Modern finishes include air-dry and baking types of petrol and benzol proof enamels and varnishes; oil and grease proof enamels and varnishes; acid and alkali proof enamels and varnishes; sulphur-vapour proof enamels and varnishes; poison-gas proof enamels and varnishes; sea and soapy water proof enamels and varnishes; steam and moisture proof enamels and varnishes; ultra-violet proof enamels and varnishes; fungus proof enamels and varnishes; alcohol and solvent proof enamels and varnishes; heat and frost proof enamels and varnishes. They provide efficient protection in a large variety of circumstances for which ordinary types of paints, varnishes and enamels would be quite unsatisfactory.

Rust-Removers.

Pickling in solutions of sulphuric acid, oil of vitriol (hot solutions), hydrochloric acid (muriatic acid or spirits of salts), hot or cold, is the process generally used in large-scale practice. A water-swill followed by a neutralising wash follows and then drying or further treatment is given.

For delicate work *in situ* or for small rust spots on large surfaces, the use of phosphoric acid base washes has become general. There are many types covering varying concentrations of acid, different solvents, metallic salts and acids and colouring matters. Some need washing off with

water or solvent after the rust is dissolved, others can be left as they dry. Patents cover the best-known types.

These rust-removers or rust solvents are essential in modern painting practice on steel surfaces (*e.g.* motor bodies, steel furniture, car wheels and accessories and hardware), as they are speedy and quite safe.

Rust-Preventers.

After cleaning the surface of the steel, a rust-preventer should be applied at once. It may be oil, grease, petroleum jelly, varnish, paint or slushing glaze. There are many proprietary materials of great merit. Compounds of wool grease (lanoline) have been marketed for many years and can be obtained in a full range of drying and non-drying compositions. Specifications issued by Government departments and the National Physical Laboratory govern the properties of the wool grease and solvents used.

Paint Removers.

The saponifying power of caustic potash or soda and lime was the basis of the old type of paint remover and for rough work a mixture of soft soap, slaked lime and washing soda applied in paste form is effective.

The powerful solvents available in the cellulose industry form the basis of modern paint removers. Combinations of solvents exert more powerful action than single solvents. High volatility renders them ineffective, so slow drying solvents and retarders are used to keep the solvents in contact with the work long enough to be effective. The solvents used are acetone, ketones, benzol, naphtha, naphthalene, methylated spirit, di-chlor-ethylene, methylene chloride methyl alcohol, etc. The retarders are either lime or soda soaps, gums or resins, rubber, cellulose acetate, glucose, paraffin wax, stearine or glue with ammonia or acetic acid. The best types will strip stoved enamel in 5 mins. per coat and ordinary paint or cellulose 'at sight.'

It is important to follow strictly the makers' instructions about removal of the old paint and treatment of the stripped surface before repainting. Caustic alkalis, lime, wax and grease or spirit-soluble compounds may damage the new paint if left in cracks or crevices of the surface.

Covering Powers of Paints.

Specific Gravities of Paints.

The following tables are reproduced by permission of Thomas Howse, Ltd. The figures marked with an asterisk indicate paints which cover solid and completely obliterate the surface at the spreading rate given.

Greens.

Material.	Sq. ft. per lb.	Sq. ft. per gall.
Engineers' Green Oil paint	*100	*1700
Olive green gloss paint	*64	*1200
Middle green marine gloss paint	*115	*1500
O.W. pale green gloss paint	*95	*1400
Middle green engine enamel	*73	*1600
Pale green engineers' outside finishing gloss paint	*80	*1500
Decorators' undercoat green house paint	*55	*700
Decorators' finishing house paint	*72	*1300

Whites.

Material.	Sq. ft. per lb.	Sq. ft. per gall.	
R.M. white lead carbonate (stack)	53½	1500	
Basic white lead carbonate R.M., 1st coat	—	520	of 28 lbs. on new wood.
" " " " " 2nd "	—	850	
" " " " " 3rd "	—	*970	
" " " Sulphate	35·5	1000	
Zinc oxide, Red Seal	37	1000	
" sulphide lithopone	78½	1500	
" oxide priming paint 1550—on wood	37·5	600	
" " " " 1550—on metal	68·5	1095	
Genuine white lead oil paint B.S.S. 261	45	1100	
White Gloss zinc oxide oil paint B.S.S. 277	52	1010	
Antimony oxide extra	*60	*1000	
Titanium oxide extra	*80	*1200	
" " 25 per cent.	68	1200	
L.T. 1551 finishing zinc oxide Varnish paint	60	825	
Undercoat white zinc base paint	*56	*990	
8957 matt opaque white	*75	*1500	
Gloss white house paint	58	1000	
Interior gloss white enamel	*42½	*600	
Heavy brushing priming white	13	360	
4/4/2 zinc lithopone whiting paint	63	1050	

Whites (cont.).

Material.	Sq. ft. per lb.	Sq. ft. per gall.
Zinc oxide 6, lithopone 4, R.M.	•66	•1250
4 Titanox 25s. 1 white lead carbonate oil paint	•75	•1280
" " 1 zinc oxide oil paint	•64	•1080
Gloss white outside varnish paint	56	1200
Undercoating hard gloss white	•42	•950
Window sash white paint	•36	•600
High gloss tramcar saloon enamel	56	900
Zinc oxide 1, basic lead sulphate and carbonate 1	54	1100
Cellulose white lacquer L type	•65	•725
" " " Z "	50	653
" " " T "	•62	•714
Yellows.		
Pale French ochre	50	1100
Zinc chromate	53½	—
Primrose lead chromate	•65	—
Middle lemon lead chromate	•69	—
Deep lead chromate	•70	—
" ochre	•70	900
Aluminium.		
Superfine lining (powder with medium)	93·3	1115
Polished brushing (powder with medium)	68	820
Blacks.		
Black bitumen solution	•102	•1032
Bitumen solution	158	1480
Stoving black japan	80	753
Black oxide of iron	•110	—
" " " "	•90	—
" sign post enamel finishing	•145	•1800
" " " " undercoating	•95	•1600
Decorators' house gloss paint	•85	•1400
B.S.S. 294 black oil paint, type V	70	1280
Air drying dull hardware Black (brushed)	•170	•1640
" " " " " (sprayed)	•153	•1410
Vegetable black super quality (in oil)	•208	•1905
Drop black (in oil) super quality	•93	—
Wagon black paint R.M.	•67	•790
Cellulose spraying lacquer, gloss	•74	•680
" " " dull	•92	•850
Blues.		
Pure Prussian blue	114–287	—
Brunswick blue	110	—
Celestial blue	60	—
Finishing blue hard gloss paint	•56	•700
Royal blue gloss enamel	46	600
Quick-drying blue varnish paint	•32	•520
Oxides of Iron.		
96 per cent. Indian red	•100	—
85 " " oxide of iron (middle red)	•100	•1400
75 " " bright red oxide	•90	—
50 " " " " "	•65	—
Venetian red (bright) 25 per cent. oxide	•45	—
Purple brown, 97 per cent.	•110	—
" " 92 " "	•53	—
" " 90 " "	•57	—
Turkey red (bright shade)	•130	—
Gasholder chocolate	•74	•1380
Brown steel priming paint 45135	•75	•1500
Engineers' oxide of iron oil paint	•96	•1600
" shop oxide compo.	•42	•600
B.S.S. 295 Grade A, type L, red oxide paint	•60	•1275
" " " " H, Red	•70	•1250
Ships, bottom paint sprayed 1st anti-corrosive	—	•470
Heavy body gloss paint, 2nd anti-fouling	—	•800
Implement enamel (gloss) 1 coating	•102	•1450

Water Paints.

Material.	Sq. ft. per lb.	Sq. ft. per gall.
Cream distemper oil-bound 1 lb. paste (thinned with water)	48	1027
Cottage distemper (1 lb. thinned with water)	42	784
Ceiling white paste (1 lb. thinned with water)	40	880
White distemper (oil-bound) (1 lb. paste thinned with water)	*67	*1460
Deep red oil-bound water paint (1 lb. paste thinned with water)	*90	*1680
Middle green oil-bound water paint (1 lb. paste thinned with water)	37½	895

Varnishes and Oils.

Exterior oil varnish B.S.S. 257, type 2	103·5	984
Seaplane varnish	135	1284
Finishing body varnish (flowing)	112	1080
Floor varnish	100	950
Boiled linseed B.S.S. 259 oil on iron	130	1210
" " " " " " wood	54	505
Raw linseed oil on iron	140	1300
" " " " wood	85	790
Clear spraying cellulose lacquer	70	620
" brushing cellulose lacquer	84	775
Common copal oak varnish	72	690
Hard church oak varnish	94·5	875
Thin dipping metal priming varnish	165	1800
Black (Brunswick) japan	100	940
Cement wall (priming) varnish	—	470
Van or carriage varnish	90	840
Shellac varnish on wood	—	450
Varnish stain (spirit) on wood	—	500

Greys.

Graphite (American)	*65	—
" (Mexican)	*90	—
" (Italian)	*114	—
" (German)	*35	—
Ferro-graphite (German)	54	—
" (English)	70	—
Grey Enamel zinc oxide base	*78	*1554
" oil paint zinc oxide base	*192	*1935
" gloss paint (outside) T.	87	1200
" factory gloss paint	*70	1240
" steelwork paint finishing	*83	1482
" " " priming	*50	1100
" tramcar roof enamel	*62	900
" engineers' shop paint	*48	875
" ironfounders' gloss paint	*37·5	600
" ships' boot topping sprayed 1st coat (900 sq. ft. per coat)		*1150
sprayed 2nd coat	—	*1230
" (dark) tramway pole enamel (gloss)	*95	1200

Reds.

Genuine red lead oil paint on wood (B.S.S. 217B)	*25	—
B.S.S. 217 Grade A linseed oil red lead paint on wood	*43	*1400
Genuine red lead oil paint on iron (B.S.S. 217A)	*48	1400
Non-setting red lead oil paint on iron (B.S.S. 315)	*50	*1400
R.M. red lead ships' anti-corrosive (sprayed)	—	*800
Orange lead	*52	*1420
Leadless pink priming on steel	*85	*1500
Red and white lead pink priming on steel	*56	*1200
Matt pillar box scarlet japan	*57	*1100
Gloss scarlet enamel	*55	*900
Genuine vermilion japan	*77	*1880
Deep orange yellow enamel	*36	650

Cream, Stone and Buff.

Material.	Sq. ft. per lb.	Sq. ft. per gall.
Deep cream gloss enamel (zinc base)	*70	1000
Stone colour lead-carbonate lead-sulphate zinc oxide, equal parts	62	2831
50/50 white lead-carbonate and zinc oxide R.M. oil paint	66	1231
Dark stone finishing outside gloss paint	*55	*1150
B.E.S.A. specification 262 white lead gloss oil paint type 2, buff	*45	*1215
Stone colour gloss oil paint (interior)	*51	*1007
Ivory white finishing gloss house paint	50	740
'Graphax' marine cream gloss R.M.	*68	1300

Browns.

Best Turkey umber	*90	—
Seconds Turkey umber	72	—
Thirds Turkey umber	37	—
Burnt Sienna (super)	44	—
Raw umber best	60	—
Purple brown (oxide) deep shade	*60	—
Brown magnetic-oxide of iron	*64	—
Red-brown ochre	*38	—
Middle brown inside gloss paint	*71	*900
Light brown outside gloss paint	*56½	*1000
Brown housing gloss paint	*87	*1200
Teak brown inside enamel gloss	*105	*1500
U/coating brown japan	*70	*1300
Purple brown gloss paint	*74	*950
Brown oil paint, K1294 (outside)	*72	*1240
Inside fawn colour u/coating	*48	*1100
" " " finishing	*57	*1060
Deep Indian red gloss tinware enamel	*112	*1650
" maroon coach enamel (finishing)	40	650

Metallic Paints.

Metal-powders (viz. aluminium, copper, brass, gun-metal, etc., are mixed with drying vehicles and applied to metal, wood, paper, fabric, etc., surfaces by spraying or brushing, etc. They may be baked or air-dried. The vehicles may be oil-varnishes, cellulose, lacquers, etc., of special types. Many types are prepared differing in their use. Some can be mixed with the metal powder and remain unaffected for weeks, others must be used immediately or the powder is attacked and loses its brilliance.

Aluminium Paints.

For resisting heat the synthetic resin types are satisfactory. For pipelines, machinery, factory plant and equipment, vehicles and hardware, street lamps, etc., the types vary considerably.

The best qualities weigh about 10 to 12 lb. per gallon, cover about 1,000 sq. ft. per gallon (sprayed) per coat. Drying times are from 5 mins. to 12 hours at 60° F.

Copper and gold bronzes require special mediums if they are to be kept in liquid form for even a few hours. They tarnish rapidly on exposure and do not resist temperatures as high as those at which aluminium is stable.

Particular care should be taken to avoid mixing metal powders with linseed oils, turpentine, copal varnishes or driers.

Painting Galvanised Iron.

The pickling processes used in preparing the sheet iron before galvanising result in the formation of corrosive zinc salts (chlorides and sulphates) on the spelter surface. Removal by washing or rain (weathering) after erection is the cheapest method and as safe as any.

Treatment of sheets that cannot be weathered (*e.g.* sheet-metal plant) must therefore include some neutralising process.

Washing with a mixture of hot water and methylated spirit is good—drying is speedy and natural as well as effective. The addition of phosphoric acid to the wash is a decided advantage. The best results are obtained by the use of the rust-removers used on rusty steel (mentioned in this section). When diluted with 5 to 10 times their volume of cold water they provide a splendid etching, cleaning and neutralising solvent for the zinc salts. Water soluble synthetic resins of the glycerine-phthallic-anhydride-urea type which are susceptible to the catalytic action of zinc salts are said to form hard and impervious coatings on zinc surfaces.

Good quality red oxide or bitumen paints or the well-known brands of steel primers are satisfactory paints for galvanised sheeting. They should be repainted at least once in every five years. The use of finely divided metals as pigments in liquid coatings has much to recommend it.

Paints for Cement, Concrete, Plaster and Asbestos Cement Surfaces.

Free lime is a necessary evil in the formation of the surface; disposal of it, or, rendering it inert within the texture of the surface is essential if the coating composition is to be permanent.

The formation of water-insoluble lime compounds is an attractive method of treatment and when aluminium and zinc salts (sulphates, fluorides) can be used there is a good chance of success. There is no hard and fast rule—each case can be dealt with on its merits, but a safe preparatory treatment for plaster consists of washing down with a 4 oz. per gallon solution of ammonia-alum (it must *not* be potash alum) repeating at 4 hours, drying off and dusting down after 24 hours. Cement, concrete and asbestos cement require stronger solutions. The preparatory sealer coat is of vital importance and too much care cannot be given to selecting a satisfactory make and following instructions fully. Combinations of unsaponifiable oils, lime hardening resins and lime resistant pigments are essential in a successful cement or plaster paint. Avoid glue or casein-bound waterpaints containing alkaline carbonates or borates and on no account use linseed oil paints as a first coat. The perfect treatment consists of carefully applying two full coats of the preparatory sealer, giving 16 hours between each, then applying a body colour and finishing with a coloured cement glaze, a sand finish, imitation stone, marble finish, or plastic textured or stipple surface, colouring and decorating as may be required.

New asbestos cement sheets and raw concrete walls treated by this process will yield satisfactory results.

For aged and weathered cement surfaces wash down with clean water, proceed with one coat of preparatory sealer and then one body colour and one finishing coat.

Dr. Angus Smith's Solution.

Strictly speaking, there has never been and cannot be a Dr. Angus Smith's solution. His patent applied to the method of dipping hot iron castings, pipes, columns, ironwork, etc., in a mixture of coal tar, pitch and tallow.

The use of the mixture has been abandoned for many years, and plastic bitumens have taken its place. Where it is necessary to use cold-drying solutions the plastic bitumens are dissolved in naphtha and boiled oil and applied by spraying, brushing or dipping. They dry in 1½ to 6 hours and are cheap and durable.

Graphite Paints.

For steelwork the inert properties of silico-carbon (graphite) to attacks by moisture, gases, acids and alkalis appear to offer advantages in protection against corrosion. The electro-positive cathodic quality of carbon (best graphites range as high as 95 per cent. carbon) to iron and steel, however, accelerates corrosion and condemns the use of carbon pigments as steel primers. For second and third coats over really inhibitive pigments such as red lead, basic chromate, zinc chromate, blue lead, etc., graphites are excellent materials if mixed in proper media. Silica-graphites, although rough in texture and poor in opacity, give best results when of high purity. The carbonaceous silica graphites should not be confused with the ferro-silicates (micaceous iron ores) compounds found in Devon (England), Wales, Germany and Sweden. These latter are of special interest as they afford excellent protection to steel when carefully refined and mixed with proper water-resisting media although they are very similar in appearance to the silica graphites which are not satisfactory in contact with iron. The ferro-silicates contain as high as 95 per cent. of iron oxide chemically combined with silica, etc. These ferro-silicates are mixed with zinc and iron oxides, white leads, green and black pigments, aluminium, etc., to make a wide range of colours for the protection of steelwork against the attacks of sulphur, acids, alkalis and sea water. They are marketed under well-known brands. They weigh about 18 lbs. per gallon (against graphite 12½ lbs.), cover 70 sq. ft. per lb. (graphite 100–110 sq. ft.). In actual exposure tests of many types of carbonaceous graphite and ferro-silicate graphites the superior durability of the ferro-silicates as steel primers is very marked being only surpassed by the lead pigments. The paints are low in cost and easily applied.

Wood Preservatives and Stains.

Treatment of timber involves the use of anti-fungus and bactericidal compounds in solvents to permit of deep penetration and ease of application. Among the fungicides are coal tar, wood tar, etc., dissolved in creosote or naphtha, creosote, wood-oil varnishes, paints and enamels, carbolic acid compounds, with alkali salts, sulphur derivatives and metallic soaps containing essential oils and volatile insecticides. In recent years the use of metallic compounds, *i.e.*, copper, iron, lead, nickel, arsenic, manganese, aluminium, magnesium, etc., with creosote or oleic acids and naphthenic acids have given satisfactory results. The copper compounds are green, those of iron and manganese are brown, whilst the transparent types are made with lead, magnesium. zinc or aluminium, being dissolved in coal tar, naphtha or white spirit. They are applied by brushing, dipping or spraying. For protection against dry rot and the fungi usually found in the woodwork of buildings, the copper compounds are very effective. To prevent decay and damage by insects, *e.g.*, the death watch beetle, special sprays are prepared from essential oils, cyclo-hexanone compounds, solvents and waxes.

The use of waxes, varnishes, polishes and stains has grown tremendously during recent years. Ceresin, carnauba, japan, montan and paraffin waxes, natural resins and drying oils in solvents mixed with bitumens and asphaltums or dyes and colours provide the range of products available. They are applied by brushing, rubbing or mechanical means. Waxes are now emulsified into colloidal liquids without the use of alkaline soaps and produce glossy and hard-wearing coatings that are antiseptic, durable and efficient.

LUMINOUS PAINTS.

Compounds of calcium, zinc and sulphur are well known for their property of retaining and reflecting light rays in the dark. They are surpassed however by the rare earth compounds, *e.g.*, radio-thorium, thorium bromide, mesothorium, etc., which are used on optical instruments, watches, gun sights, binoculars and range finders, etc., etc.

They are among the most expensive paint materials known.

NOVELTY ' FINISHES.

Research work into the causes of the cracking of paint and varnish films led to the production of means of controlling or producing the cracking, crawling or rivelling at will. The results are available as ' Crackle,' ' Ripple,' ' Frosting,' ' Crystallising,' ' Mottle,' ' Leather-grain,' ' Spottle,' etc., Types of finishes which find large application on domestic hardware, stoves, heaters, fancy goods, etc. They involve the use of specially prepared compositions of oils, resins, solvents, celluloses and chemicals and are applied by spraying with baking or air drying treatment. A very wide range of colours and finishes is available for metals, wood, glass, fabric, etc. Individual methods of working produce varied effects and colourings. Air drying and stoving processes are worked.

FINISHES FOR POLISHED METALS.

Modern metal furnishing processes involve the use of coloured glazes, or enamels applied to highly polished or polishable surfaces. Special types of enamels and lacquers are available. They may be applied successfully to the plated surfaces or may be used to treat unplated surfaces, passing through the plating bath and are polished to a high gloss with the metal plating itself.

WET-ON-WET PAINTS.

For outdoor painting conditions in which rapid drying is required of paints and compositions that have to resist weathering, heat, sea water, chemical fumes and hard-wear, the ordinary types of linseed-oil and varnish paints are quite useless, for only one durable coat can be dried in 12 to 16 hours and it is insufficient to provide adequate protection against severe and prolonged service. Successful use has been made of synthetic resin enamels and treated linseed and other drying oils which gelate quickly when applied as solutions in volatile solvents after the solvents have dried off. Owing to their peculiar properties they permit of paints being applied to painted surfaces that are still tacky. Many types are available, and decorative or anticorrosive pigments can be incorporated with the most efficient of them. Application can be made by brush or spray, brushing permits of 2 or 3 coats a day and spraying 3 to 5 coats a day, the brushed coats requiring a harder surface that takes longer to ' set off.'

The final coatings are very elastic, tough and weather resistant if made from correctly treated oils, resins and pigments.

Difficulties occur however, if the medium contains free mineral acids, sulphur compounds chlorides, aluminium, soaps or lime compounds when applied to metal or wood surfaces.

LUBRICANTS AND OILS.*

GREASES FOR RAILWAY AXLES.

(1.)

Water 1 gallon; palm oil, 6 lbs.; clean tallow, 3 lbs.; common soda, ½ lb.

(2.)

Tallow, 8 lbs.; palm oil, 10 lbs. Heat to 212° F., and well stir.

ANTIFRICTION GREASE.

(1.)

Tallow, 100 lbs.; palm oil, 70 lbs. Boil together, and when cooled to 80° F. strain through sieve and mix with 28 lbs. common soda and 1½ gallon of water.

(2.)

Blacklead, 1 part; lard, 4 parts.

*See also Section XX, Part II. (Vol. I.).

Rosin-Oil Soap.

90 lbs. powdered slaked lime, stirred into 100 lbs. rosin oil. Heat the mixture, constantly stirring it well until a uniform paste of the consistence of syrup is obtained. This rosin oil is a component of all the patent wagon greases.

Blue Patent Grease.

2 lbs. powdered slaked lime, heated in 550 lbs. of crude resin oil for 1 hour, and then allowed to cool. Skim the oil, and stir in 10 to 12 lbs. of rosin-oil soap, until the compound is of a butter consistency and of a blue colour.

Yellow Patent Grease.

1 part turpentine, 20 parts caustic lye. Add to 350 parts of blue grease.

Black Patent Grease.

2 parts lampblack, 110 parts blue grease.

Patent Palm-Oil Wagon Grease.

Melt 10 lbs. rosin-oil soap in 10 lbs. palm oil; add 550 lbs. rosin oil, with as much rosin-oil soap as will give the whole the consistence of butter; finally, add 7½ to 10 lbs. of caustic soda lye.

Adhesive Grease for Machine Belts.

Lubricate the belts with fish oil to which 10 per cent. of tallow has been added. This will make them flexible, and augment their friction on the pulleys.

Grease for Waterproofing Leather.

Oleic acid, 24 parts; crude stearic acid, 6 parts; ammoniacal soap, 18 parts; extract of tannin, 6 parts; water, 24 parts. Melt the oleic acid together with the stearic acid; then add gradually the ammoniacal soap, the extract of tannin, and finally the water. The ammoniacal soap is obtained by adding to heated oleic acid caustic ammonia until, after continued stirring, the odour of ammonia remains apparent and the whole congeals to a jelly-like mass.

Cooling Compound for Hot Bearings.

(1.)

Mercurial ointment mixed with black cylinder oil and applied every quarter of an hour, or as often as expedient. The following is also recommended as a good cooling compound for heavy bearings:—Tallow, 2 lbs.; plumbago, 6 ozs.; sugar of lead, 4 ozs. Melt the tallow with gentle heat and add the other ingredients, stirring until cold. For lubricating gearing, wooden cogs, etc., nothing better need be used than a thin mixture of soft soap and black lead.

(2.)

For cooling a hot bearing Professor Lecocq, of the Charleroi Higher Industrial School, recommends that the oil usually employed for lubrication be dissolved in an equal quantity of petroleum spirit, the evaporation of which latter will, he says, afford the cooling desired.

Watchmaker's Oil.

Expose the best olive oil to a temperature a few degrees below the freezing point, which will cause all foreign substances to separate. Pour off carefully the supernatant clear oil, and filter through a cup of linden wood or pith of elder wood. By this process an oil is obtained which will remain liquid for several years, and does not attack the delicate machinery.

Neat's-foot oil treated in the above manner furnishes a less useful oil, since it loses much fatty matter by exposure to cold.

A very useful oil is obtained by dissolving 1 part of pure neat's-foot oil in 3 of pure benzine. Allow the compound to remain for several days in a closed vessel; then filter, and expose the solution to a temperature of 40° F., at which it is again filtered, and the benzine distilled off. The oil should be kept in small, dark vials protected from the air.

A very fine lubricant for clocks and watches is, according to Artemus, obtained by mixing 2 parts of solar oil with 1 of rape-seed oil.

To Test the Fitness of Oils for Lubricating Watches and Clocks.

Pour a drop of the oil to be tested upon different metal plates, as iron, brass, tin, lead, etc.; keep them in a place free from dust, and examine the drops during 8 to 14 days in regard to their liquidity. Oil remaining liquid after the lapse of this time can be safely used.

VISCOSITY OF VARIOUS STANDARD OILS.

Description.	Viscosity.	
	At 70° F.	At 120° F.
Sperm oil	100	45
Whale oil	190	70
Olive oil	213	75
Lard oil	225	78
Neatsfoot oil	247	80
Rape-seed oil	250	88
Axle oil	600	120
Valve oil	—	550
Cylinder oil	—	700

In the above table sperm oil is taken as the standard for viscosity at 70° F.

The viscosity of thick oils can be greatly reduced by adding 8 per cent. of naphthalen to the oil.

An objectionable characteristic in an oil is its tendency to become resinous with use, due to the oil undergoing the process of oxidation. This change in the oil indicates, to some extent, the degree of refining process the oil has undergone. A simple test can be made in the following way :—Place a small quantity of oil in a glass beaker and put in a nearly equal amount of nitric acid, Should the oil have no tendency to gum, the acid and oil will combine and get very thick, while should there be any objectionable characteristic the oil will remain thin.

See also Descriptive Section, XLII.

Graphite Products, Ltd.

A. C. Wells & Co., Ltd.

SECTION XLIII

PATENTS, DESIGNS, AND TRADE MARKS

(Revised by G. W. Tookey, Barrister-at-Law.)

(Pp. 1001–1013)

SECTION XLIII

PATENTS, DESIGNS, AND TRADE MARKS.

(Revised by G. W. Tookey, Barrister-at-Law.)

[The references at the ends of the paragraphs are to the relevant sections of the Patents and Designs Acts, 1907–1946, and the Trade Marks Act, 1938.]

PATENTS.

(Patents and Designs Acts, 1907–1946.)

Acceptance.—Acceptance takes place after an application is in order which normally must be within eighteen months of the date of the application for the patent. This period for putting an application in order may be extended up to three months on payment of fees and may be further extended during the period in which an Appeal is open or pending. An application is deemed to be in order when the complete specification has been finally left after the Applicant has complied with all requirements. (8a.)

The Comptroller may postpone acceptance to a date not later than twenty-one months from the date of application if requested by the applicant.

Acceptance is advertised in the Official Journal (Patents) and specifications and samples (if any) are open to public inspection. (9.)

After acceptance the applicant has full rights under the patent but may not commence proceedings for infringement until the patent is actually sealed. (10.)

Addition, Patent of.—An applicant or patentee may apply for further patents in respect of improvements or modifications and request their grant for the unexpired term of the main patent. In such cases Patents of Addition are granted and no renewal fees are required in respect thereof. Independent patents can be converted into Patents of Addition and *vice versa.* (19.)

Amendment of Specification after grant or acceptance.—A Complete Specification and drawings may be amended by way of disclaimer correction or explanation on application to the Comptroller if good reasons are given. Opposition may be entered. An appeal lies to the Appeal Tribunal, if the patent has not yet been granted, and after grant an appeal lies to the Court.

No amendment is allowed which will make the specification claim an invention substantially larger than or different from that claimed before amendment.

If infringement or revocation proceedings are pending an application for leave to amend must be made to the Court and not to the Comptroller. (21, 22.)

Damages for infringement before amendment cannot be obtained unless the patentee establishes that his original claim was framed in good faith and with reasonable skill and knowledge. (23.)

Anticipation.—A patent is not anticipated by reason only of its publication in the United Kingdom in a specification relating to a patent over fifty years old, whether a British or foreign specification, or in any official abridgment or extract from such a specification, or in a provisional specification not followed by a complete specification.

Prior publication does not invalidate a patent if the patentee can prove that the publication was made without the knowledge and consent of the true inventor and that the published matter was obtained from the true inventor, and that the inventor applied for a patent as soon as possible after hearing of the publication. (41.)

Appeal Tribunal.—A High Court Judge is appointed as the Appeal Tribunal to hear appeals from the Comptroller on matters arising in the course of applications. (92a.)

Hearings are in private and Patent Agents have a right of audience.

Application for Patent.—An application (£1) can be made by any person whether a British subject or not claiming to be the true and first inventor, either alone or jointly with others, who need not be joint inventors. The application must be accompanied by either a provisional or complete specification. (1.)

Certificate of Validity.—In an infringement action the Court may certify that the validity of any claim came in question, and in any subsequent action, if successful, the plaintiff may receive full solicitor and client costs in respect of that claim. (35.)

Chemical Processes, Foods and Medicines.—In the case of inventions relating to substances prepared by chemical processes or intended for food or medicine, the specification must not include claims for the substance itself except when prepared by the methods or processes of manufacture particularly described and ascertained in the specification or by their obvious chemical equivalents. In the case of foods and medicines, a mere mixture simply aggregating the known properties of the ingredients is not patentable. (38a (1).)

In an infringement action relating to a patent for the production of a new substance, in the absence of proof to the contrary, any substance of the same chemical composition is deemed to have been produced by the patented process. (38a (2).)

The Comptroller will (unless he sees good reason to the contrary) grant compulsory licences in respect of patents for inventions for the production or preparation of food or medicine on terms which allow the public the product at the lowest possible price consistent with giving to the inventor due reward for the research leading to the invention. An appeal lies to the Court. (38a (3).)

Cognate Inventions.—Provisional specifications of the same applicant for inventions which are cognate or modifications one of the other may be combined into one complete specification on which a single patent is granted.

The patent bears the date of the earliest provisional specification but in considering validity and other questions under the Acts regard may be had to the individual dates of the provisional applications corresponding with the various claims of the complete specification. (16.)

Convention Applications.—Any person who has applied for protection for an invention in a Convention country is entitled to a U.K. patent having the same date as the date of application in the Convention country. Application must be made in U.K. within twelve months from date of application in the Convention country. (91.)

Convention countries are those declared to be such by Orders in Council. (91a.)

The U.K. application must be accompanied by a complete specification which will become open to inspection, together with the filed copy of the original application, at the expiration of eighteen months from the priority date claimed, whether or not the specification has by then been accepted. (91.)

Court.—Infringement and revocation proceedings and references to the Court under s. 29 are taken in the High Court (usually the Chancery Division) and may be heard by any judge.

References to ' the Court ' in sections of the Patent Acts dealing with appeals from the Comptroller and applications for extension of patents are to the special ' patent ' judge appointed by the Lord Chancellor. The special judge is usually different from the judge constituting the ' Appeal Tribunal.' Patent agents have a right of audience before the Appeal Tribunal but not before the Court.

Crown Rights.—Any Government Department or their authorised agents or contractors may make use of an invention for the services of the Crown on terms agreed with the patentee and approved by the Treasury, or settled by the Court which may appoint an arbitrator. (29 (1).) The Royal Commission on Awards to Inventors is specially empowered to deal with disputes between patentees and Government departments on this matter, provided both parties consent.

During the war period (at present extended till December 10, 1950, by Emergency Laws (Miscellaneous Provisions) Act, 1947), Governmental powers are extended to include the use of inventions for the wide purposes set out in Supplies and Services (Transitional Powers) Act, 1945.

The terms of any agreement or licence between the patentee and any other persons are inoperative so far as concerns the making, use or exercise of the invention for the service of the Crown. (29 (1).)

If an invention covered by a patent was recorded in a document or tried by a Government Department (not having been communicated by the patentee or applicant), before the date of the application for the patent, the invention can at any time be used by any Government Department or authorised agent or contractor without payment to the patentee. (29 (1.))

Crown rights cover the sale of articles no longer required for Crown service and also the sale of articles forfeited under the Customs and Excise laws. (29 (3) and (4).)

Date of Application.—The date of application is either the date of filing of an application or any date to which it may be ante- or post-dated (4).

Date of Patent.—All patents are normally dated as of the date of application, but no proceedings can be taken in respect of an infringement committed before acceptance of the complete specification (13), and proceedings cannot actually be instituted until the date of sealing of the Patent. (10.)

For cases in which patents are given a priority date see under ' Convention Applications ' and ' Fraudulent Applications.'

Effect, Extent and Form of Patent.—Patents are effective throughout the United Kingdom and the Isle of Man. Every patent shall be for one invention only but the specification may contain

more than one claim. No objection to a patent on the ground that it has been granted for more than one invention can be raised after grant. The United Kingdom includes Northern Ireland. (14.)

Employee.—Application for patents for inventions made by an employee must include the name of the employee as inventor, but the employer may be entitled to the whole beneficial interest in the patent depending on the terms of the employment. If the employer is so entitled, the employee will be required to assign his interest in the patent when granted.

Enemy Patents.—An application for a patent based on an invention made in Germany or Japan or by a German or Japanese national in any enemy territory between September 3, 1938, and December 31, 1945, will be refused. (P. & D. Act, 1946). As to revocation of such patents, if granted, see ' Revocation.'

Most German-owned U.K. patents granted before the war will have lapsed for non-payment of renewal fees, but some such patents will have been kept in force by British licensees whose interests are protected under emergency legislation and by international agreement.

Examination of Applications.—If the Examiner reports that the application is not properly prepared the application may be rejected until suitably amended and the application may be post-dated to the date of filing of a satisfactory amendment. On request by the applicant a complete specification may be treated as a provisional specification, to be followed by a complete specification in the usual manner. The applicant may before acceptance obtain post-dating of his application for a period up to six months, on payment of a fee. Notice of acceptance of an application is given to the applicant. Appeal from decision of Comptroller on these matters lies to the Appeal Tribunal. (3.)

Examination of Complete Specification.—If a complete specification is filed after a provisional and the Examiner reports that it is not properly prepared, the Comptroller may refuse to accept the specification until suitable amendments have been made. If the examiner reports that the invention described in the complete specification is not the same as that described in the provisional specification, the Comptroller may :

(*a*) refuse to accept the complete specification until satisfactorily amended ;
(*b*) (with consent of the applicant) cancel provisional specification and proceed with application on the basis of the date of leaving of the complete specification.

Any invention in a complete specification not found in the provisional specification may be removed and embodied in a separate new application which can be ante-dated to the date of leaving of the original complete specification.

Appeal lies to the Appeal Tribunal (6). See also ' Search.'

Examiner's Reports. Examiner's reports shall not be open to public inspection or production in legal proceedings except by direction of the Court. However, on payment of a fee the Comptroller may disclose references to documents and numbers of patent specifications cited as a result of the novelty search, in respect of complete specifications accepted or published. (68.)

Exhibition.—The exhibition and use of an invention at an industrial or international exhibition certified by the Board of Trade, or the use of the invention during the exhibition by any person elsewhere without the consent of the inventor, or the reading of a paper before a learned society or its publication in the society's transactions, does not invalidate a patent subsequently applied for, provided that notice is given to the Comptroller of intention to exhibit the invention or read the paper, and the application for a patent is made within six months from the opening of the exhibition or the reading of the paper. (45.)

Extension of Term on ground of inadequate remuneration.—Application for extension of the term of a patent on the ground of inadequate remuneration may be made by Petition to the High Court. If the Court considers that a patentee has been inadequately remunerated, having regard to the nature and merits of the invention in relation to the public, to the profits made by the patentee as such, and to all the circumstances of the case, the term of the patent may be extended by five years or in exceptional cases ten years, or the grant of a new patent may be ordered for a specific term and subject to any restrictions or conditions thought fit by the Court (18.) There is no appeal from the Court's decision.

Extension of Term on ground of War loss.—If a patentee not being an enemy subject or enemy controlled company has suffered loss or damage (including loss of opportunity of dealing in or developing his invention due to his being engaged in work of national importance connected with hostilities) an application may be made to the Court or at the option of the Patentee to the Comptroller for an extension of term of the patent on this ground alone. The aggregate extension under one or more applications shall not exceed ten years. A simple procedure is provided for applications made to the Comptroller but cases raising important issues may be referred by the Comptroller to the Court.

Appeal lies from decisions of the Comptroller to the Appeal Tribunal. There is no appeal from the decisions of the Court or the Appeal Tribunal. (18.)

Foods.—See ' Chemical Processes, etc.'

Fraudulent Applications.—A patent granted to the true and first inventor is not invalidated by an application in fraud of his rights or by use or publication of the invention after provisional

protection has been fraudulently obtained. If a patent is wholly or partially refused on opposition or revoked on the ground of fraud a patent for the whole or part of the invention may be granted to the true and first inventor in lieu of and bearing the same date as the refused application or revoked patent. (15.)

Grant of Patent.—A patent will be granted to an applicant, or to several applicants jointly, or to an assignee, on payment of the sealing fee. (12.) If lost or destroyed a duplicate patent may be issued. (44.)

Illegal Conditions of Sale and Licence of Patented Articles.—In contracts relating to the sale or lease of or licence to use or work any article or process protected by a patent conditions prohibiting the use of articles (whether patented or not) or patented processes supplied or owned by third parties or requiring the acquisition from the patentee or his nominee of unpatented articles are illegal unless the purchaser, lessee or licensee was offered at the time of the contract an alternative contract on reasonable terms but without the above conditions and the contract itself allows for three months' notice of termination on payment of compensation.

The foregoing provision does not affect conditions prohibiting persons from selling goods other than those of a particular person or reserving to the licensor the right to supply new parts for putting or keeping a patented article in repair.

The continuing existence of an illegal contract as above is a bar to obtaining relief in an infringement action. (38.)

Infringement Action.—In an infringement action a plaintiff may obtain an injunction, damages, delivery up of infringing articles and costs.

A defendant may deny that he has infringed the monopoly claimed in the patent specification and may attack the validity of the patent, wholly or in part. He may further by way of counterclaim seek revocation of the patent on the ground of invalidity. (32.)

The insertion of illegal conditions in licences or contracts is available as a defence while contract is in force. (38.) See 'Illegal conditions of Sale and Licence.'

If some claims are found valid and others invalid, the patentee may obtain limited relief and may be put on terms as to amendment of his patent. An order for revocation may be stayed to allow the patentee to amend on suitable terms. (32a.)

Infringement, Innocent.—No damages for infringement can be recovered from a defendant who proves he was not aware nor had reasonable means of making himself aware of the existence of the patent; the mere marking of articles 'patent,' 'patented,' or with other equivalent wording without the number of the patent, is not sufficient notice of the existence of the patent. Innocence, however, does not affect proceedings for an injunction. (33.)

Inventor.—An application for a patent other than a Convention Application must be made by the inventor himself either alone or jointly with other persons or by the legal representative of a deceased inventor. Failure to do so is a ground for revocation. (25, 43.)

The inventor of the whole or a substantial part of the invention has a right to be mentioned as such and may apply to the Comptroller to enforce his rights. The mention of an inventor as such does not confer or derogate from any rights under the patent. (11a.)

Joint Applicants.—Where a joint applicant has died, a patent may with the consent of his personal representative be granted to the survivor or survivors of the joint applicants.

Where an applicant (*e.g.* an employee) has agreed in writing to assign his interest in a patent when sealed (*e.g.* to his employers) the patent may be granted direct to the assignee. A similar provision applies where one joint applicant has assigned his interest, so that a patent may be granted to the assignee jointly with the other applicant or his assignee.

In case of disputes between joint applicants or their assignees, the Comptroller may allow one or more to proceed alone and may grant a patent to him or them subject to all parties being heard.

An appeal lies to the Appeal Tribunal. (12.)

Joint Patentees.—Where a patent is granted to several persons jointly they shall be treated legally as if they were joint tenants, but subject to any contract to the contrary each may use the invention for his own profit without accounting to the others but can only grant licences with their consent. On death the beneficial interest of a joint tenant passes to his personal representatives.

If joint patentees disagree the Comptroller may give directions as to the sale, lease, or licence of the patent, or the use or exploitation of the rights, and in default of compliance the Comptroller may empower any person to execute the requisite documents to give effect to his directions.

Appeal lies to the Court. (37.)

Licences.—Licences may be exclusive or non-exclusive. The rules of contract generally apply, but certain conditions are illegal (see 'Illegal conditions, etc.'). Furthermore a licence may be determined by either party on three months' notice after all the patents in force at the date of the licence have ceased to be in force, notwithstanding anything in the licence to the contrary. (38.) This is intended as a check on the use of the patent monopoly as a means of extracting royalties for unreasonable periods. Limited licences under patents may be granted to control price of resale, etc., but the terms of such licences must be brought to the notice of purchasers at the time of purchase.

Licences of Right.—Patents may be indorsed 'Licences of Right' with the following effect:—

1. Any person can claim a licence upon terms which, if not agreed, may be settled by the Comptroller.
2. Such a licence may exclude the importation from abroad of any goods covered by the patent.
3. If the patentee will not take action against infringers the licensee may take action in his own name. In an infringement action the defendant may take a 'Licence of Right' to avoid an injunction, with limitation of damages if his infringement is not by importation.
4. A patent indorsed 'Licences of Right' is subject to one-half of the normal renewal fees.

Applications for indorsement may be opposed and must not be precluded by contract. The indorsement of main patents extends to patents of addition thereon and *vice versa.*

An indorsement may be cancelled on request of patentee, but such applications are open to opposition.

An appeal from the Comptroller lies to the Court. (24.)

Manufacture.—A mere scheme or plan is not a 'manufacture' for which a patent may be granted.

Marking.—No marking necessary before acceptance but subsequently, see 'Infringement, Innocent.'

Medicines.—See 'Chemical Processes, etc.'

Opposition to Grant of Patent.—Within two months from date of advertisement of acceptance or within one month's extension if applied for within the said two months, notice of opposition by anyone having an interest may be filed at the Patent Office, on one or more of the following grounds only :—

(*a*) Fraud, *i.e.* whole or part of invention obtained from opponent.

(*b*) Prior publication in a U.K. patent specification dated within fifty years of the date of the application, or prior publication (in such a way as to make the invention available to the public) in any other document published in the U.K. (other than a patent specification over 50 years old).

(*bb*) Prior claiming in a U.K. specification not published in time to count as a prior publication.

(*c*) Insufficient or unfair description.

(*d*) Disconformity between provisional and complete specifications, the opponent having applied for a patent for the disconforming matter, or such matter having been published.

(*e*) In the case of applications under the Convention disconformity between the complete specification and the basic foreign specification with publication or patent application by opponent as under (*d*).

(*f*) Invention made in Germany or Japan between September 3, 1938, and December 31, 1945, or made between such dates by a German or Japanese National in any territory which was then enemy territory.

Appeal lies to the Appeal Tribunal. (11.)

Any person who would have been entitled to oppose the grant of a patent may within twelve months of sealing apply to the Comptroller for revocation on any of the above grounds. Such an application (often referred to as 'belated opposition') cannot be made if court proceedings on the patent are pending. Appeals in belated oppositions are to the Court, not to the Appeal Tribunal. (26.)

Patentee.—'Patentee' means the grantee or the subsequent registered proprietor of the patent. (93.)

Provisional Protection.—Inventions may safely be used and published between the dates of 'application' and 'acceptance' such period of protection being called 'provisional protection.' (4.)

Register of Patents.—The Register of Patents is kept at the Patent Office and contains entries of all matters pertaining to the validity and proprietorship of patents and all proceedings in relation thereto. Register is available for public inspection and certified copies of entries are obtainable. Copies of deeds, licences and any other documents entered in the Register must be supplied for filing in the Patent Office. (28) and (67.)

Restoration of Lapsed Patents.—Patents lapsed through non-payment of renewal fees may be restored by the Comptroller if he is satisfied that the omission to renew was unintentional and no undue delay occurred in applying for restoration. The Comptroller will impose terms to protect those who have started working the patent in the interim period. Opposition may be entered. The fee on application to restore is £20.

An appeal lies to the Court. (20.)

Revocation of Patent.—Revocation of a patent may be obtained, on petition to the Court, or by way of counterclaim in an infringement action, on any of the following grounds :—

(*a*) Invention already validly patented.

(*b*) Subject to special provisions as to convention applications the true and first inventor is not a party to the application.

(*c*) Patent obtained in fraud of the rights of the person seeking revocation.

(*d*) Invention not a 'manner of new manufacture.'

(*e*) Invention not new.
(*f*) Invention obvious and does not involve invention.
(*g*) Invention not useful.
(*h*) Insufficient description in the complete specification of the nature of the invention and how it is to be performed.
(*i*) Scope of the monopoly not sufficiently and clearly ascertained.
(*j*) The best method known to the applicant of carrying the invention into practice not described in the complete specification.
(*k*) Patent obtained on a false suggestion.
(*l*) Disconformity between the provisional and complete specification, the disconforming matter not being new or not invented by the applicant; or in the case of Convention applications disconformity between the complete specification and the basic foreign specification.
(*m*) Primary or intended use of the invention contrary to law.
(*n*) Patentee has contravened or not complied with the conditions of the patent.
(*o*) Invention secretly worked before date of patent on a commercial scale in the United Kingdom by the patentee or others not being a Government Department or authorised contractors therefor.
(*oo*) In a war period secret working by a Government Department or their agents before date of patent, such working not being by reason of a disclosure by the applicant.
(*p*) That in the case of substances prepared or produced by chemical processes or intended for food or medicine the specification contravenes Section 38a of the Act.
(*q*) Invention made in Germany or Japan between September 3, 1938, and December 31, 1945, or made between such dates by a German or Japanese national in any territory which was then enemy territory.

Any of the above grounds of revocation are available as a defence in an infringement action. (25.)

As to an application to revoke by way of 'belated opposition' see under 'Opposition.'

Royal Arms.—The grant of a patent does not entitle the patentee to use the Royal Arms or to place them on the patented article. Penalty £20. (90.)

Sealing of Patent.—Normally patents must be sealed (£1) not later than twenty-one months from date of application, but if any extension of time has been allowed for filing or accepting the complete specification, a period of twenty-five months is allowed for sealing.

In the case of opposition or appeal the period allowed for sealing is determined by the Comptroller or the Appeal Tribunal.

When a patent is granted to legal representative of a deceased applicant, it may be sealed at any time within twelve months of the death or later if allowed by Comptroller.

Sealing may be delayed for periods up to six months upon application and payment of extension fees and further extensions may be allowed as may be necessary to avoid hardship which might otherwise arise in connection with applications in countries abroad.

Search—Novelty.—The Examiner investigates whether the invention claimed in the complete specification has been wholly or in part claimed or described in any complete specification (or its corresponding provisional specification) left pursuant to an application for a patent in the United Kingdom dated within fifty years prior to the priority date or date of filing of the application. If invention has been wholly or in part claimed or described as aforesaid, applicant is allowed to amend his specification and the amended specification is examined in the same manner as the original specification. If the Comptroller is satisfied that no objection exists, he accepts the specification in the absence of any other lawful objection. If the Comptroller is not so satisfied he determines whether a reference to any particular prior specification ought to be made by way of notice to the public.

If the Comptroller is satisfied that the invention claimed has been wholly and specifically claimed or described in any prior specification he may, instead of inserting references, refuse to grant a patent.

If within the knowledge of the Comptroller the invention claimed has been made available to the public by publication in the United Kingdom in any document prior to the date of the application or its priority date (other than a specification more than fifty years old), the above provisions apply.

Appeal lies to the Appeal Tribunal. (7.)

The Examiner also investigates specifications relating to applications of prior date but published after the application date or priority date of the application, to ascertain whether the invention claimed has been wholly or in part claimed in any such prior specifications. In the case of whole or partial prior claiming, complete specification may be amended, or if not satisfactorily amended Comptroller may insert a reference to other specifications by way of notice to the public.

Appeal lies to the Appeal Tribunal.

Novelty investigations give no guarantee of validity of patent and Board of Trade accept no liability in respect of the investigations and reports thereon. (8.) See also 'Examiners Reports.'

Secret Patents, etc.—The inventor of any improvement in instruments or munitions of war may assign the invention and ultimate patent to the Secretary of State for War, for Air or the Admiralty, with or without valuable consideration.

When such an assignment has been made, the Secretary of State or the Admiralty can issue a certificate authorising the Comptroller to treat the application and all relevant papers as secret. (30.)

Under Emergency Legislation which will continue in force until December 10, 1950, various Ministers have powers to control applications for patents if such course is in the public interest.

Specifications.—Provisional specification must describe the nature of the invention. Complete specification must describe and define the invention in detail and the manner in which it can be carried into practice. Drawings may be required as part of the specification. Complete specifications must end with a clear statement of the invention claimed. The 'claims' of the complete specification are important because they define the ambit of the monopoly provided by the patent. In the case of chemical inventions samples may be required or may be supplied. (2.)

Complete specifications (£4) if not filed on application must be filed within twelve months from date of filing. One month's extension of this period is obtainable on payment of a fee. If a complete specification is not filed within thirteen months the application is regarded as abandoned. (5.)

Term of Patent.—The normal term of a patent granted on and after August 1, 1938, begins on the date of the patent and ends at the expiration of sixteen years from the date on which the specification accepted as complete specification is treated by the Comptroller as having been first left.

Both these dates are entered upon the Register of Patents. Continuance of the patent is, however, subject to the payment of the prescribed renewal fees which commence at the end of the fourth year from the date of the patent (not date of sealing or date of filing complete specification). Up to three months extension of time for payment can be obtained on payment of a fee (17 and Rules). The term of patents granted before August 1, 1938, is sixteen years from the date of the patent.

Threats.—If anyone by circulars, advertisements or otherwise threatens anyone with an action for infringement of patent or other like proceedings, then any person aggrieved thereby may bring an action to restrain such threats and recover damages, whether the person making the threats is or is not interested in a patent or patent application, unless the person making the threats can prove they are justified, *i.e.* the patent contains valid claims which are or would be infringed, or rights arising from the acceptance of a complete specification would be infringed in respect of a claim not capable of being successfully opposed. The defendant in such proceedings may seek by counterclaim the relief to which he would be entitled in an infringement action.

Vessels, Aircraft, etc., use of Inventions on.—A patent is not infringed by its use in a vessel, aircraft or land vehicle belonging to a Convention Country if it is used in the body of the vessel, etc., or in the machinery, tackle, etc., and if the vessel, etc., comes only temporarily or accidentally into Great Britain or its territorial waters and (in the case of a vessel) the invention is used exclusively for the actual needs of the vessel. (48.)

The Air Navigation Act, 1947, provides for wider protection for aircraft in the above matters, covering for example the storage in U.K. of spare parts and equipment for foreign aircraft.

Working (Definition).—'Working on a commercial scale' means the manufacture of the article or carrying on of the process described and claimed by a definite and substantial establishment or organisation and on an adequate or reasonable scale. (93.)

Working (Abuse of Monopoly).—At any time after three years from sealing any interested party may apply to the Comptroller for relief on the ground that the patent monopoly is being abused in any of the following circumstances :—

1. The invention is not being worked in the United Kingdom on a commercial scale and there is no satisfactory reason for such non-working.
2. The working in the United Kingdom is being hindered by importation from abroad of the patented article by the patentee or with his assent or by infringers against whom the patentee has not taken action.
3. If the demand in the United Kingdom is not being met to an adequate extent and on reasonable terms.
4. If because the patentee will not grant licences on reasonable terms trade or industry is being prejudiced, it being in the public interest for licences to be granted.
5. If any trade or industry is being prejudiced by conditions attached by the patentee to purchase, hire, licence, or use of a patented article or to the working of a patented process.
6. If the patent relates to a process involving the use of materials not protected by the patent and is being used by the patentee so as unfairly to prejudice the manufacture, use or sale of such materials in the United Kingdom.

The Comptroller may take action as follows :—

1. He may endorse patent 'Licences of Right.'
2. He may grant a licence to the applicant on suitable terms.
3. If he is satisfied that lack of capital is preventing exploitation and the patentee does not undertake to find such capital, he may grant an exclusive licence to the applicant on suitable terms. In granting such an exclusive licence the Comptroller will usually prefer an existing licensee to other persons.

4. If he thinks the manufacture use or sale of unpatented materials for use in carrying out the invention contained in the patent is being prejudiced, he may grant licences to the applicant and to any of the applicant's customers on such terms as he may think fit.

5. He may make no order if he considers it just so to do.

6. As a last resort, and subject to certain conditions he may revoke the patent.

DESIGNS.

Patents and Designs Acts, 1907–1946.

Additional Registrations.—A design already registered may be subsequently registered by the proprietor in other classes such registrations expiring with the original registration.
Similarly a design which differs from a previously registered design of the proprietor only in unimportant respects may be registered in the same class expiring with the original registration. (50.)

Anticipation.—A design registration is not invalidated by prior confidential disclosure of the design, or if the design is disclosed in breach of such confidence, or by the acceptance of a first and confidential order for a new textile design intended for registration. (55.)

Application for Registration (10*s*.).—Upon the application of a person claiming to be the proprietor the Comptroller may register any new or original design not previously published in the United Kingdom. The same design may be registered in more than one class. If the Comptroller refuses registration, an Appeal may be made to the Appeal Tribunal. Applications not completed within the prescribed time through neglect on the part of the applicant may be regarded as abandoned. Designs are registered as of the date of application for registration. (49.)

Article.—'Article' (as respects designs) means any article of manufacture and any substance wholly or partly artificial or natural. (93.)

Cancellation of Registration.—Any person may apply to the Comptroller for cancellation of a registration on the ground that it was published in the United Kingdom prior to the date of application. On such an application the only question is whether the design has been published; the question of its novelty or originality does not arise. See also 'Validity.'
Appeal lies from the Comptroller to the Appeal Tribunal, and the Comptroller may refer application to the Appeal Tribunal for trial. (58.)

Certificate of Registration.—Certificates of registration are issued and duplicates may be obtained in the event of loss. (51.)

Certificates of Validity.—Provisions similar to those for patents. (61.)

Classes of Designs.—For the purposes of registration articles are classified in fifteen classes based partly on the materials of which the article is composed, *e.g.* metal, wood, glass, etc., and partly on the nature of the article, *e.g.* books, shoes, carpets, etc. Registration in any class covers the application of the design to any article in the same class. Correspondingly novelty and originality must be judged against designs previously applied to other articles in the same class.

Compulsory Licence.—Interested parties may apply for a compulsory licence on grounds that the design is industrially used outside the United Kingdom and not to a reasonable extent within the United Kingdom. The Comptroller may make any order at his discretion. Appeal lies to Appeal Tribunal and the Comptroller may refer application to the Appeal Tribunal for trial. (58.)

Convention Applications.—Priority of date may be obtained for applications made under the International Convention within six months from the application for protection in the convention country. (91.)

Copyright in Designs.—'Copyright' in designs is the exclusive right to apply a design to an article in any class in which the design is registered. (93). The acts of piracy of design copyright in respect of which proceedings may be taken are, however, limited, see 'Infringement.' Design copyright is to be distinguished from artistic copyright under the Copyright Act, 1911. There may, however, be some overlapping of the two forms of protection, and reference should be made to fuller treatises on the subject for further information.

Crown Rights.—The provisions as regards patents also apply to registered designs. (29, 58a.)

Design.—'Design' means only the features of shape, configuration, pattern or ornament applied to any article by any industrial process or means, whether manual, mechanical, or chemical, separate or combined, which in the finished article appeal to and are judged solely by the eye; but does not include any mode or principle of construction or anything which is substance is a mere mechanical device. (93.)

Duration of Design.—Registration of a design gives copyright for five years from the date of registration. This may be extended by renewal for two successive periods of five years. (Fees £2 and £5 respectively.) (53.)

Exhibitions.—Provisions similar to those for patents apply to designs. (59.)

Infringement.—It is an infringement to apply a registered design or any fraudulent or obvious imitation thereof for the purpose of sale to any article in any class of goods in which the design is registered, except with the licence or consent of the registered proprietor or to do anything with a view to enable the design to be so applied. It is also an infringement to publish or expose for sale articles known to have a design or a fraudulent or obvious imitation thereof applied thereto without the consent of the proprietor. The proprietor may bring an action for statutory penalties (not exceeding £100) or for an injunction and damages. (60.)

Inspection of Registered Designs.—Registered designs are open to inspection by the public in most classes as soon as registration has been effected but in classes 7 and 9 (paper hangings and lace) and in classes 13, 14 and 15 (textile goods) the right to inspect is withheld for periods of two and three years respectively. Any design cited as an anticipation of a new application may be inspected by the Applicant. Copies may be obtained when design is open to inspection. (56) and Designs Rule 67.

Marking.—Before any goods are marketed under a registered design each article must, except in the case of certain textiles, be marked REGISTERED, REGD. or RD. with the number of the relevant registered design. Failing this a proprietor cannot usually recover damages for infringement of the design unless he can prove that he took all the necessary steps to ensure the marking of the article or unless he shows that the infringement took place after the infringer had been notified of the existence of the design registration. (54) and Design Rule 64.

Mechanical Devices.—A design registration is not a 'little patent' and mere mechanical devices and principles of construction cannot be protected by such registration. (93.)

Proprietor of a New or Original Design.—'Proprietor' of a new or original design means any person for whom the author executes the work for good consideration, or any person who has acquired the design or the right to apply it to any article. In any other case the proprietor means the author. (93.)

Register.—Particulars of all designs, assignments, and documents relating thereto are kept in a Register which is open to inspection on payment of a fee. (52.)

Threats.—Provisions similar to those for patents. (61.)

Validity.—The validity of a design registration may be challenged by way of counterclaim in an infringement action. A design may also be removed from the register upon application to the Court by originating motion.

GENERAL.

Applications abandoned.—Documents or samples relating to patent applications (other than Convention applications) abandoned before publication and abandoned design applications are not open to public inspection. (69.) This rule does not apply to certain applications made by enemies during the war.

Assignments, Registration of.—All assignments relating to patents and registered designs must be entered on the Register and copies of the documen ts must be supplied. After six months from execution the fee for registration becomes increased from £1 to £10. (71) and Design Rules 1st Schedule.

Clerical Errors, Correction of.—Clerical errors in patent specifications or design application s may be corrected if they do not materially alter the meaning or scope of the document which is corrected. (70.)

Costs, Award of, by Comptroller.—The Comptroller can award costs in any proceedings before him under the Act, and his order may be made a rule of Court. In the case of opponents not residing or carrying on business in the United Kingdom the Comptroller may require security for costs. (73a).

Excluded Days.—Where the last day for carrying out an act required by law falls on a Saturday, Sunday or public holiday, the act may be done on the next following day. (82.)

False Marking.—Anyone falsely representing an article sold to be patented, or falsely representing any design applied to any article sold by him to be registered, is liable to fines not exceeding £5 for every offence. Any person marking an article as 'registered' after expiry of the copyright in a registered design is liable to a fine not exceeding £5. (89.)

Frivolous Applications and Illegal Patents or Designs.—The Comptroller may refuse a patent for an invention so obviously contrary to well-established natural laws that the application is frivolous, or to grant a patent or register a design the use of which would be contrary to law or morality. A patent may be granted with a disclaimer as to illegal use of the invention. Appeal lies to the Appeal Tribunal. (75.)

TRADE MARKS.

(Trade Marks Act, 1938.)

Alteration of Mark.—The registered proprietor may amend a trade mark in any manner not substantially affecting its identity. An appeal lies against the refusal, or terms imposed by Registrar, to the Board of Trade or to the Court. (35.)

Amendment of Application.—Errors may be corrected or amendments made at any time before or after acceptance, with the consent of the Registrar, the Board of Trade, or the Court. (17.)

Application (£1).—Any person claiming to be the proprietor of a trade mark may apply for registration. If registration is refused a written statement of grounds can be obtained and an appeal lies against the decision to the Board of Trade or the Court. (17.)

Assignment of Trade Mark.—A registered trade mark is, and is deemed always to have been, assignable and transmissible either in connection with the goodwill of a business or not. There are, however, certain conditions which are fully set out in the Act. (22.)

All assignments must be recorded in the Register. (25.)

Associated Marks.—Marks closely resembling other marks of the same proprietor already on the register may be registered subject to association with the prior mark or marks. (23.)

Portions of a trade mark may be registered as associated marks. (21.)

Several marks for the same description of goods, and similar as regards essential parts, but differing in respect of (*a*) statement of goods, (*b*) statements of number, price, quality or names of places, (*c*) other matter of a non-distinctive character not substantially affecting the identity of trade marks, or (*d*) colour, may be registered as a series of associated marks under one registration. (21.)

Associated trade marks can only be assigned or transferred together. Otherwise they are treated as independent marks. (23.)

Certificate of Registration.—Certificates of Registration are issued under the seal of the Patent Office. (19.)

Certificate of Validity.—A certificate of validity of a registered trade mark may be given by the Court and if successful in subsequent action, the proprietor may obtain full solicitor and client costs. (47.)

Certification Trade Marks.—The Board of Trade may permit the registration of a mark adapted to distinguish goods certified by any person in respect of origin, material, mode of manufacture, quality, accuracy, or other characteristic, from goods not so certified. Such a mark may not be registered in the name of a person who carries on a trade in goods of the kind certified. (37.)

Chemical Names.—No word which is the commonly used and accepted name of any single chemical element or single chemical compound, as distinguished from a mixture, may be registered. (15 (3).)

Coloured Marks.—A trade mark may be wholly or partly limited to specified colours; otherwise it is deemed to be registered for all colours. (16.)

Convention Applications.—Priority under the International Convention may be claimed provided application is made within six months of the foreign application.

Correction of Register.—The Registrar, on the request of the registered proprietor, may at any time correct errors in the name, address or description of the registered proprietor or a registered user, enter any change of name, address or description of the registered proprietor or a registered user, cancel the entry of a trade mark on the register, exclude any desired goods from the registration or enter any disclaimer or memorandum which does not extend the rights under the registration. An appeal lies to the Board of Trade or to the Court. (34.)

Costs and Security for Costs.—Registrar may award costs at his discretion and the award may be enforced by leave of the Court in the same manner as an Order of the Court. (44.)

Any party giving or contesting notice of opposition or appeal who does not reside or carry on business in the United Kingdom may be required to give security for costs. (18.)

Date of Registration.—Trade marks are registered as of the date on which the application for their registration was filed. See also 'Convention Applications.' (19.)

Defensive Trade Marks.—Proprietors of very well known invented word trade marks may obtain registration of those marks in respect of goods for which they do not use or propose to use them, provided they can show that the public would expect the mark, if used upon the additional goods, to indicate the registered proprietor. (27.)

Descriptive Marks.—Purely descriptive marks are not registrable but marks *prima facie* descriptive may become distinctive of a particular trader by long use (see under 'Registrable Marks'). Trade marks, originally distinctive, which in the course of time have come to be used in the trade in a purely descriptive way or which comprise the only practical name or description of the subject matter of an expired patent may not remain on the Register. (15.)

Disclaimers.—Registration of marks containing additional matter common to the trade, or of a non-distinctive character, may be permitted subject to disclaimer of any right to the exclusive use of such additional matter. (14.)

Discretionary Powers.—Where the Registrar is allowed any discretionary power it must not be used adversely to the applicant or registered proprietor without giving the applicant or registered proprietor an opportunity to be heard. (43.)

In appeals the Court has the same discretionary powers as the Registrar. (52.)

Duration.—Registration lasts for seven years, and is subject to renewal thereafter for periods of fourteen years without limit. (20.)

False Claim of Registration.—Any person representing an unregistered trade mark as registered is liable to a fine not exceeding £5 for each offence. The use of the word 'registered' or any other word falsely implying that registration has been obtained is not allowed. (60.)

Identical Marks.—Except as stated below, no trade mark shall be registered in respect of any goods if it is identical with or closely similar to a trade mark belonging to a different proprietor already on the Register in respect of the same goods or goods of the same description. (12 (1).)

In the case of honest concurrent user of identical or nearly identical marks or other special circumstances, the Court or Registrar may allow the registration of more than one proprietor for the same goods or description of goods with or without suitable conditions or limitations as regards mode or place of user. (12 (2).)

Where there are rival claimants to identical or nearly identical marks, Registrar may refuse to register any of them until their respective rights have been determined by the Court or settled by agreement approved by him. (12 (3).)

Infringement.—A registered proprietor may claim an injunction, damages, delivery up of infringing labels, etc., and costs in an infringement action. The validity of the registration may be attacked by the defendant by way of counterclaim. Infringement may take place either by marking goods with the offending mark, or by using such mark in advertisements or circulars issued to the public. It is not an infringement to use a registered mark on the genuine goods or so far as is reasonably necessary to indicate that spare parts or accessories are adapted to be used with the genuine goods.

Infringement by Breach of Certain Conditions.—The proprietor of a registered trade mark may by contract impose certain conditions in relation to his goods and any person having notice of such conditions and not observing them is liable to be sued for infringement. The conditions must relate to such matters as prohibition of use of the mark upon the goods after they have been altered, removal or obliteration of the mark, application of other marks or injurious matter, etc. (6.)

Inspection of Register.—The Trade Marks Register kept at the Patent Office is open to inspection by the public on payment of a fee. (1.)

International and Colonial Arrangements.—Priority based on foreign or colonial registrations can be obtained under the International Convention and International Arrangements, subject to application in the United Kingdom within six months of the basic application abroad. Patents and Designs Act, s. 91.

Lapsed Marks.—Trade Marks which have lapsed may be cited against new applications for one year after they have lapsed, unless it is proved that the mark was not in fact used for two years prior to its removal from the Register, or that no deception or confusion would be likely to arise from the use of the mark applied for. (20.)

Licences.—Licences under trade marks vitiate the validity of the mark except in the case of permitted use by registered users.

Mark.—A 'mark' includes a device, brand, heading, label, ticket, name, signature, word, letter, numeral or any combination thereof. (68.)

Merchandise Marks.—Merchandise marks include trade marks and trade descriptions and are protected in the public interest under the Merchandise Marks Acts.

Name and Address, etc., Use of.—No trade mark registration shall interfere with the *bona fide* use by any person of his own name or place of business or that of his predecessor in business, or the use of any *bona fide* description of the character or quality of the goods, provided that the description is not likely to be taken as importing a reference to the owner (or registered user) of the mark. (8.)

Non-completion.—Twelve months is allowed for the completion of registration, and after issue of a non-completion notice applications may be treated as abandoned if not completed within the time allowed in the notice. (19.)

Non-user of Trade Mark.—Any person aggrieved may apply to the Court or the Registrar for the removal from the Register, in respect of any goods for which it is registered, of any trade mark on the ground that it was registered without any *bona fide* intention to use it in connection with the goods in question and there has in fact been no user, or that there has been no *bona fide* user during the previous five years up to one month before the date of application. Non-user is not a ground for removal if it is due to special circumstances in the trade, and if there was no real

intention to abandon the trade mark in respect of the goods in question. (26). Special provision was made under Emergency legislation for protection of marks which could not be used on account of reduction of trading activities under Government orders during the war.

Opposition to Registration.—At any time within one month of the advertisement of an application in the *Trade Marks Journal*, any person may lodge opposition. The notice must state the grounds of opposition, and the applicant must file a counter-statement if the application is not to be treated as abandoned. Evidence may be filed in support of the opposition and application and after a hearing the Registrar determines whether the registration shall be allowed, and subject to what conditions, if any.

Appeal lies to the Court. No further material and no further grounds of objection may be brought forward on appeal, except by special leave of the Court. (18.)

Parts A and B of the Register.—The Register of Trade Marks is divided into two parts known as Part A and Part B. See 'Registrable Marks.'

Passing-off.—A trader has the right at common law, and apart from registration, to prevent the goods of another being passed off as his goods by reason of similarity in marking or get-up.

Prior User, Rights of.—No registered proprietor can prevent anyone else from continuing to use a trade mark which has been in use since before the date of first user of the registered mark, or the date of its registration, whichever is the earlier, nor can he object to such other person obtaining registration under Section 12 (2). (7.) See 'Identical Marks.'

Royal Arms.—The use of the Royal Arms or devices closely similar thereto in connection with any trade, business, calling or profession, so as to suggest Royal Patronage by any person not authorised may be prevented by injunction, but without prejudice to the right, if any, of the proprietor of a trade mark containing such arms to continue using it. (61.)

Rectification of Register.—Any person having good cause may apply to the Registrar or the Court to have the register corrected or rectified.

Marks may be transferred from Part A to Part B of the Register. (32.)

Registered Users.—Provision is made for the entry on the Register of the names of persons to whom the proprietor of a registered trade mark desires to allow a 'permitted use' of his mark on specified goods, provided the Registrar approves the arrangement as not being contrary to the public interest. There must be a connection in the course of trade between the registered user and the marked goods, and suitable conditions or restrictions must be laid down. (28.)

Registrable Marks.—A mark registrable in Part A must contain or consist of at least one of the following particulars :—

1. The name of a company, individual, or firm, represented in a special or particular manner.
2. Signature of the applicant or a predecessor in his business.
3. One or more invented words.
4. A word or words having no direct reference to the character or quality of the goods, and not being according to its ordinary signification a geographical name or a surname.
5. Any other distinctive mark, but names, signatures or words not complying with 1, 2, 3 or 4 shall not be registrable except upon proof of distinctiveness.

Distinctive in this section means adapted to distinguish the goods of the proprietor of the mark from those of other persons, having regard to inherent distinctiveness or actual distinctiveness acquired by use. (9.)

A mark registrable in Part B must be capable in relation to goods for which registration is sought of distinguishing goods with which the proprietor is or may be connected in the course of trade from goods in the case of which no such connection exists either generally or subject to limitations.

In the case of Part B applications importance will be attached to the extent that the mark is

(*a*) inherently capable of distinguishing.

(*b*) by reason of use or other circumstances is in fact capable of distinguishing. (10.)

All applications whether in Part A or Part B are subject to a search as to conflict with prior marks and are subject to Sections 11 and 12 of the Act, see 'Identical Marks' and 'Unlawful Marks.' The Registrar may refuse such applications or may accept them absolutely, or subject to amendments, modifications, conditions or limitations. The Registrar's decision is subject to Appeal to the Court or the Board of Trade. (17.)

Marks or any part or parts thereof may be registered in both Parts A and B by the same proprietor. (10.)

Registration for Specific Goods.—Trade Marks must be registered in respect of particular goods or classes of goods. (3.)

Registration valid after Seven Years.—After seven years the original registration of all registered marks in Part A shall be regarded as valid in all respects, unless obtained by fraud, or contrary to Section 11. See 'Unlawful Marks.' (13.)

Rights of Proprietor.—A valid trade mark registration in Part A gives to the registered proprietor the exclusive right to the use of the trade mark upon or in connection with the goods for

which it is registered, saving the rights of registered proprietors of identical or substantially identical marks. (4.)

Rights under Part B registrations are the same except that no relief can be obtained in an infringement action if the defendant establishes that the use of which the plaintiff complains is not likely to deceive or cause confusion. (5.)

Sale of Goods.—There is an implied warranty upon a sale of goods that no false trade mark has been applied (Merchandise Marks Act, 1887, s. 17). False marking may also be a breach of the conditions implied by Sections 12 and 14 of the Sale of Goods Act, 1893.

Sheffield Marks.—Subject to the approval of the Registrar, the Cutlers' Company may enter metal marks in the Sheffield Register on behalf of persons carrying on business in Hallamshire, in the County of York, or within six miles thereof, and any marks so registered shall also be entered by the Registrar in the Trade Marks Register. The Cutlers' Company is notified by the Registrar of all applications received by him in respect of metal marks. (38.)

Textile Marks.—An application for the registration of a trade mark in respect of textile goods may be made to the Registrar, either at the Patent Office or at the Manchester Branch of the Trade Marks Registry. All applications are notified to the Registrar and their allowance or refusal is determined by the Registrar subject to appeal to the Court or the Board of Trade.

In respect of textile goods being piece goods :

(*a*) No line heading alone shall be registered or deemed a ' registrable mark.'

(*b*) Registration of a trade mark shall not give any exclusive right to the use of a line heading. (39.)

Trade Mark.—A ' trade mark ' other than a ' Certification trade mark ' means a mark used or proposed to be used in relation to goods for the purpose of indicating, or so as to indicate, a connection in the course of trade between the goods and some person having the right either as proprietor or as registered user to use the mark. (68.)

Unlawful Marks.—Deceptive marks, or those contrary to law or morality, or any scandalous design, may not be registered. (11.)

SECTION XLIV

DEPRECIATION OF PLANT AND MACHINERY—INCOME-TAX ALLOWANCES

(Contributed by Alfred B. Searle, Past-Pres. Valuers' Inst., Licensed Appraiser and Valuer.)

(Pp. 1017–1021)

SECTION XLIV

DEPRECIATION OF PLANT AND MACHINERY—INCOME-TAX ALLOWANCES.

(Contributed by Alfred B. Searle, Past-Pres. Valuers' Inst., Licensed Appraiser and Valuer.)

Depreciation is a term applied to the reduction in value which occurs in plant or machinery, no matter whether they are in use or stored. The amount of depreciation is affected by the condition in which they are kept and by the availability or otherwise of more suitable appliances.

The loss in value is most properly met by an annual charge (to which the term 'depreciation' is commonly applied); the total of such charges should, when the appliance is worn out, discarded, or replaced by another, be equal to the difference between the original cost and the sum obtained (if any) by the sale of the worn-out or discarded appliance. Thus if a machine costing 1,000*l.* is discarded as useless after ten years and then is sold for only 10*l.* the total depreciation is 990*l.*, the annual depreciation is 99*l.*, and the residual value is 10*l.* The depreciation may be affected by any sums spent on renewing parts of a machine or plant though these are usually charged separately in the accounts, as *renewals*.

For methods of calculating depreciation, see p. 1019.

Deterioration is the loss in value due to the lapse of time and to the use of the appliance. It occurs more rapidly when a machine is used than when it is stored under good conditions, but may be just as serious if a machine is stored under unsuitable conditions. In deterioration is also included the loss in value which occurs directly the ownership of anything has been transferred from the seller to the buyer, the 'second-hand value' being always less than the market price when new; the difference forms part of the loss by deterioration.

Obsolescence is the loss in value which occurs when a more suitable appliance can be used for the same purpose. It occurs when a machine is discarded, long before it is worn out, because another machine is more profitable to use.

Renewals are the cost of replacing worn-out or damaged portions of machinery or plant. To some extent they reduce the depreciation, but for convenience are usually treated quite separately. The Inland Revenue officials for some purposes, take renewals instead of depreciation when calculating allowances for income-tax assessments. Thus, instead of allowing 3*l.* per annum as the depreciation on a typewriter, they allow nothing until the typewriter is replaced by another one; then they allow the cost of the new machine less the sum obtained by the sale of the old one, or a sum estimated to be obtainable for it.

Repairs are essential to the maintenance of machinery and equipment in good condition but, as it is almost impossible to estimate their cost long before they are required, no allowance is made for repairs when calculating the allowance for depreciation.

Residual Value is the value of a machine or plant at the end of a given period—usually, but not necessarily, when it is worn out or discarded. In dealing with the estimated life of plant or machinery, the residual value of such plant or machinery when its period of usefulness has expired must be taken into account, and with plant of special nature it may generally be assumed that in proportion to its special character its residual value will be low Plant of general character, engines, boilers, shaftings, pulleys, etc., carries a high residual value in the event of displacement.

Diminishing Value is the reduction in the value of plant or machinery as the result of depreciation or obsolescence, or both. The term is also used to indicate the value obtained by deducting depreciation for one year at a given rate, and then applying the same rate to the reduced value so obtained. Thus, if a machine costing 1,000*l.* is depreciated at 10 per cent. per annum on the diminishing value it will be worth 900*l.* at the end of the first year, 900*l.* − 90*l.* or 810*l.* at the end of the second year, 810*l.*—81*l.* or 729*l.* at the end of the third year, and so on. This method is largely used because of its convenience, but it requires a larger rate of depreciation than when the

rate is calculated thoughout on the original value. In recent years there has been a tendency to revert to the basis of original cost. For instance, a rate of 10 per cent. on the original cost will exhaust the value in ten annual instalments; but a rate of about 16½ per cent. would be required to exhaust the value in the ten years if calculated on the diminishing value remaining after the deduction of depreciation in each year.

The argument in favour of basing the calculation on the diminishing value is that the sums written off in the earlier years were greater when the repairs were small, and this tended to equalise matters. Too often, however, the two factors of rate and basis have not been settled together, and consequently the basis of diminishing value has had the effect of leaving a considerable value in the books when the plant has disappeared. For instance, a 5 per cent. rate generally carries the impression of a twenty years' life, yet taking 100*l.* as the cost, and writing off on the diminishing value, the balance left at the end of the twentieth year is 37*l.* 14*s.* After twenty-one years' service the residual value of the plant would be, roughly, one-third of the first cost a figure far too high in most cases.

Rate of Depreciation.—There are several methods of calculating the amounts to be set aside annually for depreciation. One of the most widely used—but not necessarily the best—is to deduct a pre-arranged percentage of the original cost of the machine or equipment. This simple method has serious objections in some cases (see p. 1019). The determination of the rate of depreciation of various kinds of machines, etc., requires special skill and experience, and has given rise to specially trained appraisers or valuers, each of whom has expert knowledge of certain classes of plant or machinery; a valuer who is an expert on brewing appliances would probably know little of cotton-spinning, bricks, or presses, and *vice versa*. It is, therefore, important that valuations should be made by men of *special* knowledge and not (as is usually the case in valuations for rating assessments and some other purposes) by men with only a general knowledge.

As different parts of a plant and different machines wear out or become obsolescent at different speeds, it is desirable to decide on a rate for each separate machine or appliance, though some grouping is desirable.

There is much difference of opinion as to what are suitable rates of depreciation, especially when accurate data on which to base them are not available. For several years past the great increases in the cost of machinery and other equipment has resulted in all allowances for depreciation being much too low. A correct allowance would enable a new machine to be purchased with the total sum set aside for depreciation plus the proceeds of sale of the old machine. At present, new machines cost so much more, that the depreciation allowance may not equal more than half the cost of a new machine. This fact should be borne in mind when calculating rates of depreciation.

The following figures are only approximate, but they give a useful *general* idea, which may be greatly modified in particular cases.

Motive Power, Boilers, Engines, etc.—Some authorities place the depreciation on engines as high as 12½ per cent., and boilers at 10 to 15 per cent. Much depends on stress of working and the water. Mathieson places the life of a boiler at fifteen years, with renewal of the furnace at the end of ten years, and recommends a rate of 7½ per cent. per annum on the diminishing value on this class, no provision being made for obsolescence. This is, however, an extreme view.

Power Transmitters.—Shafting, pulleys, and plant under this head have, as a rule, a high residual value, even if displaced, and a rate of 5 per cent. will usually be found sufficient. Belting should be classified as loose tools. In this class a rate of 10 per cent. would be desirable. No provision is made for obsolescence.

Process Plant, Fixed.—Under this head would be placed all heavy process plant, the renewal portions of which are classified under their separate heading. The deterioration would then be met by a rate of 10 per cent. With a lesser rate, an obsolescence rate depending on the nature of the processes will also be needed; it may be 6 per cent.

Process Plant (Wearable).—For this it is necessary to provide a heavy renewal rate. By the dissection of this plant the separation of the different details can be secured, and it is easy to apply to each a distinctive rate for renewals applicable to its character and requirements, based on experience. It will usually be at least 10 per cent., and may be 50 per cent.

Process Plant (Loose).—In the case of loose tools, the yearly valuation gives the most reliable results. In some cases there is an annual increase in this class, but this should verify itself. It is to be remembered that a going-concern valuation can seldom work out at more than half the original cost value. Hence, the depreciation on loose tools and on some parts of process-plant should be at least one-third of the original cost.

Accessory Plant.—Usually has a high residual value due to general utility, and consequently a rate for deterioration of 5 per cent. and a renewal rate of 10 per cent. should suffice.

Horses, Carts, Harness, etc.—The risks and depreciation on this class are too great to include in any average rate. A revaluation annually gives probably the best results.

The following table gives (A) the approximate life in years of the various classes of machinery etc., in an electrical undertaking if properly maintained, and (B) the probable residual value at the end of that term in per cent. of original cost. It affords a useful means of checking any suggested rate of depreciation on the applicances, etc., mentioned, but is very incomplete and some figures in column A are too large for many conditions.

	A.	B.		A.	B.
Accumulators	15	10	**Electric motors**	15	9
Arc lamp-posts	30	5	**Engines and other machinery**	20	6
Arc lamps	12	5	**Foundations**	100	nil
Belting and ropes	10	nil	**Machine tools**	3	nil
Boilers (Lancashire)	20	3	**Meters, switch-boards**	3	nil
Boilers (water tube)	20	5	**Motor trucks**	4	20
Buildings	80	nil	**Pulleys**	20	10
Buildings (sub-station)	50	nil	**Railway wagons**	15	10
Cables (armoured)	20	15	**Shafts**	25	10
Cables (solid in wood trough)	30	12	**Sub-station equipment**	25	12
Cranes	20	5	**Tools and loose plant**	3	nil
Dynamos and alternators	20	8	**Water tanks (G.I.)**	5	nil

QUINQUENNIAL VALUATIONS.

It is very important that an independent valuation of all land, buildings, plant, and machinery should be made by a qualified valuer at least once every five years.

Such a valuation if properly made will correct any errors due to wrong rates of depreciation and to over-valuation of assets. It will enable the accountants to prepare a more satisfactory balance sheet and will, in many cases, save much disappointment and loss.

Many large engineering firms rightly attach great importance to these five-yearly valuations and find them invaluable in many ways.

It is essential that such valuation should be made by a Valuer having special knowledge of the articles to be valued. A Valuer accustomed to doing mainly the work of an Auctioneer cannot be expected to have the necessary special knowledge.

METHODS OF ESTIMATING DEPRECIATION.

Appraisers and valuers have various methods of estimating the depreciation of plant and machinery, and two men of equal skill may not agree in their estimate because they use a different method, or because they have different ideas as to the durability of a particular appliance; the latter affects the rate of depreciation. Several methods of estimating the allowance to be made for depreciation are, therefore, in use, viz :—

(1) The *Residual Value* (p. 1017) is deducted from the original cost and the difference is divided by the number of years which the machine or equipment is expected to be in condition, suitable for its satisfactory use. By this means, the same amount is deducted each year so that at the end of the period the total of the sums so deducted plus the amount for which the machine or equipment can be sold will equal its original cost.

(2) The *Diminishing Value* (p. 1017) is calculated year by year and the resulting figure is the amount allowed for depreciation. Unless special care is taken this method tends to give too low a result (see p. 1018), and the calculation are much more tedious than those in method (1).

A Table showing Diminishing Values is shown on p. 1024.

The chief advantage of using the method of Diminishing Values is that it enables large deductions to be made when the machine or equipment is new and the cost of repairs and renewals is likely to be small, and smaller deductions are made when the cost of repairs and renewals will probably be greater. By this means the *total* annual charge will be kept more constant.

(3) *Amortization* is, theoretically the best method, either a *Vertual Rent* being charged or a *Sinking Fund* created. This method is only suitable when the amounts set aside annually are actually reinvested at the rates of (compound) interest assumed in the calculations. If no such investment is possible amortization will give far too low a figure for depreciation.

If the annual allowances calculated by methods (1) and (2) are merely placed in a box, their total, at the end of the period, will be precisely the sum accumulated. If, however, each regular payment is promptly invested at a suitable rate of interest and the interest each year is also invested at the same rate, the sum finally accumulated will be considerably larger than the total of the regular payments. For instance, the regular payment of £84 per annum and its prompt investment at 3 per cent. per annum compound interest would complete the repayment of a Loan of £1,000 at 3 per cent. per annum in 15 years. If only a lower rate of interest is allowed the time taken would be correspondingly longer.

In calculating amortization there are two distinct aims to be considered :—

(a) The accumulation of an agreed sum at the end of a specified number of years. For this purpose a *Sinking Fund* consisting of regular annual payments, suitably invested will suffice. Each payment must then be the sum which, under the prescribed conditions, will, with interest, provide the desired amount. The annual payment required can be calculated mathematically or ascertained from Sinking Fund Tables.

(b) The accumulation of an amount which will extinguish a loan and also repay the annual interest on the loan is best effected by the payment of a *Virtual Rent*, which can also be calculated mathematically or ascertained from Virtual or Sitting Rent Tables.

The great difficulties, at the present time, in investing regular amounts at compound interest have made amortization a matter of great uncertainty and many engineers and others tend to ignore its possibilities and so suffer a corresponding loss.

One very convenient method of amortization is to arrange an *Endowment Insurance Policy* with a reliable Company, so as to provide the sum required at the date arranged. The premiums paid for such a policy will be the amounts set aside for depreciation or obsolescence or for the repayment of a loan and will be entered as such in the accounts. Such a policy can be based on the amount which will be required to purchase a new machine or equipment at the end of the period and can therefore provide for increases in the cost in a manner not allowed for by other methods.

(4) *Valuation.*—The machine or equipment is valued each year by a skilled valuer and the difference between this figure and that in the Accounts is allowed for depreciation. This method is not used each year but at intervals of five or more years. In the intervening periods one of the other methods is used.

(5) *The Inland Revenue* in some cases, makes allowances on renewals instead of a depreciation allowance (see *Renewals*, p. 1017 and *Income Tax*, below).

None of these methods are wholly satisfactory because of the complex nature of the problems involved. Thus, if a machine or equipment is valued as a whole the allowance may be too low, because different parts depreciate at different rates, and no single figure or ' average ' can be accurate. Even if the plant and machinery are divided into groups or each individual item is taken separately errors may still occur, as it is by no means easy to find an accurate rate of depreciation for some appliances. A common device is to divide the plant and machinery into five or more groups, make a separate valuation of each group, and to adjust this by a revaluation by an expert every five years. This is probably the best compromise between accuracy and cheapness.

A distinction must be made between the first and second method for they do not necessarily yield the same results.

Revaluation by a professional valuer is so expensive that it can seldom be done annually, though a yearly valuation of some of the machinery and tools may best be made by this means.

The last method (renewals) is only satisfactory for relatively small appliances costing not more than £50 and requiring to be replaced every few years (see p. 1021).

***Fire Insurance and Depreciation.*—Whilst it is a sound, general policy to write as much off for depreciation as possible, such a course has an adverse effect on insurance against fire, as the value in the case of a fire would be affected in the claim for replacement. It is usual in some offices to append a declaration on the face of the policy that the depreciation written off in the books shall not be taken for the purpose of assessing a fire loss. Usually the assured is covered for the value of the plant at the time of the fire, and that the original cost of the plant, less depreciation, is not necessarily the basis of the claim. There appears to be no scale of depreciation in force for different kinds of plant recognised by the insurance companies, but these losses are generally settled as between the assessors and surveyors acting for both parties, or, failing their agreement, by arbitration : no special rates are applicable, but each case is settled on its merits. The principal factor appears to be the possession of definite and specific information on the value and cost of the plant, and the rate of depreciation should not in any way govern the claim in case of fire.**

It is a general provision in fire insurances and policies that the company may elect to replace the plant ; instances of this have rarely occurred, if ever, and probably the replacement clause was never intended as other than a protection to the insurance company.

***Depreciation and Income-Tax.*—The rate of depreciation allowed by the inspectors appointed under the Income-tax Acts differs considerably in different localities. For instance, the rate of depreciation allowed to engineers in Leicester is 7½ per cent. on the full value, while in Cardiff the rate is only 5 per cent. on the diminished value. Hosiery machinery in Leicester obtains 7½ per cent. as against 5 per cent. on lace machinery in Nottingham. Dyers' machinery in Leicester can get 10½ per cent., if justified on inquiry. Generally speaking, a distinction is made between motive power and process plant, and the general practice assumes 5 per cent. on motive power and 7½ per cent. to 10 per cent. on process plant ; these allowances are in respect of both deterioration and renewals.**

The Association of British Chambers of Commerce in 1933 reached a series of Agreed Normal Rates of Depreciation Allowances with the Inland Revenue. Those relating to Engineering are shown below; all are on the *written down value*.

	Per cent.
Brass foundries general plant and machinery	7
Colliery surface plant (not electrical)	6¼
Colliery underground plant	10
Drop forgers and stampers general plant and machinery (excluding furnaces)	7½
Electric furnaces and plant and machinery used in connection, but excluding foundations, buildings, cranes, buckets, shop tools or equipment	12½
Electric light undertakings:	
Cables	3
Plant and machinery	5
Electric motors, dynamos, and other electric plant	7½
Engineers precision tools, manufacture of such as twist drills, milling cutters, reamers, tap dies and screwing tackle, but excluding engines, boilers, shafting, pulleys, and electrical plant	9
Engines, boilers, shafting	5
Lorries driven by internal combustion engines	20
Lorries (steam)	15
Pig iron and steel manufactures, on general plant and machinery, but excluding furnace structures	7½
Railway wagons owned by traders	6¼
Refrigerating machinery, compressors, condensers, ice tanks, coolers, conduits, moulds, coils, travellers, etc.	10
Saw mills (sawing plant only)	7½
Wrought iron industry general plant and machinery	7½

Where plant and machinery is used by night as well as by day no extra allowance for steam raising plant, but an additional 25 per cent. of the normal rates on all other plant run both day and night. No allowance for less than 24 hours per day.

Furnaces and kilns: no depreciation allowance, but the cost of repairs, renewals, and rebuilding are allowed against revenue. Extensions and enlargements to be charged against capital.

Lagging, belting, loose plant, and utensils to be dealt with by way of renewals.

The Inland Revenue authorities have now reverted, in a number of cases, to the basis of original cost, and where, say, 5 per cent. is allowed on original cost, there is an alternative rate of 7½ per cent. allowed on diminishing value.

On *Fixtures* no allowance for depreciation is made, unless they are rightly included under the head of plant. Any replacements are allowed in full against the year's working.

Renewals.—The practice adopted by the income-tax authorities provides for the charging up of renewals to capital account where depreciation is allowed, and they usually decline to allow both renewals and depreciation out of the year's profits. They will, however, permit, as an alternative to the allowance for wear and tear and obsolescence of plant and machinery, the cost of renewing plant and machinery, under Schedule D. When this course is adopted, the amount to be allowed is the actual cost of the new plant and machinery (excluding any part of such cost which is attributable to additions or improvements, *i.e.* to an increase in capital) after deducting the scrap value or realised price of the plant and machinery replaced.

Example (a).—A machine which originally cost 1,000*l.* is worn out and replaced by a machine of similar power or size or capacity which now costs 1,500*l.* The whole of this expense of 1,500*l.* less, say, 100*l.*, the scrap value of the old machine (making a net amount of 1,400*l.*), is allowable from the profits of the year in which it is incurred.

Example (b).—A machine which originally cost 1,000*l.* is worn out and replaced by one of greater power or size or capacity costing 2,500*l.* The amount to be allowed as an expense is in this case not the full 2,500*l.*, but only the cost of replacing the old machine by one of similar power of capacity, say, 1,500*l.*, less the scrap value of the old machine.

Although this method of allowance is alternative to the wear and tear allowance for the same class of plant, the two principles may run concurrently for different classes of assets in the same business. For example, the wear and tear allowance may be applied to fixed machinery, and the renewal method used for loose plant.

Example (c).—Assuming the total written down value of the trader's plant and machinery before the sale of the obsolete machine to be 10,000*l.*, the new machine to cost 1,400*l.* and the old one to be sold for 400*l.*, future wear and tear allowances will be made upon a value of 11,000*l.*, thus:—

	£
Written down value of total plant, etc.	10,000
Deduct written down value of the obsolete machine	400
	£9,600
Add cost of new machine	1,400
	£11,000

The *Wear and Tear Allowance* was formerly based on the average of the preceding three years' trading, but in 1927 this was stopped and since then only the preceding year is taken into account.

If the allowance for wear and tear exceeds the assessable profits, the excess may be carried to the following year's assessment. Thus:—

	£
Gross assessment, 1927–8	6,000
Wear and tear allowance, 6,500*l*., of which 6,000*l*. is allowed for 1927–8 and 500*l*. is carried forward	6,000
Net amount on which tax is payable for 1927–8	nil
Gross assessment, 1928–9	7,600
Wear and tear allowance, 6,800*l*. for 1928–9 plus the unexhausted balance carried forward for 1927–8, 500*l*.	7,300
Net amount on which tax is payable for 1928–9	£300

Another example of wear and tear allowance is as follows:—

Example.—A machine is purchased on May 10, 1949, for £200, sold on November 18, 1951, for £110 and replaced by another costing £224. The rate of depreciation allowable is 7½ per cent. per annum.

		£	£
1949–50.	Cost Price	200	
	Wear and Tear 7½%		15
	+ one fifth allowance		3
	Total allowance for year		18
1950–51.	Written-down Value	185	
	Wear and Tear 7½%		14
	+ one fifth allowance		3
	Total allowance for year		17
1951–52.	Written-down Value	171	
	Wear and Tear 7½% for 6 months		6
	+ one fifth allowance		1
	Total allowance for year £7		7
Claim.	Written-down Value	165	
	Less three one fifth allowances		7
	Less Sale Price		110
		117	£117
	Obsolescence Claim 1951–52	£48	

No allowance is made on the increased cost of the new machine.

A special loose-leaf book, showing all details of cost, written-down values, allowances and price obtained on sale should be kept up-to-date. It is best to have the particulars of each separate machine entered on a separate page. Such a book can be shown to H.M. Inspector of Taxes, if equired, and often facilitates the granting of the allowances claimed.

IMPORTANT.—It frequently occurs that machines wear out more rapidly or have, for other reasons, to be replaced sooner than the allowable rate of depreciation provides. For this reason, the rate of depreciation and obsolescence in the firm's accounts may, rightly, be much larger than those allowed by H.M. Inspector of Taxes. This is particularly the case when machines are worked continuously on two shifts or are subjected to any other cause of additional wear and tear.

AGREED RATES OF NORMAL DEPRECIATION ALLOWANCES.

The National Federation of Clay Industries in 1938 agreed to the following with the Income Tax Commissioners:—

	Per cent.
Kilns for bricks tiles, etc.*	8
(Cost of repairs and renewals of parts is allowed as an ' expense ')	
Chimneys (if an integral part of a kiln) *	10
If the kiln could be replaced without affecting the chimney, it is to be treated as a separate structure, it is to be excluded from the wear and tear or renewals computation	—
Dryers (either Tunnel Dryers or Sheds) are eligible for renewals allowances	—
Brickmaking machinery	5
Boilers	5
Mixing Machines	5
Steam Engines	5
Shafting	5
Electrical plant	7½
Crushing and grinding plant	7½

* Alternatively, an arrangement for renewals may be made.

All additions to and replacements of plant and machinery to be charged to capital *except* those renewals and replacements which do not destroy the identity of the machine (these being allowed as an expense).

An additional allowance of one-fifth under S. 18 of the Finance Act (or subsequent Acts) is applicable to all allowances for wear and tear.

Commercial Rates of Depreciation.

The rates of depreciation allowed by the Inland Revenue Department are much lower than those considered to be prudent by many valuers and other authorities. It is therefore wise to include in a firm's accounts allowances for depreciation, etc., which are suitable for that particular business and not to suppose that the allowances permitted by the Inland Revenue Department are sufficient for the best interests of the firm.

No single set of figures can apply to all cases, but the following will be found to be as safe as any such general figures can be. In each case other factors, such as the probable duration of the business, must be taken into account.

	Per cent.
Air compressing plant (whole)	10
Air compressors	5·5
Autoclave	8·5
Belting (driving), leather	20
" " rubber	25
Belting (conveyor), canvas	40
" " rubber	30
Boiler, Lancashire	6
" heating	8·5
" small	10
" water tube	7
Buildings, all wood, cheap construction	10
" all wood, well built	5
" brick and steel	3
" brick and wood	4
" concrete	5
" corrugated iron on wood frame, concrete floor	5
" " " " wood floor	7
" reinforced concrete	8·5
Chimneys, brick (or concrete)	3·5
" metal	10
Cranes	8·5
Crushers, jaw	15
" rolls	10
Dryers	6
" rotary	8
Dwellings rented to employees	5
Dynamos	7
Electrical distribution system (inside)	5·5
" " " (outside)	6·5
Electrical motors	8·5
Elevators, bucket	15
Engines, gas	7·5
" oil	5
" steam, high speed	8–9
" " low speed	7·5
Fans	7·5
Furnaces	12·5
Pulleys, steel	6
" wood	7
Pumps, centrifugal	7
" geared	5
" vacuum	8
Rolls, crushing	10
Screens	10
" for heavy stone	12½
Separators	10
Shafting, main line	5
Stokers, mechanical	7·5
Tanks	5–20
Transformers	7
Water mains	5

Note:—Some of these figures do not agree with those shown in the Table on p. 1021. The differences show the importance of considering local factors and of not adhering too rigidly to any general set of figures.

Depreciation Table for Plant and Machinery, Showing the Total Amount Written Off in a Given Number of Years if Depreciated at the Rate of 5 to 10 per Cent. per Annum on Diminishing Values.

Years	5 Per Cent.	6 Per Cent.	7 Per Cent.	8 Per Cent.	9 Per Cent.	10 Per Cent.
1	5·00	6·00	7·00	8·00	9·00	10·00
2	9·75	11·64	13·51	15·36	17·19	19·00
3	14·26	16·94	19·56	22·13	24·64	27·10
4	18·55	21·92	25·19	28·36	31·42	34·39
5	22·62	26·61	30·43	34·09	37·59	40·95
6	26·49	31·01	35·31	39·36	43·21	46·86
7	30·16	35·15	39·84	44·21	48·32	52·17
8	33·66	39·04	44·05	48·67	52·97	56·95
9	36·97	42·70	47·97	52·78	57·20	61·26
10	40·12	46·14	51·61	56·56	61·06	65·13
11	43·12	49·37	55·00	60·03	64·56	68·62
12	45·96	52·41	58·15	63·23	67·75	71·76
13	48·67	55·26	61·08	66·17	70·65	74·58
14	51·23	57·95	63·80	68·87	73·29	77·12
15	53·67	60·47	66·33	71·37	75·70	79·41
16	55·99	62·84	88·69	73·66	77·88	81·47
17	58·18	65·07	70·88	75·77	79·87	83·32
18	60·28	67·16	72·92	77·70	81·68	84·99
19	62·26	69·14	74·81	79·48	83·33	86·49
20	64·15	70·99	76·78	81·12	84·83	87·84
21	65·94	72·73	78·29	82·63	86·20	89·06
22	67·65	74·36	79·81	84·02	87·44	90·15
23	69·26	75·90	81·22	85·30	88·57	91·14
24	70·80	77·35	82·53	86·42	89·60	92·02
25	72·26	78·71	83·76	87·51	90·53	92·82
30	78·54	84·37	88·70	91·77	94·10	95·76
40	87·15	91·58	94·46	96·42	97·70	98·52

Special Allowances.

Special concessions are being arranged by the Inland Revenue Authorities, so as to enable **the burden of the increased income tax to be removed from industry. These take the** form **of increased allowances for depreciation. Particulars can be obtained from local** Inspector **of Taxes.**

Rating Allowances.

The changes effected by the recent Rating Acts are so numerous and so serious that very little guidance can be given in these pages. The soundest procedure is to employ a skilled valuer and an equally skilled accountant and, where practicable, to obtain a 'settlement by consent.'

To avoid unnecessary litigation in the form of appeals it is customary for such valuers to place a lower value on the assessed property than it would realise if sold to a willing buyer as part of a (successful) going concern.

It is increasingly becoming the custom to base rating assessments on the output or sales, and although this is not strictly legal it is often accepted as convenient to all concerned. Thus, a stone-quarry may be assessed : (*a*) on the unworked land in reserve, and (*b*) on the output or sales, the latter being assessed at 9*d*. per ton or some other figure. Similarly, a brickworks may be assessed in three parts : (*a*) the unworked land (held in reserve) on an agricultural or surface-rent ; (*b*) the buildings and plant at (say) £100 per annum as a nominal figure, and (*c*) the clay at (say) 9*d*. per thousand bricks produced. A further simplification is sometimes effected by combining (*b*) and (*c*) into a single figure, such as 1*s*. 0*d*. per thousand bricks produced.

Many industrial properties are now de-rated, but the calculations necessary in this connection cannot be explained briefly. The officials of the local Rating Authority will usually afford ample information, but they do not always interpret the law correctly as to whether a particular property or what parts of it should be de-rated.

Important Note.

The 1939–1945 war in Europe has caused so much abnormal wear and tear on buildings, plant and machinery used by many firms that it is necessary in many cases to take this specially into consideration. The calculation of *obsolescence* is similarly affected.

Patents.

The Income Tax Act, 1945, makes special provisions for writing off the value of Patents.

SECTION XLV

LEGAL NOTES FOR ENGINEERS.

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(pp. 1027 1036)

SECTION XLV

LEGAL NOTES FOR ENGINEERS.

(Contributed by Sir W. Valentine Ball, O.B E., M.A., Barrister-at-Law.)

(1) LEGAL STATUS OF AN ENGINEER. (2) FEES. (3) EMPLOYMENT OF AN ENGINEER. (i.) *Generally;* (ii.) *Employment Abroad:* (iii.) *Wrongful Dismissal;* (iv.) *The Restrictive Covenant;* (4) SECRET COMMISSIONS. (5) DISCLOSURE OF SECRETS. (6) RIGHTS OF AN ENGINEER TO THE INVENTIONS OF A SERVANT. (7) THE ENGINEER AS WITNESS. (i). *Generally;* (ii.) *Arrangements as to Fees;* (iii.) *Evidence before Parliamentary Committees.* (8) THE COURT EXPERT. (9) LIABILITY FOR NEGLIGENCE. (10) RIGHTS, POWERS, AND DUTIES OF AN ENGINEER UNDER A CONTRACT FOR WORKS. (i.) *Generally;* (ii.) *Specifications;* (iii.) *Plans and Drawings:* (iv.) *Extras;* (v.) *Certificates;* (vi.) *Penalties and Bonuses;* (vii.) *Defects, Repairs, and Maintenance;* (viii.) *Sub-contractors and Specialists,* (ix.) *Whether the Employer's Engineer can act as Arbitrator in Relation to Disputes Arising under a Contract.* (11) THE ENGINEER AS ARBITRATOR. (i.) *Generally;* (ii.) *Who may be Arbitrator:* (iii.) *Proceedings before the Arbitrator;* (iv.) *The Award;* (v.) *Time for Making the Award;* (vi.) *The Fees of an Arbitrator;* (vii.) *Stating a Case for the Opinion of the Court;* (viii.) *Costs of a Reference.* (12) NOTES ON SALE AND PURCHASE OF MACHINERY.

(1) LEGAL STATUS OF AN ENGINEER.

Any person may describe himself or practise as an engineer, as the word engineer is not a term of art in the legal sense. Although the Institution of Civil Engineers may make rules of professional etiquette to be observed by its members, compliance with such rules will not be enforced in a court of law, save in so far as they involve a restatement of the ordinary law. For instance a rule which states that no consulting engineer shall advertise could not be enforced, whereas infringement of a rule that 'he shall not pay by commission or otherwise anyone who introduces clients' might offend against the common law (*Panama Telegraph Co.* v. *India Rubber Co.* (1875) L. R. 10 Ch. 515) or against the Secret Commissions Act.

(2) FEES.

If an engineer is consulted, there is an implied contract to pay him reasonable remuneration (*Munson* v. *Baillie* (1855) 2 Mac. H. L. Cas. 80). It is more satisfactory, however, to provide for payment of fees beforehand. In the case of a contract to design and supervise erection of works it is well to provide for payment of ⅓ on signing the contract, ⅓ when half the contract price has been paid to the contractor, and ⅓ when the contractor has been fully paid. Provision should also be made for (1) payment for plans and drawings in case the works are abandoned; (2) travelling and hotel expenses. There is, of course, no fixed scale of remuneration for the engineer, but in deciding what is a reasonable charge, the Court would have regard to (i.) the standing of the engineer; (ii.) the nature and extent of the work undertaken and carried out. It is no answer to a claim for fees (*a*) that a particular undertaking has proved to be a failure if the failure was not due to any fault of the engineer—as where, for instance, an engineer is employed merely to *supervise* the erection of a dam which bursts on completion; or where an engineer is employed to draw plans for a newly invented machine which fails to work (*aliter* if the employer was relying on the engineer's skill and judgment: *Duncan* v. *Blundell* (1820) 3 Stark. N.P. 6); (*b*) that the work in question is not completed, if the failure to complete is not due to the engineer. Where an engineer is to be paid on completion and the work is abandoned before the time, the engineer is entitled to what is due for services rendered, and to what he has lost by not being allowed to complete the contract (*Horton* v. *Hemsley* (1908) *Times*, February 19).

(3) EMPLOYMENT OF AN ENGINEER.

(i) Generally.—Where an engineer is to be employed for more than a year certain, the agreement must be in writing; and this is so, even if the employment is terminable at any time by six months' notice on either side (*Hanau* v. *Ehrlich* (1912), A.C. 39). The document must set forth all the terms. It is safer, however, to have the terms of an agreement always put into writing. Special care should be taken to (*a*) specify the times when the salary is to be paid; (*b*) provide for the contingency of illness, as illness does not necessarily determine the contract, nor will it justify dismissal without regular notice, unless the employee is so smitten with disease that he can never be expected to work again (*Cuckson* v. *Stones* (1858) 1 E. & E. 248); (*c*) provide for the notice which may be given on either side to determine the contract. If nothing is specified in the agreement, an engineer employed at a yearly salary is entitled to a year's notice (*Buckingham* v. *Surrey and Hants Canal Co.* (1882) 46 L. T. 885). An engineer wrongfully quitting his employment without notice forfeits the salary due for the part of the current year which he has served. The fact that a contract of employment for an indefinite period contains no express term as to notice to be given for its determination does not exclude the implication that such a notice is required (*Payzu* v. *Hannaford*, 34 T. L. R. 442). The suspension of servant from his duties which prevents his earning his commission may amount to a wrongful dismissal (*Rubel* v. *Vos*, 34 T. L. R. 171). Dismissal without notice is only justifiable in a case of gross misconduct. If the 'employer' is a company or a municipal corporation or other local authority it is safer to insist that the contract be under seal. The Public Health Act, 1875, s. 174 (1) provides that every contract made by an urban authority whereof the value or amount exceeds 50*l*, shall be in writing and sealed with the common seal of such authority. A surveyor who prepared plans for a local board under a verbal contract was held entitled to recover nothing (*Hunt* v. *Wimbledon Local Board* (1878) L. R. 4 C. D. D. 48). Close examination of a contract between an engineer and a local authority may, however, show that it is not within this section (see *Douglass* v. *Rhyl U.D.C.* (1913) 2 Ch. 407). Where the engineer is in the employment of a local authority he must have no interest in any contract into which that body enters (see *Mellis* v. *Shirley Local Board* (1885) L. R. 14 Q.B.D. 911). An attempt was recently made to raise the question whether a consulting engineer is liable to pay excess profits duty under s. 39 of the Finance Act, 1915, but the Court of Chancery refused to decide the point, leaving it to the King's Bench Division (*Smeeton* v. *Attorney General* 35 T. L. R. 706).

(ii.) Employment Abroad.—Where an engineer is about to accept employment abroad, he should be careful to see that provision is made for the payment of travelling and other expenses. No agreement to pay travelling expenses would be implied from an undertaking, say, to employ an engineer at a salary of 1,000*l*. a year at Malacca as from January 1, 1933. It is also well to provide in the agreement that any disputes which may arise shall be determined either by arbitration or according to English law.

(iii.) Wrongful Dismissal.—If an engineer is wrongfully dismissed, it is his duty to reduce the damages as far as possible by taking other employment if he can procure it (see *Storey* v. *Fulham Steel Works* (1908) 23 T. L. R. 306). The damages are then ascertained by subtracting what he has earned from what he would have earned.

(iv.) The Restrictive Covenant.—When employing an assistant, an engineer may find it necessary to bind him not to set up close by in competition. A restrictive covenant is legal, if it is not too wide; it must be only what is reasonably necessary for the protection of the parties. A covenant to prevent a dentist practising within 200 miles of York was held to be too wide (*Horner* v. *Graves*, 7 Bing. 735). On the other hand, a covenant which bound a canvasser not to enter an employment 'within twenty-five miles of London where the plaintiffs carry on business' was held by the House of Lords to be unreasonable (*Provident Clothing and Supply Co.* v. *Mason* (1913) 29 T. L. R. 727). Care should be taken not to make the restriction perpetual unless that is absolutely necessary, for the Court criticises a perpetual covenant more closely (*Eastes* v. *Russ* (1915) 30 T. L. R. 237). In a recent case the validity of a covenant affecting a man who was employed as works engineer at a bottleworks was considered (*Forsters* v. *Suggett*, 35 T. L. R. 87).

(4) SECRET COMMISSIONS.

The receipt of a secret commission or a bribe by an engineer would clearly justify his dismissal (*Boston Deep Sea Fishing Co.* v. *Ansell* (1883) 39 Ch. D. 339). In such a case there is a presumption that he was influenced, against his employers' interest, by the bribe (see *Hovenden* v. *Millhoff* (1900) 83 L. T. 11).

(5) DISCLOSURE OF SECRETS.

An engineer may be employed to do work of a confidential nature for others; or he may have to employ workmen to do confidential work for him (*e.g.* by employing a workman to make a model of a new and patentable machine). The law implies an agreement by the servant to keep his master's secrets, and he may be restrained by injunction from disclosing them (*Alperton Rubber Co.* v. *Manning and Others* (1917) 116 L. T. 499; 33 T. L. R. 205).

(6) RIGHTS OF AN ENGINEER TO THE INVENTIONS OF A SERVANT.

If a servant invents a machine in his employer's time and even when using his employer's materials, the invention does not become the property of the employer so as to prevent the servant taking out a patent for it (*Heald's Case*, 8 R. P. C. 430). But if the servant were to steal an idea and endeavour to patent it either before or after the termination of the service, the employer could oppose the grant of a patent in respect of it (*Thwaites' Application*, 9 R. P. C. 515). And if the servant obtains a grant of a patent in respect of the invention, he can only hold it as trustee for his employers (*British Reinforced Concrete Engineering Co.* v. *Lind* (1917) 33 T. L. R. 170). The engineer when employing a servant should take care to provide, by the agreement, that the inventions of the servant shall belong to him; but when himself entering into a contract of service he should be careful to sign nothing which will abrogate his rights.

(7) THE ENGINEER AS WITNESS.

(I.) Generally.—As the engineer frequently comes into contact with the law and the lawyers in the capacity of a *witness*, a few notes of a practical character on the duties of a witness may be found useful.

When invited to give evidence as an expert the engineer should: (*a*) arrange for a qualifying fee to be paid in any event; (*b*) examine the plans and documents, and, if necessary, obtain a view of the *locus in quo* before consenting to give evidence; (*c*) warn the parties consulting him of all the difficulties which are manifest to him; (*d*) ascertain, if possible, the contentions which are going to be put forward by the other side, and make himself familiar with any views which they may have published upon the matters in dispute. Having consented to give evidence, the engineer should (*a*) be ready to satisfy the lawyers with regard to any technical point which may be raised in consultation; (*b*) listen to the arguments and evidence in Court, and be prepared to deal with any points made by witnesses on the other side; (*c*) bring with him any entries in his diary or any notes which he may have made *at the time* with reference to the matters in dispute, for he may use such notes to refresh his memory. Thus, if an engineer has to report as to the condition of a structure, he may use in the witness box the notes which he made on the spot. But he could not use a copy of such notes unless he was prepared to produce the original; (*d*) make and bring to court a plan or model of the structure or machine under discussion, as this will save much wordy explanation; (*e*) search in libraries and in the published proceedings of the various institutions for any books and papers dealing with the matters in hand—especially any papers written by himself, in order that he may not be confronted with an adverse opinion given on a former occasion. A witness should bear in mind that it is his duty to state facts and express opinions, not to advocate the cause. As soon as he appears to be taking a biased view or attempting to fight the case it is more than likely that the tribunal will pay little heed to his evidence. A candid answer to an awkward question is often less dangerous than a long and evasive statement. In *Crosfield and Sons, Ltd.* v. *Techno-Chemical Laboratories Ltd.* (1913, 29 T. L. R. 378) Neville, J., expressed some views as to the function of an expert witness which may be thus summarised:—

(*a*) It is not competent in any action for witnesses to express their opinion upon any of the issues, whether of law or fact, which the Court or a jury has to determine.

(*b*) In patent cases, expert evidence is often necessary (i) to explain words or terms of science or art, or to inform the Court in case the import of a word in a document to be construed differs from its popular meaning; (ii) to instruct the Court in the laws of science; (iii) to describe the state of public knowledge at the date of the patent. 'It is rare,' added his lordship, 'to find any substantial difference of opinion between eminent experts upon matters of science whenever it is possible to dissociate the question from immediate connection with the issues in the action.'

(II.) Arrangements as to Fees.—On being *subpœnaed* one is bound to attend as a witness provided sufficient conduct money is paid with the subpœna. Failure to attend might lead to an action for damages, while in the County Court the judge has power to fine a witness for non-attendance. A witness may refuse to give evidence until his expenses have been paid (*Newton* v. *Harland* (1840) 1 M. & G. 956). It should be noticed that the person liable to pay the fees is the party to the litigation, not the solicitor, unless he has expressly made himself liable (*Lee* v. *Everest* (1857) 26 L. J. Ex. 334). In order to save trouble it is best to arrange for and if possible secure payment of fees in advance. The fees 'allowed on taxation,' that is to say, the fees which an unsuccessful litigant will be ordered to pay in respect of witnesses called by the other side, are very small, but the party calling the witness will have to pay his agreed fee whatever it is. Where a case is tried at assizes in the town where the engineer lives or in London (when the engineer lives in London), the fee allowed is 1*l*. 1*s*. *per diem*, or where the engineer lives away from the town 1*l*. 1*s*. to 3*l*. 3*s*. But the taxing masters may also allow a qualifying fee of so much a day varying with the magnitude of the case. In 1875 a fee of 7*l*. 7*s*. was not considered excessive payment to a scientific witness. It is probable that a larger sum would be allowed at the present day. In addition he is allowed the travelling expenses reasonably and actually paid. In County Courts the fee allowed for qualifying to give evidence on Scale B is 1*l*. 1*s*. to 3*l*. 3*s*.; on Scale C, 1*l*. 1*s*. to 5*l*. 5*s*.; for attending Court on trial (*per diem*), Scale B, 1*l*. 1*s*. to 2*l*. 2*s*.; Scale C, 1*l*. 1*s*. to 3*l*. 3*s*.

The above scales were fixed before the war, and have not been altered. So far as they can the taxing masters will probably grant the maximum owing to high prices.

(iii) Evidence before Parliamentary Committees.—When an engineer is summoned to appear as a witness before a committee of either House of Parliament he looks for his fees to the party who summons him—whether he is the promoter or an opponent. According to May's 'Parliamentary Practice' (1906), p. 434, an engineer summoned as a witness before a committee should report himself to the committee clerk on his arrival in London, or he will not be allowed his expenses for residence prior to the day of making his report; *3l. 3s.* a day is the parliamentary allowance, but of course as between the engineer and the party summoning him the *quantum* of the fee is a matter of agreement.

(8) COURT EXPERT.

The Rules of the Supreme Court enable the Court to appoint what is called a 'Court Expert,' being a person whose status is intermediate between that of a 'witness' and an 'assessor.' He is appointed at the instance of any party to a suit, to report on any question of *fact*. His report is treated as information supplied to the Court, and the writer of it may be cross examined upon it. (For further information as to the Court Expert, see Order 37A of the Rules of the Supreme Court.) An engineer should consult his solicitor before accepting this *rôle*, because the position with regard to his claim for fees is not very satisfactory.

(9) LIABILITY FOR NEGLIGENCE.

The liability of an engineer for negligence is similar to that of any other professional man. To render him liable it is not enough that there has been a less degree of skill than some other professional man might have shown. A fair average degree of skill is all that can be insisted upon (*Lamphier* v. *Phipos* (1838) 8 Car. & P. 475). If an engineer were so careless in the preparation of a tender that the cost of the works was grossly underestimated, he would probably be liable to pay damages, and he certainly could not recover his fees (*Moneypenny* v. *Hartland* (1826) 2 C. & P. 379). An engineer should be acquainted with any local bye-laws appertaining to the work in hand (*Monks* v. *Dillon* (1884) 12 L. R. Ir. 321). On the principle of *respondent superior*, an engineer is responsible for the acts or defaults of assistants employed by him (*Mackersy* v. *Ramsays* (1842) 9 C. & F. 818). The question 'To whom is the engineer liable for negligence?' depends upon the conditions of employment. If he is engineer in relation to a large contract, he is not liable for anything done by him in his capacity as arbitrator between the employer and the contractor; but he is liable for damage resulting from negligence in supervision, *e.g.*, failure to examine and test work before it is hidden from view (*Leicester Guardians* v. *Trollope* (1911) 75 J. P. 197). The contractor, however, could not sue the engineer for an error in the specifications on the basis of which he made his tender. Where, however, the 'quantities' are prepared by the engineer, and his fees for their preparation are paid by the contractor, it is conceived that he could be sued for damages resulting from an error in those quantities (*Bolt* v. *Thomas*, 'The Law Affecting Engineers,' p. 58). If an engineer order extras without authority, the contractor may possibly sue him for damages for breach of warranty of authority (*Randall* v. *Trimen* (1856) 18 C.B. 786). The amount of damages recoverable for negligence is to be measured by the loss which the plaintiff has suffered. In *Saunders* v. *Broadstairs Local Board* (1890) 2 H. B. C. 160, two engineers claimed fees for preparing surveys, obtaining tenders, and superintending the execution of a contract for certain drainage works. The board counterclaimed for negligence and were awarded 2,640*l.*, the amount overpaid to the contractor by reason of the negligence of the engineers in over-certifying the quantities; 2,400*l.* estimated cost of doing bad work over again; 240*l.* actual cost of repairs already done to defective work. Damages recovered by a workman against his employer for an accident might also be recovered from the engineer if it were shown that the accident was caused by the engineer's negligence (*Mosdell* v. *Mitchell*, 'Times,' January 20, 1891).

(10) RIGHTS, POWERS, AND DUTIES OF AN ENGINEER UNDER A CONTRACT FOR WORKS.

(i.) Generally.—Common forms of contract for large works usually set forth the powers, etc., of the engineer very clearly. The preparation of the contract is generally left to a lawyer, and the engineer should be on his guard against allowing a common printed form to be applied indiscriminately to the construction of works of a special nature. When difficulties arise it may be found that the common form does not provide for their elucidation.

(ii.) Specifications.—The contract should contain a clause providing that the employer will not be responsible for any mistakes or inaccuracies in the specifications which form part of the contract. Such a clause will, in the absence of fraud, afford him protection (*Pearson* v. *Dublin Corporation* (1907) A. C. 357). There should also be a clause calling the would-be contractor's attention to the fact that he must make surveys and borings for himself before making his tender. In that case, if unforeseen difficulties are met with the employer will not be responsible (*Thorn* v. *Mayor of London* (1876) L. R. 1 A.C. 120). Again, where a man undertakes to make *e.g.* sound hydraulic cylinders according to a specification, capable of withstanding a certain pressure, he takes upon himself the risk of the cylinders being able to withstand the named pressure when made according to the specification (*Hydraulic Co.* v. *Spencer* (1886) 2 T. L. R. 554). Carelessness in the preparation of a specification may involve the employer in added expense, for which the engineer may be liable (*Moneypenny* v. *Hartland* (1828) 2 C. & P. 378).

(iii.) Plans and Drawings.—These form part of the contract, and may, in the absence of a disclaiming clause, operate as a warranty that dimensions, measurements, etc., are correct. A clause disclaiming responsibility for accuracy should therefore be inserted in the contract. The engineer has no implied authority to alter plans and drawings so as to increase or lessen the burden imposed upon the contractor (*Cooper* v. *Langdon* (1841) 9 M. & W. 60). Delay in supplying the necessary plans and drawings may excuse the contractor from carrying out the work within the contract time (*Roberts* v. *Bury Commissioners* (1869) L. R. 5 C. P. 310). It is for the engineer who prepares them to see that they conform to local bye-laws (*Cubitt* v. *Smith* (1864) 11 L. T. 298). Plans and drawings made for the purpose of carrying out works are the property of the employer—not of the engineer (*Gibbon* v. *Pease* (1905) 1 K. B. 810). As to copyright in plans, see the Copyright Act, 1911, s. 5.

(iv.) Extras.—The duty of deciding what are 'extras' within the meaning of the contract is usually entrusted to the engineer. He has no implied authority to order extras (*Cooper* v. *Langdon* (1841) 9 M. & W. 60), although an agreement to work *to the satisfaction* of an engineer has been held to give him implied authority to order extras (*Robertson* v. *Jarvie* (1910) 45 S. L. R. 260). It has been held, however, that where the plaintiffs were employed by an architect (whose functions, for this purpose, correspond to those of an engineer) to supply door handles and door furniture for a building, and shortly before the delivery of the goods he informed the defendants, who were the builders, that the goods were being supplied by the plaintiffs, the defendants who used the goods in the building were liable to pay for them (*Ramsden* v. *Chessum* (1914) 78 J. P. 49). Extras should be authorised in the manner provided by the contract, which usually provides that all extras shall be ordered in writing (*Russell* v. *Saida Baudeira* (1862) 13 C. B. N. S. 149); but, in the case of a contract with a local authority, which must be under seal, the order for extras need not be under seal (*Williams* v. *Barmouth Urban Council* (1897) 77 L. T. 383). If a written certificate is rendered necessary by the contract, nothing else will suffice; but in a recent case a contractor proceeded with extra work as required by an engineer without any written order. On his claiming payment, an arbitrator found that the engineer was wrong and unreasonable, and that the work and materials required by him were not included in the contract. After much litigation the House of Lords eventually held that this decision was correct (*Brodie* v. *Cardiff Corporation* (1919) 120 L. T. 417). Although the whole scheme of works cannot be altered by virtue of the extra clause (*R.* v. *Peto*, 1 Y. & J. 37) power to *vary* the work will give the engineer authority to substitute one piece of machinery for another of which it is well known to be the mechanical equivalent (*Stevens* v. *Mewes & Davies*, Court of Appeal, June 7, 1910; Emden's 'Building Contracts,' Supplement 1911). The engineer should therefore see that he has power to *vary* the work. The engineer's decision as to what are extras may or may not be subject to revision in accordance with the arbitration clause. So far as the contractor is concerned, the fact that the engineer has given his final certificate (see *sub tit. Certificates, post*) may involve a final decision as to what is and what is not an extra (*Connor* v. *Belfast Water Commissioners* (1871) 5 L. R. Ir. C. L. 55). Power to extend the time for completion is generally given to the engineer. It may be that owing to the action of the Government acting under the Defence of the Realm Acts, the performance of a contract for works becomes illegal during the continuance of the War. In such a case it is conceived that no extension of time by the engineer will avail to keep the contract alive (see *Metropolitan Water Board* v. *Dick, Kerr & Co.*, 34 T. L. R. 1111).

(v.) Certificates.—It is the duty of an engineer in relation to a contract for large works to grant or withhold the certificates upon which the contractor is paid. 'Progress certificates' are given from time to time as the work proceeds. They generally authorise payment to the extent of 80 per cent. upon the contract value of the work done from time to time. The remaining 20 per cent. is kept back usually, as to 15 per cent. until the final certificate is granted and as to 5 per cent. until the end of some stated period after the final certificate. This 20 per cent. is called the retention money (see *West Yorkshire Bank Ltd.* v. *Isherwood Bros.* (1912) 28 T. L. R. 593). The 'final certificate' is granted when the work is finished to the satisfaction of the engineer, and the contractor can sue for nothing until it has been granted (*Wallace* v. *Brandon and Byshottles U.D.C.* (1903) 2 H. B. C. 392). In deciding as to certificates the engineer acts as an arbitrator, and his decisions cannot be called in question (*Wadsworth* v. *Smith* (1871) L. R. 6 Q.B. 332); nor can he be compelled, and is better advised not to give reasons for his decision (*Stevenson* v. *Watson* (1879) L. R. 4 C. P. D. 148). No action lies against him for negligence in relation to certificates (*ib.*) (but see the case of *Goddard* v. *Ferguson*, *infra*). In exercising his discretion he must act as an arbitrator between employer and contractor (*Chambers* v. *Goldthorpe* (1901) 1 K.B. 624). It may, however, be that the decision of the engineer as to a certificate is subject to the arbitration clause, which has been held to be the case in relation to the form of contract approved by the Royal Institute of British Architects (*Robins* v. *Goddard* (1905) 1 K.B. 294). A fraudulent agreement between the employer and the engineer would however affect the validity of a certificate (*Kellett* v. *New Mills U.D.C.* (1900) 2 H. B. C. 230). The certificate of the engineer may be made a condition precedent to the contractor's right of action against the employer. If so an action cannot be maintained unless there has been improper dealing between the engineer and the employer (*Eaglesham* v. *McMaster* (1920) 2 K. B. 169). The statement that an engineer is not liable for negligence in relation to a certificate does not appear to be supported by the case of *Goddard* v. *Ferguson* (referred to by Mr. Saxon Snell in a paper read by him at the Royal Institute

of British Architects, April 15, 1912). It was there held that an architect who grants a certificate which is successfully challenged by an employer in an action brought against him by a builder to recover the amount so certified to be due to him, is liable to repay to the employer the costs incurred by him in resisting the builder's claim. The Official Referee held the architect liable on the ground that when acting as agent for the building owner in granting certificates, he must be held responsible for the reasonable and probable consequences of giving a certificate subsequently held to be inaccurate or excessive. Certificates should always be given in writing in a formal manner. No engineer should pass any contract which does not contain a provision to this effect. As to when an engineer is disqualified from granting certificates, see paragraph ix., *infra*. An engineer has no right to delegate the power to grant a certificate to a subordinate, *e.g.* to the resident engineer (*Freeman & Sons* v. *Middleton Electric Traction Co.*, per Hamilton J., July 15, 1910; Court of Appeal, January, 1911; Emden's 'Building Contracts,' Supplement, 1911).

(vi.) Penalties and Bonuses.—The question whether the contractor must pay penalties or not depends almost entirely upon the engineer. Penalties are imposed if the work is not carried out in accordance with the provisions of the contract as to time. The fact that extras are ordered will excuse the contractor from compliance with the conditions as to time (*Dodd* v. *Churton* (1897) 1 Q.B. 562), but the jurisdiction of the engineer does not extend to deciding this point (*Gallivan* v. *Killarney Urban Council* (1912) 2 Ir. R. 356). To avoid complication, however, it is best to insert a clause in the contract giving the engineer power to extend the time. With regard to the penalty clause generally nice questions sometimes arise as to whether it is capable of enforcement. If it is unreasonable on the ground that the punishment is too severe, the Court may refuse to enforce it; and it is for this reason that the words 'the sums to be regarded as liquidated damages and not as a penalty' are usually inserted after the sum named. The breach of an engineering contract involves damage of so uncertain a nature that the courts will generally treat the sum which the parties themselves have named as liquidated damages. Penalties will not be imposed if (*a*) the delay was caused by the act or default of the employer (*Thornhill* v. *Neats* (1860) 8 C. B. N. S. 831); (*b*) if there is delay in setting out the site or in delivering plans (*Roberts* v. *Bury Improvement Commissioners* (1870) L. R. 5 C. P. 310). The fact that the contractor has been delayed by a strike of his workmen, however, affords no excuse for delay (*Budgett* v. *Rimington & Co.* (1890) 25 Q. B. D. 320), nor, apparently, is the employer to be held liable for the acts of third parties causing delay (*Porter* v. *Tottenham U.C.* (1915) 1 K. B. 776). While a penalty clause is usually optional, it is obligatory in contracts made with certain local authorities (see Public Health Act (1875) s. 174 (2)). A bonus clause is sometimes inserted by virtue of which a contractor is enabled so much extra for expedition. If a contract contains such a clause, and the engineer has power to extend the time, it seems that his power to extend must be exercised not to enable the contractor to earn the bonus but only to avoid penalties (*Ware* v. *Lyttelton Harbour Board* (1882) 1 N. Z.R. 191).

(vii.) Defects, Repairs, and Maintenance.—In contracts relating to the erection of machinery clauses should be inserted providing for any defects which appear either before or after the issue of the final certificate. A useful form of this clause will be found in the printed conditions of contract issued by the Institution of Electrical Engineers. Its importance will be recognised when it is pointed out that when it is absent the final certificate of the engineer may possibly bar any further claim by the employer (*Lord Bateman* v. *Thompson* (1875) 2 H. B. C. 23). The time for the remedy of defects should be prolonged as much as possible inasmuch as the very fact of there being a defects clause impliedly releases the contractor from liability for defects appearing after the stated time (*Sharp* v. *Great Western Railway Co.* (1841) 11 L. J. Ex. 17). The contractor can only be called upon to repair and maintain when compelled to do so by a repairing clause and a maintenance clause. The employer must give notice of want of repair. Maintenance should be provided for in cases where the machinery or work is of a special character which is best known to the contractor. In the absence of any clause relating to defects and maintenance, it does not follow that on payment of the full price the employer thereby waives all claim for damages for work improperly done. It is a question of fact having regard to all the circumstances of the case whether the settlement was final and complete.

(viii.) Subcontractors and Specialists.—Most modern contracts involve the employment of subcontractors or specialists. A clause should be inserted in the head contract prohibiting the employment of any subcontractor or specialist unless approved by the engineer. In the absence of such a clause, the contractor might, possibly, farm out portions of the work to all and sundry (*British Wagon Co.* v. *Lea* (1880) 5 Q. B. D. 149). The engineer, apparently, has no implied authority to employ subcontractors (*Cowan* v. *Goderich* (1859) 10 U. C. C. P. 87). It is often well to specify the names of manufacturers of the raw materials which are to be used in the body of the contract. In order to preserve his right to supervise the work of any subcontractor or specialist, the engineer should secure the insertion in the head contract of a clause providing that payments to subcontractors shall be made by the contractor only upon the certificates of the engineer. It is also well to secure the insertion of a clause to the effect that, in the event of the contractor becoming insolvent, the subcontractor or specialist may be paid directly by the employer. All responsibility for delay or negligence on the part of a subcontractor is generally, by a special clause, placed upon the contractor (see, generally, *Crittall* v. *L. C. C.* (1910) 75 J. P. 203; *Hampton* v. *Glamorgan County Council* (1916) 115 L. T. R. 426, and the case of *Ramsden* v. *Chessum*, cited, *supra*, s. (9) (v.) i.)).

(ix.) Whether the Employer's Engineer can act as Arbitrator in Relation to Disputes Arising under a Contract.—The position of the engineer in relation to a large contract has been rendered very difficult by certain recent decisions of the courts. In theory he is agent for the employer in the preparation of plans and specifications and in the supervision of the work, and he is arbitrator or *quasi* arbitrator between the employer and contractor in granting certificates, approving work done, granting extensions of time, etc. Sometimes he is made sole arbitrator to decide all disputes between the employer and the contractor, and a serious question has arisen in several recent cases as to how far a contractor can be forced to accept his decision whether in the matter of granting certificates, etc., or as arbitrator. In *Jackson* v. *Barry Railway Co.* (1893) 1 Ch. 238, it was held that the fact of the employer's engineer having expressed a view of the matter in dispute hostile to the contractor was not a sufficient justification for refusing to allow him to act as arbitrator in relation to that dispute. The Court of Appeal pointed out that the contractors had agreed to this form of reference, and what they relied on was the professional honour, practice, and intelligence of the engineer. There was nothing in the circumstances of the case to render it improbable that he would not arrive at an honest decision.

In *Ives* v. *Willans* (1894, 2 Ch. 478) the same result was arrived at, although in giving judgment Lord Justice Davey said: 'No doubt in a sense the engineer will be the judge of his own conduct, and no doubt that is a position which *prima facie* raises some surprise in a judicial mind; but that is the contract of the parties.' As already indicated later decisions appear to have practically altered the law, although the judges always took care to say that *Ives* v. *Willans* is still good law. The result of the recent cases may be thus summarised. A contractor will not be bound to submit to the arbitrament of the employer's engineer (*a*) if the matter in issue is an unseemly personal dispute raising a vindictive feeling between the engineer and the contractor, and the engineer has so strongly expressed his view that it amounts to a prejudgment (*Nuttall* v. *Manchester Corporation* (1893) 8 T. L. R. 513); (*b*) if the nature of the dispute is such that the cross-examination of the engineer is essential (*Freeman* v. *Chester R. D. C.* (1911) 1 K.B. 783); (*c*) if the matter in issue is something outside the original agreement, *e.g.*: a dispute as to whether the engineer and contractor had agreed to vary the original agreement (*Aird* v. *Bristol Corporation* (1913) A. C. 241); (*d*) if the conduct of the engineer himself is practically the only point in dispute (*Blackwell* v. *Derby Corporation* (1909) 7 J. P. 129). Finally, it was held in *Roberts* v. *Hickman* (1913) A. C. 229 that the grant of a certificate by an architect will not be a condition precedent to the builder's right to sue if, by writing letters to the employer, and allowing himself to be influenced by their views, the architect has disqualified himself from acting judicially although guilty of no fraud or improper conduct.

If the method of arbitration provided for in many existing forms of contract is declared to be impossible for any of the above reasons the only alternative is litigation, a method of settling disputes which employers, contractors, and engineers always seem to desire to avoid. What can be done to avoid such a calamity in relation to future contracts? The following courses are open. Insert a clause (*a*) referring all disputes to an arbitrator to be mutually agreed or to be nominated by the president of one of the Institutes (see, *e.g.*, the form approved by the Institution of Electrical Engineers, clause 47); or (*b*) referring disputes to a *named* arbitrator; or (*c*) refer disputes to the president of one of the Institutes whoever he may be. The first of these suggestions appears to offer the best practical solution of the difficulty. The writer, however, ventures to make the following suggestion which might possibly work in practice. Frame the arbitration clause so as to leave the decision of every question arising under the contract to the engineer but subject to a proviso that should it appear to the Court on an application to stay any action or proceeding brought in relation to the contract that the engineer is disqualified from acting as arbitrator the matters in dispute will be referred to an engineer to be agreed between the parties, or failing agreement to be nominated by the president of the Institution of Civil Engineers. Another alternative is suggested by Mr. E. J. Rimmer in his Arbitration Clause in 'Engineering Contracts' (Constable & Co.). It is that a proviso should be added to the arbitration clause stating that if any action is brought the defendant may either (*a*) have the action stayed or (*b*) have the matter referred to an independent arbitrator appointed in one or other of the manners above suggested.

(11) THE ENGINEER AS ARBITRATOR.*

(i.) Generally.—Engineers are frequently called upon to act as arbitrators to decide disputes of a technical nature wholly unconnected with any contract in which the engineer has been employed. Arbitrations are conducted in accordance with the provisions of the Arbitration Act, 1889. An arbitrator may be appointed (*a*) by the Court; (*b*) by the terms of the contract itself; or (*c*) by a person nominated for that purpose in the contract, *e.g.* the president of the Institution

* The Institute of Arbitrators, 10 Norfolk Street, London, W.C. 2, is always willing to suggest the names of qualified arbitrators with special technical knowledge. Reference to this body, therefore, tends to ensure that the arbitrator has the requisite technical knowledge as well as the legal knowledge and experience, which are so essential to the satisfactory conduct of an arbitration.

of Civil Engineers. As to appointment by the Court, this occasionally takes place when a prolonged scientific enquiry is necessary. A 'referee' of this kind may have power either to wholly dispose of the dispute or make an enquiry and report to the Court, in which latter case the Court itself gives the judgment after considering the report. As references of this kind are rare in practice they will not be further dealt with.

As a rule the engineer finds himself made arbitrator pursuant to the arbitration clause in an agreement. Any dispute arising under any contract may thus be referred, and the parties will as a general rule be bound to proceed to arbitration for the purpose of settling a dispute instead of bringing an action at law. Where a contract contains an arbitration clause, if one party commences an action, the other may apply to the Court to stay the action pending a reference, and the action will generally be stayed, unless there is some good reason for not allowing the arbitration to proceed. For instance a contractor alleged that he had been induced by fraudulent representations as to the nature of the soil to enter into a particular sewerage contract, and brought an action for fraudulent misrepresentation. It was held that such an action ought not to be sent to arbitration (*Monro* v. *Bognor Urban District Council* (1915) 84 L. J. K. B. 1091). But the Court will stay an action, and allow an arbitration to proceed, although the question at issue is one which might be more suitably tried in a court of law—*e.g.* the construction of a contract (*Metropolitan Tunnel, etc.* v. *London Electric Ry.* (1926) 1 Ch. 371). (See also par. 9 (ix.), *supra.*)

(ii.) Who may be Arbitrator.—Any person upon whom the parties agree may act as arbitrator. Thus if in a contract with a local authority it is specified that the borough engineer shall be arbitrator there is no objection to his so acting, and the contractor can be compelled to accept his award (*Ives* v. *Willins*, *supra*, 2 Ch. 478). That is the broad rule, but the Court, as has been shown, always has power to refuse to enforce the arbitration clause in a case where it is obvious that the arbitrator ought not to act owing to bias, etc. Frequently the clause provides for the appointment of an arbitrator by each party with power to the arbitrators in case of differences to appoint an umpire.* When this is done the umpire and arbitrators generally find it convenient to sit together to decide the case.

(iii.) Proceedings before the Arbitrator.—Witnesses must be sworn in the usual way. Due notice must be given to either party of the time when it is proposed to hold the sitting. In a case of great complexity it is well to have a preliminary sitting in order to require the parties to formulate the points in difference between them. By this means the issues may be considerably narrowed. The arbitrator should hear both sides, who may be represented by solicitor or counsel. He must never hold private communication with one party on the subject matter of the reference. He has no power to examine witnesses in the absence of the parties. An arbitrator who takes evidence in the absence of, and without notice to, a party, is guilty of legal misconduct, for which his award may be set aside (*In re O'Connor*, 88 L. J. K. B. 1242; *Ramsden* v. *Jacobs* (1922) 1 K. B. 640). It is for the arbitrator to decide every question whether of law or fact. He may, however, consult a solicitor or, for that matter, any other expert on legal points. Indeed the writer was recently concerned in a case in which, with the consent of the parties, the arbitrator had his solicitor sitting by his side. He should take full notes of the evidence, unless, as is frequently the case, the parties agree to have a shorthand note.

(iv.) The Award.—If the case is complex, the engineer is well advised to consult his solicitor as to the form of the award. The following are the features of a valid award, which should be in writing. (*a*) *It must not exceed the submission.* Thus if the question submitted was whether the stone used for building a dam was according to specification, the arbitrator would have no power to award that the stone used was better than as specified and that the employer must pay for it. (*b*) *It must extend to all matters in dispute.* If several distinct questions are referred and the arbitrator omits to decide upon any one of them, the whole award is bad. If in the case mentioned he were asked to decide (i.) was the stone as specified; (ii.) was it suitable for the purpose, he would have to decide both questions. (*c*) *It must be certain.* The award must be so framed that its meaning is clear to the parties; but according to a legal maxim, that is certain which is capable of being rendered certain. Thus to award that A. and B. shall pay a debt in the proportion in which A. and B. hold shares in a ship is a sufficiently definite award (*Watson* v. *Watson* (1671) Sty. 28). (*d*) *It must be final.* The award must peremptorily state the arbitrator's finding. A mere expression of opinion that A. ought to pay B. such sum as C. thinks proper would not be a final award. Finally (*e*) *it must not be impossible, unreasonable, inconsistent*, or *illegal.* An award which directed that one party should do something illegal, *e.g.* trespass upon the land of another, would be invalid.

(v.) Time for Making an Award.—Formerly an award had to be made within three months of the date of the submission. Now, however, by virtue of the Arbitration Act, 1934, s. 6 (3), an award may be made at any time. But observe that an arbitrator who is *dilatory* may be removed by the Court; so that he should do his work with all reasonable dispatch.

* An engineer who is appointed by one party to a dispute, the other party appointing another arbitrator, will usually find it advisable to nominate a Fellow of the Institute of Arbitrators with special knowledge of the subject in dispute as the umpire.

(vi.) The Fees of an Arbitrator.—The arbitrator should arrange his fee beforehand. It is best fixed at so much *per diem* for the hearing. If no preliminary arrangement is made, he is entitled to charge what is fair and reasonable, and he can keep back his award until the fees are paid (*Ponsford* v. *Swain*, J. & H. 433). If the arbitrators keep back his award on such a ground, the Court may order that the award be delivered, on payment of the amount of fees claimed into Court. The fees can then be 'taxed' by a taxing master. If they are excessive, the 'excess' is paid out to the party paying in; and the balance to the arbitrator, who, by the way, has the right to be present at the taxation. When deciding in the award as to costs he should state who is to pay his fees; but he may insist upon their being paid by the party who takes up the award. He should not reveal his decision until after these formalities have been observed. Sometimes the *quantum* of an arbitrator's fees is left to the decision of a taxing master. On such a taxation, if the evidence goes to show that, in the opinion of persons in the same profession as the arbitrator, that his charges are fair and reasonable, the taxing master must allow them. In *Mason* v. *Lovatt* (1907, 23 T. L. R. 486) a quantity surveyor was allowed to charge 527*l*. 10*s*. for a reference which lasted 22 days, the arbitrator subsequently taking 13 days to consider his opinion. In *Llandrindod Wells Water Co.* v. *Hawksley* (1904, 20 T. L. R. 241) the fees of an umpire and two arbitrators all being engineers were allowed at 476*l*. 12*s*., in respect of a reference which lasted 2 days. The arbitrator should state how his fees are made up.

(vii.) Stating a Case for the Opinion of the Court.—Where a specific question is referred to an arbitrator, and he decides it, the fact that his decision is erroneous in point of law does not make the award bad on its face so as to admit of its being set aside (*In re King and Duveen* (1913) 2 K. B. 32); nor will the mere fact that the dispute involves questions of law induce the Court to stay a reference to arbitration (*Rowe* v. *Grossley* (1913) 108 L. T. 11). But where a point of law arises during a reference, the arbitrator, if he does not care to decide it, may state a case for the opinion of the Court (Arbitration Act, 1889, s. 19), or the Court may compel him to state a case (*ib.*). He can also make his award in the form of a special case, leaving it to the Court to decide how the judgment shall be entered. Refusal to state a case may constitute misconduct on the part of an arbitrator (*Czarnikow* v. *Roth Schmidt & Co.* (1922) 38 T. L. R. 595).

(viii.) Miscellaneous Powers of an Arbitrator.—By virtue of the Arbitration Act, 1934, an arbitrator may order security for costs; 'discovery of documents'; the examination of witnesses at home or abroad (where (*e.g.*) he is not able to take such examination); and the detention (for safe custody) of property in dispute. He may also order samples to be taken or observations or experiments to be made with a view to obtaining information or evidence.

(ix.) Costs of a Reference.—On the question of costs, and as to the form of that part of his award which relates to costs, an arbitrator should consult his solicitor, because (*a*) highly technical questions arise with regard to costs and their taxation which no layman can be expected to understand, and (*b*) it is desirable that in this matter an arbitrator should observe as nearly as possible the principles which are recognised in courts of justice. Again, the defendant in the proceedings may have preferred a counter-claim in which he may have been successful wholly or in part. This further complicates the question of costs. It may be observed that there is no objection to an arbitrator employing a lawyer to advise him in the matter of costs or with reference to any other question which arises in a reference. The importance of taking this course is that a layman has not, and in the nature of things cannot have, the same mental outlook as a court of justice in dealing with the question of costs. Let a recent example suffice. An arbitrator made an award of £200, but as the plaintiff had claimed £1,200 and only succeeded as to £200, he made each side pay its own costs. In the result the plaintiff got less than nothing at all, as his costs far exceeded £200. Such a decision as to costs was, from a lawyer's view-point, entirely wrong. The arbitrator wholly overlooked the fact that the plaintiff was compelled to sue to recover anything at all. In any event, and whether a solicitor is employed or not, the question of costs should be carefully considered, as the expenses of a reference are often wholly out of proportion to the sum in dispute. In litigation the general rule is 'costs follow the event'—an application of the maxim *væ victis*. Unless the submission contains some special term as to costs, the arbitrator has full power to decide who is to pay them, and he should exercise the power; otherwise the parties may be put to the expense of applying to him for an order as to costs. (See Arbitration Act, 1934, s. 11 (2.). He may award costs to be taxed 'as between solicitor and client'—which means that the successful party is nearly indemnified—or he may award costs to be taxed as between litigants in the ordinary way. He may also, in his discretion, award a fixed sum for costs (Arbitration Act, 1889, Schedule I. (i)), or he may declare that each party shall pay his own costs. In deciding whether to make a defendant pay costs, the arbitrator should (it is submitted) consider the nature of any offer made before the proceedings were commenced. Thus supposing a contractor claimed 5,000*l*. and the building owner made a firm offer of 4,000*l*. which was refused. After hearing the evidence, the arbitrator awards 4,000*l*. or less. In such a case he would be justified in disallowing the contractor any costs. An arbitrator can only make a successful plaintiff pay costs in exceptional circumstances as when (*e.g.*) it appears that although successful he should not have prosecuted his claim. It was held, however, in a case arising under the Small Holdings Act, 1908, the arbitrator in the exercise of his powers under that Act had wider powers in the matter of costs than a High Court judge (*Gray* v. *Baron Ashburton*, 115 L. T. R. 729).

(12) NOTES ON SALE AND PURCHASE OF MACHINERY.

The question which most frequently arises on the sale of machinery, motor cars, etc., is what is the warranty of fitness? Sometimes, of course, there is an express written warranty, but in the absence of that the matter is regulated by section 14 of the Sale of Goods Act, 1893, which provides (in effect) that there is no implied warranty or condition as to the quality or fitness for any particular purpose of goods supplied under a contract of sale, except (1) where the buyer expressly makes known the purpose for which the goods are required so as to show that the buyer relies on the seller's skill or judgment and the goods are of a description which it is in the course of the seller's business to supply (whether he be the manufacturer or not) there is an implied condition that the goods shall be reasonably fit for such purpose; provided that in the case of the sale of a specified article under its patent or other trade name, there is no implied condition as to its fitness for any particular purpose; (2) where the goods are bought by description from a seller who deals in goods of that description (whether he be the manufacturer or not) there is an implied condition that the goods shall be of merchantable quality; provided that if the buyer has examined the goods there shall be no implied condition as regards defects which such examination ought to have revealed; (3) an implied warranty or condition as to quality or fitness for a particular purpose may be annexed by the usage of trade; (4) an express warranty or condition does not negative a warranty or condition implied by the Sale of Goods Act unless inconsistent therewith. The result of these provisions is that in certain cases the vendor is made liable for latent defects, that is to say, for defects which would not be discoverable on an ordinary examination of the article. Sometimes a contract for the sale of machinery contains provisions for payment of the price by instalments after production of the certificate of the engineer that such instalments are due and payable. It was held in a Scotch case (*Howden* v. *Powell Duffryn Steam Coal Co.* (1912), S. C. 920, Ct. of Sess.) that where, in such circumstances, part of the machinery was rejected, but the manufacturers refused to accept the rejection, the production of the engineer's certificate was not a condition precedent to the right to recover payment.

In the case of an engine which is attached to the freehold, a question may arise, on a sale, as to when the property passes. In *Underwood Ltd.* v. *Burgh Castle Brick and Cement Syndicate* (1922, 1 K. B. 343), a horizontal condensing engine was sold at a price free on rail, London. It was bolted to and embedded in a flooring of concrete. Before it could be delivered on rail it had to be detached and dismantled. The sellers detached it, but in loading it on a truck they damaged it by accident so that the buyers refused to accept it. In an action by the sellers for goods bargained and sold it was held that the property in the engine had not passed to the buyers, inasmuch as the sellers had failed to do something necessary to put it in a deliverable state. Consequently the loss fell on the sellers and not on the buyers.

SECTION XLVI

MATHEMATICS

CIRCUMFERENCE AND AREA OF CIRCLES — TRIGONOMETRICAL TABLES — SQUARES — CUBES — ROOTS — RECIPROCALS—LOGARITHMS.

(pp. 1039–1090)

TABLE I.—Diameters advancing by Units.*

Diameter.	Circum.	Area.	Diameter.	Circum.	Area.	Diameter.	Circum.	Area.	Diameter.	Circum.	Area.
1	3·1416	0·7854	61	191·64	2922·5	121	380·13	11499	181	568·63	25730
2	6·2832	3·1416	62	194·78	3019·1	122	383·27	11690	182	571·77	26016
3	9·4248	7·0686	63	197·92	3117·2	123	386·42	11882	183	574·91	26302
4	12·566	12·5664	64	201·06	3217·0	124	389·56	12076	184	578·05	26590
5	15·708	19·6350	65	204·20	3318·3	125	392·70	12272	185	581·19	26880
6	18·850	28·2743	66	207·35	3421·2	126	395·84	12469	186	584·34	27172
7	21·991	38·4845	67	210·49	3525·7	127	398·98	12668	187	587·48	27465
8	25·133	50·2655	68	213·63	3631·7	128	402·12	12868	188	590·62	27759
9	28·274	63·6173	69	216·77	3739·3	129	405·27	13070	189	593·76	28055
10	31·416	78·54	70	219·91	3848·5	130	408·41	13273	190	596·90	28353
11	34·558	95·03	71	223·05	3959·2	131	411·55	13478	191	600·04	28652
12	37·699	113·10	72	226·19	4071·5	132	414·69	13685	192	603·19	28953
13	40·841	132·73	73	229·34	4185·4	133	417·83	13893	193	606·33	29255
14	43·982	153·94	74	232·48	4300·8	134	420·97	14103	194	609·47	29559
15	47·124	176·71	75	235·62	4417·9	135	424·12	14314	195	612·61	29865
16	50·265	201·06	76	238·76	4536·5	136	427·26	14527	196	615·75	30172
17	53·407	226·98	77	241·90	4656·6	137	430·40	14741	197	618·89	30481
18	56·549	254·47	78	245·04	4778·4	138	433·54	14957	198	622·04	30791
19	59·690	283·53	79	248·19	4901·7	139	436·68	15175	199	625·18	31103
20	62·832	314·16	80	251·33	5026·6	140	439·82	15394	200	628·32	31416
21	65·973	346·36	81	254·47	5153·0	141	442·96	15615	201	631·46	31731
22	69·115	380·13	82	257·61	5281·0	142	446·11	15837	202	634·60	32047
23	72·257	415·48	83	260·75	5410·6	143	449·25	16061	203	637·74	32365
24	75·398	452·39	84	263·89	5541·8	144	452·39	16286	204	640·89	32685
25	78·540	490·87	85	267·04	5674·5	145	455·53	16513	205	644·03	33006
26	81·681	530·93	86	270·18	5808·8	146	458·67	16742	206	647·17	33329
27	84·823	572·56	87	273·32	5944·7	147	461·81	16972	207	650·31	33654
28	87·965	615·75	88	276·46	6082·1	148	464·96	17203	208	653·45	33979
29	91·106	660·52	89	279·60	6221·1	149	468·10	17437	209	656·59	34307
30	94·248	706·86	90	282·74	6361·7	150	471·24	17671	210	659·73	34636
31	97·389	754·77	91	285·88	6503·9	151	474·38	17908	211	662·88	34967
32	100·53	804·25	92	289·03	6647·6	152	477·52	18146	212	666·02	35299
33	103·67	855·30	93	292·17	6792·9	153	480·66	18385	213	669·16	35633
34	106·81	907·92	94	295·31	6939·8	154	483·81	18627	214	672·30	35968
35	109·96	962·11	95	298·45	7088·2	155	486·95	18869	215	675·44	36305
36	113·10	1017·88	96	301·59	7238·2	156	490·09	19113	216	678·58	36644
37	116·24	1075·21	97	304·73	7389·8	157	493·23	19359	217	681·73	36984
38	119·38	1134·11	98	307·88	7543·0	158	496·37	19607	218	684·87	37325
39	122·52	1194·59	99	311·02	7697·7	159	499·51	19856	219	688·01	37668
40	125·66	1256·63	100	314·16	7854·0	160	502·65	20106	220	691·15	38013
41	128·81	1320·25	101	317·30	8011·9	161	505·80	20358	221	694·29	38360
42	131·95	1385·44	102	320·44	8171·3	162	508·94	20612	222	697·43	38708
43	135·09	1452·20	103	323·58	8332·3	163	512·08	20867	223	700·58	39057
44	138·23	1520·52	104	326·73	8494·9	164	515·22	21124	224	703·72	39408
45	141·37	1590·43	105	329·87	8659·0	165	518·36	21382	225	706·86	39761
46	144·51	1661·90	106	333·01	8824·7	166	521·50	21642	226	710·00	40115
47	147·65	1734·94	107	336·15	8992·0	167	524·65	21904	227	713·14	40471
48	150·80	1809·55	108	339·29	9160·9	168	527·79	22167	228	716·28	40828
49	153·94	1885·74	109	342·43	9331·3	169	530·93	22432	229	719·42	41187
50	157·08	1963·5	110	345·58	9503·3	170	534·07	22698	230	722·57	41548
51	160·22	2042·8	111	348·72	9676·9	171	537·21	22966	231	725·71	41910
52	163·36	2123·7	112	351·86	9852·0	172	540·35	23235	232	728·85	42273
53	166·50	2206·2	113	355·00	10028·8	173	543·50	23506	233	731·99	42638
54	169·65	2290·2	114	358·14	10207·0	174	546·64	23779	234	735·13	43005
55	172·79	2375·8	115	361·28	10386·9	175	549·78	24053	235	738·27	43374
56	175·93	2463·0	116	364·42	10568·3	176	552·92	24328	236	741·42	43744
57	179·07	2551·8	117	367·57	10751·3	177	556·06	24606	237	744·56	44115
58	182·21	2642·1	118	370·71	10935·9	178	559·20	24885	238	747·70	44488
59	185·35	2734·0	119	373·85	11122·0	179	562·35	25165	239	750·84	44863
60	188·50	2827·4	120	376·99	11310·0	180	565·49	25447	240	753·98	45239

* For Diameters advancing by 'Tenths' shift the decimal point one place to the left for Circumferences, and two places to the left for Areas.

EXAMPLE.—What is the circumference and the area of a circle whose diameter is 1·7 (i.e. 17 tenths)?
Circumference = 5·3407 Area = 2·2698.

TABLE I. (*continued*).

Diameter.	Circum.	Area.	Diameter.	Circum.	Area.	Diameter.	Circum.	Area.	Diameter.	Circum.	Area.
241	757·12	45617	305	958·19	73062	369	1159·25	106941	433	1360·31	147254
242	760·27	45996	306	961·33	73542	370	1162·39	107521	434	1363·45	147934
243	763·41	46377	307	964·47	74023	371	1165·53	108103	435	1366·59	148617
244	766·55	46759	308	967·61	74506	372	1168·67	108687	436	1369·73	149301
245	769·69	47144	309	970·75	74991	373	1171·81	109272	437	1372·88	149987
246	772·83	47529	310	973·89	75477	374	1174·96	109858	438	1376·02	150674
247	775·97	47916	311	977·04	75964	375	1178·10	110447	439	1379·16	151363
248	779·12	48305	312	980·18	76454	376	1181·24	111036	440	1382·30	152053
249	782·26	48695	313	983·32	76945	377	1184·38	111628	441	1385·44	152745
250	785·40	49087	314	986·46	77437	378	1187·52	112221	442	1388·58	153439
251	788·54	49481	315	989·60	77931	379	1190·66	112815	443	1391·73	154134
252	791·68	49876	316	992·74	78427	380	1193·81	113411	444	1394·87	154830
253	794·82	50273	317	995·88	78924	381	1196·95	114009	445	1398·01	155528
254	797·96	50671	318	999·03	79423	382	1200·09	114608	446	1401·15	156228
255	801·11	51071	319	1002·17	79923	383	1203·23	115209	447	1404·29	156930
256	804·25	51472	320	1005·31	80425	384	1206·37	115812	448	1407·43	157633
257	807·39	51875	321	1008·45	80928	385	1209·51	116416	449	1410·58	158337
258	810·53	52279	322	1011·59	81433	386	1212·65	117021	450	1413·72	159043
259	813·67	52685	323	1014·73	81940	387	1215·80	117628	451	1416·86	159751
260	816·81	53093	324	1017·88	82448	388	1218·94	118237	452	1420·00	160460
261	819·96	53502	325	1021·02	82958	389	1222·08	118847	453	1423·14	161171
262	823·10	53913	326	1024·16	83469	390	1225·22	119459	454	1426·28	161883
263	826·24	54325	327	1027·30	83982	391	1228·36	120072	455	1429·42	162597
264	829·38	54739	328	1030·44	84496	392	1231·50	120687	456	1432·57	163313
265	832·52	55155	329	1033·58	85012	393	1234·65	121304	457	1435·71	164030
266	835·66	55572	330	1036·73	85530	394	1237·79	121922	458	1438·85	164748
267	838·81	55990	331	1039·87	86049	395	1240·93	122542	459	1441·99	165468
268	841·95	56410	332	1043·01	86570	396	1244·07	123163	460	1445·13	166190
269	845·09	56832	333	1046·15	87092	397	1247·21	123786	461	1448·27	166914
270	848·23	57256	334	1049·29	87616	398	1250·35	124410	462	1451·42	167639
271	851·37	57680	335	1052·43	88141	399	1253·50	125036	463	1454·56	168365
272	854·51	58107	336	1055·58	88668	400	1256·64	125664	464	1457·70	169093
273	857·66	58535	337	1058·72	89197	401	1259·78	126293	465	1460·84	169823
274	860·80	58965	338	1061·86	89727	402	1262·92	126923	466	1463·98	170544
275	863·94	59396	339	1065·00	90259	403	1266·06	127556	467	1467·12	171287
276	867·08	59828	340	1068·14	90792	404	1269·20	128190	468	1470·27	172021
277	870·22	60263	341	1071·28	91327	405	1272·35	128825	469	1473·41	172757
278	873·36	60699	342	1074·42	91863	406	1275·49	129462	470	1476·55	173494
279	876·50	61136	343	1077·57	92401	407	1278·63	130100	471	1479·69	174234
280	879·65	61575	344	1080·71	92941	408	1281·77	130741	472	1482·83	174974
281	882·79	62016	345	1083·85	93482	409	1284·91	131382	473	1485·97	175716
282	885·93	62458	346	1086·99	94025	410	1288·05	132025	474	1489·11	176460
283	889·07	62902	347	1090·13	94569	411	1291·19	132670	475	1492·26	177205
284	892·21	63347	348	1093·27	95115	412	1294·34	133317	476	1495·40	177952
285	895·35	63794	349	1096·42	95662	413	1297·48	133965	477	1498·54	178701
286	898·50	64242	350	1099·56	96211	414	1300·62	134614	478	1501·68	179451
287	901·64	64692	351	1102·70	96762	415	1303·76	135265	479	1504·82	180203
288	904·78	65144	352	1105·84	97314	416	1306·90	135918	480	1507·96	180956
289	907·92	65597	353	1108·98	97868	417	1310·04	136572	481	1511·11	181711
290	911·06	66052	354	1112·12	98423	418	1313·19	137228	482	1514·25	182467
291	914·20	66508	355	1115·27	98980	419	1316·33	137885	483	1517·39	183225
292	917·35	66966	356	1118·41	99538	420	1319·47	138544	484	1520·53	183984
293	920·49	67426	357	1121·55	100098	421	1322·61	139205	485	1523·67	184745
294	923·63	67887	358	1124·69	100660	422	1325·75	139867	486	1526·81	185508
295	926·77	68349	359	1127·83	101223	423	1328·89	140531	487	1529·96	186272
296	929·91	68813	360	1130·97	101788	424	1332·04	141196	488	1533·10	187038
297	933·05	69279	361	1134·11	102354	425	1335·18	141863	489	1536·24	187805
298	936·19	69747	362	1137·26	102922	426	1338·32	142531	490	1539·38	188574
299	939·34	70215	363	1140·40	103491	427	1341·46	143201	491	1542·52	189345
300	942·48	70686	364	1143·54	104062	428	1344·60	143872	492	1545·66	190117
301	945·62	71158	365	1146·68	104635	429	1347·74	144545	493	1548·81	190890
302	948·76	71631	366	1149·82	105209	430	1350·88	145220	494	1551·95	191665
303	951·90	72107	367	1152·96	105785	431	1354·03	145896	495	1555·09	192442
304	955·04	72583	368	1156·11	106362	432	1357·17	146574	496	1558·23	193221

TABLE I. (*continued*).

Diameter.	Circum.	Area.	Diameter.	Circum.	Area.	Diameter.	Circum.	Area.	Diameter.	Circum.	Area.
497	1561·37	194000	561	1762·43	247181	625	1963·50	306796	689	2164·56	372845
498	1564·51	194782	562	1765·58	248063	626	1966·64	307779	690	2167·70	373928
499	1567·65	195565	563	1768·72	248947	627	1969·78	308763	691	2170·84	375013
500	1570·80	196350	564	1771·86	249832	628	1972·92	309748	692	2173·98	376099
501	1573·94	197136	565	1775·00	250719	629	1976·06	310736	693	2177·12	377187
502	1577·08	197923	566	1778·14	251607	630	1979·20	311725	694	2180·27	378276
503	1580·22	198713	567	1781·28	252497	631	1982·35	312715	695	2183·41	379367
504	1583·36	199504	568	1784·42	253388	632	1985·49	313707	696	2186·55	380459
505	1586·50	200296	569	1787·57	254281	633	1988·63	314700	697	2189·69	381554
506	1589·65	201090	570	1790·71	255176	634	1991·77	315696	698	2192·83	382649
507	1592·79	201886	571	1793·85	256072	635	1994·91	316692	699	2195·97	383746
508	1595·93	202683	572	1796·99	256970	636	1998·05	317690	700	2199·11	384845
509	1599·07	203482	573	1800·13	257869	637	2001·19	318690	701	2202·26	385945
510	1602·21	204282	574	1803·27	258770	638	2004·34	319692	702	2205·40	387047
511	1605·35	205084	575	1806·42	259672	639	2007·48	320695	703	2208·54	388151
512	1608·50	205887	576	1809·56	260576	640	2010·62	321699	704	2211·68	389256
513	1611·64	206692	577	1812·70	261482	641	2013·67	322705	705	2214·82	390363
514	1614·78	207499	578	1815·84	262389	642	2016·90	323713	706	2217·96	391471
515	1617·92	208307	579	1818·98	263298	643	2020·04	324722	707	2221·11	392580
516	1621·06	209117	580	1822·12	264208	644	2023·19	325733	708	2224·25	393692
517	1624·20	209928	581	1825·27	265120	645	2026·33	326745	709	2227·39	394805
518	1627·35	210741	582	1828·41	266033	646	2029·47	327759	710	2230·53	395919
519	1630·49	211556	583	1831·55	266948	647	2032·61	328775	711	2233·67	397035
520	1633·63	212372	584	1834·69	267865	648	2035·75	329792	712	2236·81	398153
521	1636·77	213189	585	1837·83	268783	649	2038·89	330810	713	2239·96	399272
522	1639·91	214008	586	1840·97	269702	650	2042·04	331831	714	2243·10	400393
523	1643·05	214829	587	1844·11	270624	651	2045·18	332853	715	2246·24	401515
524	1646·20	215651	588	1847·26	271547	652	2048·32	333876	716	2249·38	402639
525	1649·34	216475	589	1850·40	272471	653	2051·46	334901	717	2252·52	403765
526	1652·48	217301	590	1853·54	273397	654	2054·60	335927	718	2255·66	404892
527	1655·62	218128	591	1856·68	274325	655	2057·74	336955	719	2258·81	406020
528	1658·76	218956	592	1859·82	275254	656	2060·88	337985	720	2261·95	407150
529	1661·90	219787	593	1862·96	276184	657	2064·03	339016	721	2265·09	408282
530	1665·04	220618	594	1866·11	277117	658	2067·17	340049	722	2268·23	409416
531	1668·19	221452	595	1869·25	278051	659	2070·31	341083	723	2271·37	410550
532	1671·33	222287	596	1872·39	278986	660	2073·45	342119	724	2274·51	411687
533	1674·47	223123	597	1875·53	279923	661	2076·59	343157	725	2277·65	412825
534	1677·61	223961	598	1878·67	280862	662	2079·73	344196	726	2280·80	413965
535	1680·75	224801	599	1881·81	281802	663	2082·88	345237	727	2283·94	415106
536	1683·89	225642	600	1884·96	282743	664	2086·02	346279	728	2287·08	416248
537	1687·04	226484	601	1888·10	283687	665	2089·16	347323	729	2290·22	417393
538	1690·18	227329	602	1891·24	284631	666	2092·30	348368	730	2293·36	418539
539	1693·32	228175	603	1894·38	285578	667	2095·44	349415	731	2296·50	419686
540	1696·46	229022	604	1897·52	286526	668	2098·58	350464	732	2299·65	420835
541	1699·60	229871	605	1900·66	287475	669	2101·73	351514	733	2302·79	421986
542	1702·74	230722	606	1903·81	288426	670	2104·87	352565	734	2305·93	423139
543	1705·88	231574	607	1906·95	289379	671	2108·01	353618	735	2309·07	424292
544	1709·03	232428	608	1910·09	290333	672	2111·15	354673	736	2312·21	425447
545	1712·17	233283	609	1913·23	291289	673	2114·29	355730	737	2315·35	426604
546	1715·31	234140	610	1916·37	292247	674	2117·43	356788	738	2318·50	427762
547	1718·45	234998	611	1919·51	293206	675	2120·58	357847	739	2321·64	428922
548	1721·59	235858	612	1922·65	294166	676	2123·72	358908	740	2324·78	430084
549	1724·73	236720	613	1925·80	295128	677	2126·86	359971	741	2327·92	431247
550	1727·88	237583	614	1928·94	296092	678	2130·00	361035	742	2331·06	432412
551	1731·02	238448	615	1932·08	297057	679	2133·14	362101	743	2334·30	433578
552	1734·16	239314	616	1935·22	298024	680	2136·28	363168	744	2337·34	434746
553	1737·30	240182	617	1938·36	298992	681	2139·42	364237	745	2340·49	435916
554	1740·44	241051	618	1941·50	299962	682	2142·57	365308	746	2343·63	437087
555	1743·58	241922	619	1944·65	300934	683	2145·71	366380	747	2346·77	438259
556	1746·73	242795	620	1947·79	301907	684	2148·85	367453	748	2349·91	439433
557	1749·87	243669	621	1950·93	302882	685	2151·99	368528	749	2353·05	440609
558	1753·01	244545	622	1954·07	303858	686	2155·13	369605	750	2356·19	441786
559	1756·15	245422	623	1957·21	304836	687	2158·27	370684	751	2359·34	442965
560	1759·29	246301	624	1960·35	305815	688	2161·42	371764	752	2362·48	444146

TABLE I. (*continued*).

Diameter.	Circum.	Area.	Diameter.	Circum.	Area.	Diameter.	Circum.	Area.	Diameter.	Circum.	Area.
753	2365·62	445328	815	2560·40	521681	877	2755·18	604073	939	2949·96	692502
754	2368·76	446511	816	2563·54	522962	878	2758·32	605451	940	2953·10	693978
755	2371·90	447697	817	2566·68	524245	879	2761·46	606831	941	2956·24	695455
756	2375·04	448883	818	2569·82	525529	880	2764·60	608212	942	2959·38	696934
757	2378·19	450072	819	2572·96	526814	881	2767·74	609595	943	2962·52	698415
758	2381·33	451262	820	2576·11	528102	882	2770·88	610980	944	2965·66	699897
759	2384·47	452453	821	2579·25	529391	883	2774·03	612366	945	2968·81	701380
760	2387·61	453646	822	2582·39	530681	884	2777·17	613754	946	2971·95	702865
761	2390·75	454841	823	2585·53	531973	885	2780·31	615143	947	2975·09	704352
762	2393·89	456037	824	2588·67	533267	886	2783·45	616531	948	2978·23	705840
763	2397·04	457234	825	2591·81	534562	887	2786·59	617927	949	2981·37	707330
764	2400·18	458434	826	2594·96	535858	888	2789·73	619321	950	2984·51	708822
765	2403·32	459635	827	2598·10	537157	889	2792·88	620717	951	2987·65	710315
766	2406·46	460837	828	2601·24	538456	890	2796·02	622114	952	2990·80	711809
767	2409·60	462041	829	2604·38	539758	891	2799·16	623513	953	2993·94	713307
768	2412·74	463247	830	2607·52	541061	892	2802·30	624913	954	2997·08	714803
769	2415·88	464454	831	2610·66	542365	893	2805·44	626315	955	3000·22	716303
770	2419·03	465663	832	2613·81	543671	894	2808·58	627718	956	3003·36	717804
771	2422·17	466873	833	2616·95	544979	895	2811·73	629124	957	3006·50	719306
772	2425·31	468085	834	2620·09	546288	896	2814·87	630530	958	3009·65	720810
773	2428·45	469298	835	2623·23	547599	897	2818·01	631938	959	3012·79	722316
774	2431·59	470513	836	2626·37	548912	898	2821·15	633348	960	3015·93	723823
775	2434·73	471730	837	2629·51	550226	899	2824·29	634760	961	3019·07	725332
776	2437·88	472948	838	2632·65	551541	900	2827·43	636173	962	3022·21	7268[illegible]
777	2441·02	474168	839	2635·80	552858	901	2830·58	637587	963	3025·35	728354
778	2444·16	475389	840	2638·94	554177	902	2833·72	639003	964	3028·50	729867
779	2447·30	476612	841	2642·08	555497	903	2836·86	640121	965	3031·64	731382
780	2450·44	477836	842	2645·22	556819	904	2840·00	641840	966	3034·78	732899
781	2453·58	479062	843	2648·36	558142	905	2843·14	643261	967	3037·92	734417
782	2456·73	480290	844	2651·50	559467	906	2846·28	644683	968	3041·06	735937
783	2459·87	481519	845	2654·65	560794	907	2849·42	646107	969	3044·20	737458
784	2463·01	482750	846	2657·79	562122	908	2852·57	647533	970	3047·34	738981
785	2466·15	483982	847	2660·93	563452	909	2855·71	648960	971	3050·49	740506
786	2469·29	485216	848	2664·07	564783	910	2858·85	650388	972	3053·63	742032
787	2472·43	486451	849	2667·21	566116	911	2861·99	651818	973	3056·77	743559
788	2475·58	487688	850	2670·35	567450	912	2865·13	653250	974	3059·91	745088
789	2478·72	488927	851	2673·50	568786	913	2868·27	654684	975	3063·05	746619
790	2481·86	490167	852	2676·64	570124	914	2871·42	656118	976	3066·19	748151
791	2485·00	491409	853	2679·78	571463	915	2874·56	657555	977	3069·34	749685
792	2488·14	492652	854	2682·92	572803	916	2877·70	658993	978	3072·48	751221
793	2491·28	493897	855	2686·06	574146	917	2880·84	660433	979	3075·62	752758
794	2494·42	495143	856	2689·20	575490	918	2883·98	661874	980	3078·76	754296
795	2497·57	496391	857	2692·34	576835	919	2887·12	663317	981	3081·90	755837
796	2500·71	497641	858	2695·49	578182	920	2890·27	664761	982	3085·04	757378
797	2503·85	498892	859	2698·63	579530	921	2893·41	666207	983	3088·19	758922
798	2506·99	500145	860	2701·77	580880	922	2896·55	667654	984	3091·33	760466
799	2510·13	501399	861	2704·91	582232	923	2899·69	669103	985	3094·47	762013
800	2513·27	502655	862	2708·05	583585	924	2902·83	670554	986	3097·61	763561
801	2516·42	503912	863	2711·19	584940	925	2905·97	672006	987	3100·75	765111
802	2519·56	505171	864	2714·34	586297	926	2909·11	673460	988	3103·89	766662
803	2522·70	506432	865	2717·48	587655	927	2912·26	674915	989	3107·04	768215
804	2525·84	507694	866	2720·62	589014	928	2915·40	676372	990	3110·18	769769
805	2528·98	508958	867	2723·76	590375	929	2918·54	677831	991	3113·32	771325
806	2532·12	510223	868	2726·90	591738	930	2921·68	679291	992	3116·46	772882
807	2535·27	511490	869	2730·04	593102	931	2924·82	680752	993	3119·60	774441
808	2538·41	512758	870	2733·19	594468	932	2927·96	682216	994	3122·74	776002
809	2541·55	514028	871	2736·33	595835	933	2931·11	683680	995	3125·88	777564
810	2544·69	515300	872	2739·47	597204	934	2934·25	685147	996	3129·03	779128
811	2547·83	516573	873	2742·61	598575	935	2937·39	686615	997	3132·17	780693
812	2550·97	517848	874	2745·75	599947	936	2940·53	688084	998	3135·31	782260
813	2554·11	519124	875	2748·89	601320	937	2943·67	689555	999	3138·45	783828
814	2557·26	520402	876	2752·04	602696	938	2946·81	691028			

CIRCUMFERENCE AND AREA OF CIRCLES.

TABLE II.—Diameters advancing by 8ths.

Diameter.	Circ.	Area.	Diameter.	Circ.	Area.	Diameter.	Circ.	Area.	Diameter.	Circ.	Area.
1/32	·0981	·00076	3/4	21·20	35·784	1/2	45·55	165·13	1/4	69·90	388·82
1/16	·1963	·00306		21·57	37·122		45·94	167·98		70·29	393·20
1/8	·3926	·01227	7	21·99	38·484	3/4	46·33	170·87	1/2	70·68	397·60
3/16	·5890	·02761		22·38	39·871		46·73	173·78		71·07	402·03
1/4	·7854	·04908	1/4	22·77	41·282	15	47·12	176·71	3/4	71·47	406·49
5/16	·9817	·07669		23·16	42·718		47·51	179·67		71·86	410·97
3/8	1·178	·1104	1/2	23·56	44·178	1/4	47·90	182·65	23	72·25	415·47
7/16	1·374	·1503		23·95	45·663		48·30	185·66		72·64	420·00
1/2	1·570	·1963	3/4	24·34	47·173	1/2	48·69	188·69	1/4	73·04	424·55
9/16	1·767	·2485		24·74	48·707		49·08	191·74		73·43	429·13
5/8	1·963	·3067	8	25·13	50·265	3/4	49·48	194·82	1/2	73·82	433·73
11/16	2·159	·3712		25·52	51·848		49·87	197·93		74·21	438·30
3/4	2·356	·4417	1/4	25·91	53·456	16	50·26	201·06	3/4	74·61	443·01
13/16	2·552	·5184		26·31	55·088		50·65	204·21		75·	447·69
7/8	2·748	·6013	1/2	26·70	56·745	1/4	51·05	207·39	24	75·39	452·39
15/16	2·945	·6902		27·09	58·426		51·44	210·59		75·78	457·11
1	3·141	·7854	3/4	27·48	60·132	1/2	51·83	213·82	1/4	76·18	461·86
	3·534	·9940		27·88	61·862		52·22	217·07		76·57	466·63
1/4	3·927	1·227	9	28·27	63·617	3/4	52·62	220·35	1/2	76·96	471·43
	4·319	1·484		28·66	65·396		53·01	223·65		77·36	476·25
1/2	4·712	1·767	1/4	29·05	67·200	17	53·40	226·98	3/4	77·75	481·10
	5·105	2·073		29·45	69·029		53·79	230·33		78·14	485·97
3/4	5·497	2·405	1/2	29·84	70·882	1/4	54·19	233·70	25	78·54	490·87
	5·890	2·761		30·23	72·759		54·58	237·10		78·93	495·79
2	6·283	3·141	3/4	30·63	74·662	1/2	54·97	240·52	1/4	79·32	500·74
	6·675	3·546		31·02	76·588		55·37	243·97		79·71	505·71
1/4	7·068	3·976	10	31·41	78·539	3/4	55·76	247·45	1/2	80·10	510·70
	7·461	4·430		31·80	80·515		56·16	250·94		80·50	515·72
1/2	7·854	4·908	1/4	32·20	82·516	18	56·54	254·46	3/4	80·89	520·70
	8·246	5·411		32·59	84·540		56·94	258·01		81·28	525·83
3/4	8·639	5·939	1/2	32·98	86·590	1/4	57·33	261·58	26	81·68	530·93
	9·032	6·491		33·37	88·664		57·72	265·18		82·07	536·04
3	9·424	7·068	3/4	33·77	90·762	1/2	58·11	268·80	1/4	82·46	541·18
	9·817	7·669		34·16	92·885		58·51	272·44		82·85	546·35
1/4	10·21	8·295	11	34·55	95·033	3/4	58·90	276·11	1/2	83·25	551·54
	10·60	8·946		34·95	97·205		59·29	279·81		83·64	556·76
1/2	10·99	9·621	1/4	35·34	99·402	19	59·69	283·52	3/4	84·03	562·00
	11·38	10·320		35·73	101·62		60·08	287·27		84·43	567·26
3/4	11·78	11·044	1/2	36·12	103·86	1/4	60·47	291·03	27	84·82	572·55
	12·17	11·793		36·52	106·13		60·86	294·83		85·21	577·87
4	12·56	12·566	3/4	36·91	108·43	1/2	61·26	298·64	1/4	85·60	583·20
	12·95	13·364		37·30	110·75		61·65	302·48		86·	588·57
1/4	13·35	14·186	12	37·69	113·09	3/4	62·04	306·35	1/2	86·39	593·95
	13·74	15·033		38·09	115·46		62·43	310·24		86·78	599·37
1/2	14·13	15·904	1/4	38·48	117·85	20	62·83	314·16	3/4	87·17	604·80
	14·52	16·800		38·87	120·27		63·22	318·09		87·57	610·26
3/4	14·92	17·720	1/2	39·27	122·71	1/4	63·61	322·06	28	87·96	615·75
	15·31	18·665		39·66	125·18		64·01	326·05		88·35	621·26
5	15·70	19·635	3/4	40·05	127·67	1/2	64·40	330·06	1/4	88·75	626·79
	16·10	20·629		40·44	130·19		64·79	334·10		89·14	632·35
1/4	16·49	21·647	13	40·84	132·73	3/4	65·18	338·16	1/2	89·53	637·94
	16·88	22·690		41·23	135·29		65·58	342·25		89·92	643·54
1/2	17·27	23·758	1/4	41·62	137·88	21	65·97	346·36	3/4	90·32	649·18
	17·67	24·850		42·01	140·50		66·36	350·49		90·71	654·83
3/4	18·06	25·967	1/2	42·41	143·13	1/4	66·75	354·65	29	91·10	660·52
	18·45	27·108		42·80	145·80		67·15	358·84		91·49	666·22
6	18·84	28·274	3/4	43·19	148·48	1/2	67·54	363·05	1/4	91·89	671·95
	19·24	29·464		43·58	151·20		67·93	367·28		92·28	677·71
1/4	19·63	30·679	14	43·98	153·93	3/4	68·32	371·54	1/2	92·67	683·49
	20·02	31·919		44·37	156·69		68·72	375·82		93·06	689·29
1/2	20·42	33·183	1/4	44·76	159·48	22	69·11	380·13	3/4	93·46	695·12
	20·81	34·471		45·16	162·29		69·50	384·46		93·85	700·98

TABLE II. (*continued*).

Diameter.	Circ.	Area.	Diameter.	Circ.	Area.	Diameter.	Circ.	Area.	Diameter.	Circ.	Area.
30	94·24	706·86	¾	118·6	1119·2	½	142·9	1625·9	¼	167·2	2227·0
	94·64	712·76		118·9	1126·6		143·3	1634·9		167·6	2237·5
¼	95·03	718·69	38	119·3	1134·1	¾	143·7	1643·8	½	168·	2248·0
	95·42	724·64		119·7	1141·5		144·1	1652·8		168·4	2258·5
½	95·81	730·61	¼	120·1	1149·0	46	144·5	1661·9	¾	168·8	2269·0
	96·21	736·61		120·5	1156·6		144·9	1670·9		169·2	2279·6
¾	96·60	742·64	½	120·9	1164·1	¼	145·2	1680·0	54	169·6	2290·2
	96·99	748·69		121·3	1171·7		145·6	1689·1		170·	2300·8
31	97·38	754·76	¾	121·7	1179·3	½	146·	1698·2	¼	170·4	2311·4
	97·78	760·86		122·1	1186·9		146·4	1707·3		170·8	2322·1
¼	98·17	766·99	39	122·5	1191·5	¾	146·8	1716·5	½	171·2	2332·8
	98·56	773·14		122·9	1202·2		147·2	1725·7		171·6	2343·5
½	98·96	779·31	¼	123·3	1209·9	47	147·6	1734·9	¾	172·	2354·2
	99·35	785·51		123·7	1217·6		148·	1744·1		172·3	2365·0
¾	99·74	791·73	½	124·	1225·4	¼	148·4	1753·4	55	172·7	2375·8
	100·1	797·97		124·4	1233·1		148·8	1762·7		173·1	2386·6
32	100·5	804·24	¾	124·8	1240·9	½	149·2	1772·0	¼	173·5	2397·4
	100·9	810·54		125·2	1248·7		149·6	1781·3		173·9	2408·3
¼	101·3	816·86	40	125·6	1256·6	¾	150·	1790·7	½	174·3	2419·2
	101·7	823·21		126·	1264·5		150·4	1800·1		174·7	2430·1
½	102·1	829·57	¼	126·4	1272·3	48	150·7	1809·5	¾	175·1	2441·0
	102·4	835·97		126·8	1280·3		150·1	1818·9		175·5	2452·0
¾	102·8	842·39	½	127·2	1288·2	¼	151·5	1828·4	56	175·9	2463·0
	103·2	818·83		127·6	1296·2		151·9	1837·9		176·3	2474·0
33	103·6	855·30	¾	128·	1304·2	½	152·3	1847·4	¼	176·7	2485·0
	104·	861·79		128·4	1312·2		152·7	1856·9		177·1	2496·1
¼	104·4	868·30	41	128·8	1320·2	¾	153·1	1866·5	½	177·5	2507·1
	104·8	874·81		129·1	1328·3		153·5	1876·1		177·8	2518·2
½	105·2	881·41	¼	129·5	1336·4	49	153·9	1885·7	¾	178·2	2529·4
	105·6	888·00		129·9	1344·5		154·3	1895·3		178·6	2540·5
¾	106·	894·61	½	130·3	1352·6	¼	154·7	1905·0	57	179·	2551·7
	106·4	901·25		130·7	1360·8		155·1	1914·7		179·4	2562·9
34	106·8	907·92	¾	131·1	1369·0	½	155·5	1924·4	¼	179·8	2574·1
	107·2	914·61		131·5	1377·2		155·9	1934·1		180·2	2585·4
¼	107·5	921·32	42	131·9	1385·4	¾	156·2	1943·9	½	180·6	2596·7
	107·9	928·06		132·3	1393·7		156·6	1953·6		181·	2608·0
½	108·3	934·82	¼	132·7	1401·9	50	157·	1963·5	¾	181·4	2619·3
	108·7	941·60		133·1	1410·2		157·4	1973·3		181·8	2630·7
¾	109·1	948·41	½	133·5	1418·6	¼	157·8	1983·1	58	182·2	2642·0
	109·5	955·25		133·9	1426·9		158·2	1993·0		182·6	2653·4
35	109·9	962·11	¾	134·3	1435·3	½	158·6	2002·9	¼	182·9	2664·9
	110·3	968·99		134·6	1443·7		159·	2012·8		183·3	2676·3
¼	110·7	975·90	43	135·	1452·2	¾	159·4	2022·8	½	183·7	2687·8
	111·1	982·81		135·4	1460·6		159·8	2032·8		184·1	2699·3
½	111·5	989·80	¼	135·8	1469·1	51	160·2	2042·8	¾	184·5	2710·8
	111·9	996·78		136·2	1477·6		160·6	2052·8		184·9	2722·4
¾	112·3	1003·7	½	136·6	1486·1	¼	161·	2062·9	59	185·3	2733·9
	112·7	1010·8		137·	1494·7		161·3	2072·9		185·7	2745·5
36	113·	1017·8	¾	137·4	1503·3	½	161·7	2083·0	¼	186·1	2757·1
	113·4	1024·9		137·8	1511·9		162·1	2093·2		186·5	2768·8
¼	113·8	1032·0	44	138·2	1520·5	¾	162·5	2103·3	½	186·9	2780·5
	114·2	1039·1		138·6	1529·1		162·9	2113·5		187·3	2792·2
½	114·6	1046·3	¼	139·	1537·8	52	163·3	2123·7	¾	187·7	2803·9
	115·	1053·5		139·4	1546·5		163·7	2133·9		188·1	2815·6
¾	115·4	1060·7	½	139·8	1555·2	¼	164·1	2144·1	60	188·4	2827·4
	115·8	1067·9		140·1	1564·0		164·5	2154·4		188·8	2839·2
37	116·2	1075·2	¾	140·5	1572·8	½	164·9	2164·7	¼	189·2	2851·0
	116·6	1082·4		140·9	1581·6		165·3	2175·0		189·6	2862·8
¼	117·	1089·7	45	141·3	1590·4	¾	165·7	2185·4	½	190·	2874·7
	117·4	1097·1		141·7	1599·2		166·1	2195·7		190·4	2886·6
½	117·8	1104·4	¼	142·1	1608·1	53	166·5	2206·1	¾	190·8	2898·5
	118·2	1111·8		142·5	1617·0		166·8	2216·6		191·2	2910·5

TABLE II. (*continued*).

Diameter.	Circ.	Area.	Diameter.	Circ.	Area.	Diameter.	Circ.	Area.	Diameter.	Circ.	Area.
61	191·6	2922·4	¾	215·9	3712·2	½	240·3	4596·3	¼	264·6	5574·8
	192·	2934·4		216·3	3725·7		240·7	4611·3		265·	5591·3
¼	192·4	2946·4	69	216·7	3739·2	¾	241·1	4626·4	½	265·4	5607·9
	192·8	2958·5		217·1	3752·8		241·5	4641·5		265·8	5624·5
½	193·2	2970·5	¼	217·5	3766·4	77	241·9	4656·6	¾	266·2	5641·1
	193·6	2982·6		217·9	3780·0		242·2	4671·7		266·6	5657·8
¾	193·9	2994·7	½	218·3	3793·6	¼	242·6	4686·9	85	267·	5674·5
	194·3	3006·9		218·7	3807·3		243·	4702·1		267·4	5691·2
62	194·7	3019·0	¾	219·1	3821·0	½	243·4	4717·3	¼	267·8	5707·9
	195·1	3031·2		219·5	3834·7		243·8	4732·5		268·2	5724·6
¼	195·5	3043·4	70	219·9	3848·4	¾	244·2	4747·7	½	268·6	5741·4
	195·9	3055·7		220·3	3862·2		244·6	4763·0		268·9	5758·2
½	196·3	3067·9	¼	220·6	3875·9	78	245·	4778·3	¾	269·3	5775·0
	196·7	3080·2		221·	3889·8		245·4	4793·7		269·7	5791·9
¾	197·1	3092·5	½	221·4	3903·6	¼	245·8	4809·0	86	270·1	5808·8
	197·5	3104·8		221·8	3917·4		246·2	4824·4		270·5	5825·7
63	197·9	3117·2	¾	222·2	3931·3	½	246·6	4839·8	¼	270·9	5842·6
	198·3	3129·6		222·6	3945·2		247·	4855·2		271·3	5859·5
¼	198·7	3142·0	71	223·	3959·2	¾	247·4	4870·7	½	271·7	5876·5
	199·	3154·4		223·4	3973·1		247·7	4886·1		272·1	5893·5
½	199·4	3166·9	¼	223·8	3987·1	79	248·1	4901·6	¾	272·5	5910·5
	199·8	3179·4		224·2	4001·1		248·5	4917·2		272·9	5927·6
¾	200·2	3191·9	½	224·6	4015·1	¼	248·9	4932·7	87	273·3	5944·6
	200·6	3204·4		225·	4029·2		249·3	4948·3		273·7	5961·7
64	201·	3216·9	¾	225·4	4043·2	½	249·7	4963·9	¼	274·1	5978·9
	201·4	3229·5		225·8	4067·3		250·1	4979·5		274·4	5996·0
¼	201·8	3242·1	72	226·1	4071·5	¾	250·5	4995·1	½	274·8	6013·2
	202·2	3254·8		226·5	4085·6		250·9	5010·8		275·2	6030·4
½	202·6	3267·4	¼	226·9	4099·8	80	251·3	5026·5	¾	275·6	6047·6
	203·	3280·1		227·3	4114·0		251·7	5042·2		276·	6064·8
¾	203·4	3292·8	½	227·7	4128·2	¼	252·1	5058·0	88	276·4	6082·1
	203·8	3305·5		228·1	4142·5		252·5	5073·7		276·8	6099·4
65	204·2	3318·3	¾	228·5	4156·7	½	252·8	5089·5	¼	277·2	6116·7
	204·5	3331·0		228·9	4171·0		253·2	5105·4		277·6	6134·0
¼	204·9	3343·8	73	229·3	4185·3	¾	253·6	5121·2	½	278·	6151·4
	205·3	3356·7		229·7	4199·7		254·	5137·1		278·4	6168·8
½	205·7	3369·5	¼	230·1	4214·1	81	254·4	5153·0	¾	278·8	6186·2
	206·1	3382·4		230·5	4228·5		254·8	5168·9		279·2	6203·6
¾	206·5	3395·3	½	230·9	4242·9	¼	255·2	5184·8	89	279·6	6221·1
	206·9	3408·2		231·3	4257·3		255·6	5200·8		279·9	6238·6
66	207·3	3421·2	¾	231·6	4271·8	½	256·	5216·8	¼	280·3	6256·1
	207·7	3434·1		232·	4286·3		256·4	5232·8		280·7	6273·0
¼	208·1	3447·1	74	232·4	4300·8	¾	256·8	5248·8	½	281·1	6291·2
	208·5	3460·1		232·8	4315·3		257·2	5264·9		281·5	6308·8
½	208·9	3473·2	¼	233·2	4329·9	82	257·6	5281·0	¾	281·9	6326·4
	209·3	3486·3		233·6	4344·5		258·	5297·1		282·3	6344·0
¾	209·7	3499·3	½	234·	4359·1	¼	258·3	5313·2	90	282·7	6361·7
	210·	3512·5		234·4	4373·8		258·7	5329·4		283·1	6379·4
67	210·4	3525·6	¾	234·8	4388·4	½	259·1	5345·6	¼	283·5	6397·1
	210·8	3538·8		235·2	4403·1		259·5	5361·8		283·9	6414·8
¼	211·2	3552·0	75	235·6	4417·8	¾	259·9	5378·0	½	284·3	6432·6
	211·6	3565·2		236·	4432·6		260·3	5394·3		284·7	6450·4
½	212·	3578·4	¼	236·4	4447·3	83	260·7	5410·6	¾	285·1	6468·2
	212·4	3591·7		236·7	4462·1		261·1	5426·9		285·4	6486·0
¾	212·8	3605·0	½	237·1	4476·9	¼	261·5	5443·2	91	285·8	6503·8
	213·2	3618·3		237·5	4491·8		261·9	5459·6		286·2	6521·7
68	213·6	3631·6	¾	237·9	4506·6	½	262·3	5476·0	¼	286·6	6539·6
	214·	3645·0		238·3	4521·5		262·7	5492·4		287·	6557·6
¼	214·4	3658·4	76	238·7	4536·4	¾	263·1	5508·8	½	287·4	6575·5
	214·8	3671·8		239·1	4551·4		263·5	5525·3		287·8	6593·5
½	215·1	3685·2	¼	239·5	4566·3	84	263·8	5541·7	¾	288·2	6611·5
	215·5	3698·7		239·9	4581·3		264·2	5558·2		288·6	6629·5

TABLE II. (*continued*).

Diameter.	Circ.	Area.	Diameter.	Circ.	Area.	Diameter.	Circ.	Area.	Diameter.	Circ.	Area.
92	289·	6647·6	¼	296·	6976·7	½	303·1	7313·8	¾	310·2	7658·8
	289·4	6665·7		296·4	6995·2		303·5	7332·8		310·6	7678·2
¼	289·8	6683·8	½	296·8	7013·8	¾	303·9	7351·7	99	311·	7697·7
	290·2	6701·9		297·2	7032·3		304·3	7370·7		311·4	7717·1
½	290·5	6720·0	¾	297·6	7050·9	97	304·7	7389·8	¼	311·8	7736·6
	290·9	6738·2		298·	7069·5		305·1	7408·8		312·1	7756·1
¾	291·3	6756·4	95	298·4	7088·2	¼	305·5	7427·9	½	312·5	7775·6
	291·7	6776·4		298·8	7106·9		305·9	7447·0		312·9	7795·2
93	292·1	6792·9	¼	299·2	7125·5	½	306·3	7466·2	¾	313·3	7814·7
	292·5	6811·1		299·6	7144·3		306·6	7485·3		313·7	7834·3
¼	292·9	6829·4	½	300·	7163·0	¾	307·	7504·5	100	314·1	7853·9
	293·3	6847·8		300·4	7181·8		307·4	7523·7		314·5	7853·6
½	293·7	6866·1	¾	300·8	7200·5	98	307·8	7542·9	¼	314·9	7893·3
	294·1	6884·5		301·2	7219·4		308·2	7562·2		315·3	7913·1
¾	294·5	6902·9	96	301·5	7238·2	¼	308·6	7581·5	½	315·7	7932·7
	294·9	6921·3		301·9	7257·1		309·	7600·8		316·	7942·4
94	295·3	6939·7	¼	302·3	7275·9	½	309·4	7620·1	¾	316·4	7972·2
	295·7	6958·2		302·7	7294·9		309·8	7639·4		316·8	7991·9

AREA OF CIRCLES AND SQUARES.

TABLE OF THE AREAS OF CIRCLES AND OF THE SIDES OF SQUARES OF THE SAME AREA.

Diam. of Circle in Ins.	Area of Circle in Sq. Ins.	Sides of Square of same Area in Sq. Ins.	Diam. of Circle in Ins.	Area of Circle in Sq. Ins.	Sides of Square of same Area in Sq. Ins.	Diam. of Circle in Ins.	Area of Circle in Sq. Ins.	Sides of Square of same Area in Sq. Ins.
1	·785	·89	17	226·98	15·07	33	855·30	29·25
1½	1·767	1·33	17½	240·53	15·51	33½	881·41	29·69
2	3·142	1·77	18	254·47	15·95	34	907·92	30·13
2½	4·909	2·22	18½	268·80	16·40	34½	934·82	30·57
3	7·069	2·66	19	283·53	16·84	35	962·11	31·02
3½	9·621	3·10	19½	298·65	17·28	35½	989·80	31·46
4	12·566	3·54	20	314·16	17·72	36	1017·88	31·90
4½	15·904	3·99	20½	330·06	18·17	36½	1046·35	32·35
5	19·635	4·43	21	343·36	18·61	37	1075·21	32·79
5½	23·758	4·87	21½	363·05	19·05	37½	1104·47	33·23
6	28·274	5·32	22	380·13	19·50	38	1134·12	33·68
6½	33·183	5·76	22½	397·61	19·94	38½	1164·16	34·12
7	38·485	6·20	23	415·48	20·38	39	1194·59	34·56
7½	44·179	6·65	23½	433·74	20·83	39½	1225·42	35·01
8	50·266	7·09	24	452·39	21·27	40	1256·64	35·45
8½	56·745	7·53	24½	471·44	21·71	40½	1288·25	35·89
9	63·617	7·98	25	490·88	22·16	41	1320·26	36·34
9½	70·882	8·42	25½	510·71	22·60	41½	1352·66	36·78
10	78·540	8·86	26	530·93	23·04	42	1385·45	37·22
10½	86·590	9·30	26½	551·55	23·49	42½	1418·63	37·66
11	95·03	9·75	27	572·56	23·93	43	1452·20	38·11
11½	103·87	10·19	27½	593·96	24·37	43½	1486·17	38·55
12	113·10	10·63	28	615·75	24·81	44	1520·53	38·99
12½	122·72	11·08	28½	637·94	25·26	44½	1555·29	39·44
13	132·73	11·52	29	660·52	25·70	45	1590·43	39·88
13½	143·14	11·96	29½	683·49	26·14	45½	1625·97	40·32
14	153·94	12·41	30	706·86	26·59	46	1661·91	40·77
14½	165·13	12·85	30½	730·62	27·03	46½	1698·23	41·21
15	176·72	13·29	31	754·77	27·47	47	1734·95	41·65
15½	188·69	13·74	31½	779·31	27·92	47½	1772·06	42·10
16	201·06	14·18	32	804·25	28·36	48	1809·56	42·58
16½	213·83	14·62	32½	829·58	28·80	48½	1847·46	42·98

Table of the Areas of Circles and of the Sides of Squares (*continued*)

Diam. of Circle in Ins.	Area of Circle in Sq. Ins.	Sides of Square of same Area in Sq. Ins.	Diam. of Circle in Ins.	Area of Circle in Sq. Ins.	Sides of Square of same Area in Sq. Ins.	Diam. of Circle in Ins.	Area of Circle in Sq. Ins.	Sides of Square of same Area in Sq. Ins.
49	1885·75	43·43	53	2206·19	46·97	57	2551 76	50·51
49½	1924·43	43·87	53½	2248·01	47·41	57½	2596·73	50·96
50	1963·50	44·31	54	2290·23	47·86	58	2642·09	51·40
50½	2002·97	44·75	54½	2332·83	48·30	58½	2687·84	51·84
51	2042·83	45·20	55	2375·83	48·74	59	2733·98	52·29
51½	2083·08	45·64	55½	2419·23	49·19	59½	2780·51	52·73
52	2123·72	46·08	56	2463·01	49·63	50	2827·74	53·17
52½	2164·76	46·53	46½	2507·19	50·07	60½	2874·76	53·62

TRIGONOMETRY.

Radians to Degrees.

$(a^\circ = 57^\circ \cdot 295798 \times r)$

Radians (r)	Degrees (a°)	Radians (r)	Degrees (a°)	Radians (r)	Degrees (a°)	Radians (r)	Degrees (a°)	Radians (r)	Degrees (a°)
·01	0·57	·33	18·91	·65	37·24	·97	55·58	1·29	73·91
·02	1·15	·34	19·48	·66	37·82	·98	56·15	1·30	74·48
·03	1·72	·35	20·05	·67	38·39	·99	56·72	1·31	75 06
·04	2·29	·36	20·63	·68	38·96	1·00	57·30	1·32	75·63
·05	2·86	·37	21·20	·69	39·53	1·01	57·87	1·33	76·20
·06	3·44	·38	21·77	·70	40·11	1·02	58·44	1·34	76·78
·07	4·01	·39	22·34	·71	40·68	1·03	59·01	1·35	77·35
·08	4·58	·40	22·92	·72	41·25	1·04	59·59	1·36	77·92
·09	5·16	·41	23·49	·73	41·83	1·05	60·16	1·37	78·50
·10	5·73	·42	24·06	·74	42·40	1·06	60·73	1·38	79·07
·11	6·30	·43	24·64	·75	42·97	1·07	61·31	1·39	79·64
·12	6·88	·44	25·21	·76	43·54	1·08	61·88	1·40	80·21
·13	7·45	·45	25·78	·77	44·12	1·09	62·45	1·41	80·79
·14	8·02	·46	26·36	·78	44·69	1·10	63·03	1·42	81·36
·15	8·59	·47	26·93	·79	45·26	1·11	63·60	1·43	81·93
·16	9·17	·48	27·50	·80	45·84	1·12	64·17	1·44	82·51
·17	9·74	·49	28·07	·81	46·41	1·13	64·74	1·45	83·08
·18	10·31	·50	28·65	·82	46·98	1·14	65·32	1·46	83·65
·19	10·89	·51	29·22	·83	47·56	1·15	65·89	1·47	84·23
·20	11·46	·52	29·79	·84	48·13	1·16	66·46	1·48	84·80
·21	12·03	·53	30·37	·85	48·70	1·17	67·04	1·49	85·37
·22	12·61	·54	30·94	·86	49·27	1·18	67·61	1·50	85·94
·23	13·18	·55	31·51	·87	49·85	1·19	68·18	1·51	86·52
·24	13·75	·56	32·09	·88	50·42	1·20	68·75	1·52	87·09
·25	14·32	·57	32·66	·89	50·99	1·21	69·33	1·53	87·66
·26	14·90	·58	33·23	·90	51·57	1·22	69·90	1·54	88·24
·27	15·47	·59	33·80	·91	52·14	1·23	70·47	1·55	88·81
·28	16·04	·60	34·38	·92	52·71	1·24	71·05	1·56	89·38
·29	16·62	·61	34·95	·93	53·29	1·25	71·62	1·57	89·95
·30	17·19	·62	35·52	·94	53·86	1·26	72·19	$\pi/2$	90·00
·31	17·76	·63	36·10	·95	54·43	1·27	72·77		
·32	18·33	·64	36·67	·96	55·00	1·28	73·34		

Degrees to Radians.

($r = \cdot 0174533 \times a^{\circ}$)

Degrees (a°)	Radian (r)	Degrees (a°)	Radians (r)	Degrees (a°)	Radians (r)	Degrees (a°)	Radians (r)	Degrees (a°)	Radians (r)
1	·0175	37	·6458	73	1·274	109	1·902	145	2·530
2	·0349	38	·6632	74	1·291	110	1·920	146	2·548
3	·0524	39	·6807	75	1·309	111	1·937	147	2·565
4	·0698	40	·6981	76	1·326	112	1·954	148	2·583
5	·0873	41	·7156	77	1·344	113	1·972	149	2·600
6	·1047	42	·7330	78	1·361	114	1·989	150	2·618
7	·1222	43	·7505	79	1·379	115	2·007	151	2·635
8	·1396	44	·7679	80	1·396	116	2·024	152	2·652
9	·1571	45	·7854	81	1·413	117	2·042	153	2·670
10	·1745	46	·8029	82	1·431	118	2·059	154	2·687
11	·1920	47	·8203	83	1·448	119	2·077	155	2·705
12	·2094	48	·8378	84	1·466	120	2·094	156	2·722
13	·2269	49	·8552	85	1·483	121	2·112	157	2·740
14	·2443	50	·8727	86	1·501	122	2·129	158	2·757
15	·2618	51	·8901	87	1·518	123	2·146	159	2·775
16	·2793	52	·9076	88	1·536	124	2·164	160	2·792
17	·2967	53	·9250	89	1·553	125	2·181	161	2·810
18	·3142	54	·9425	90	1·571	126	2·199	162	2·827
19	·3316	55	·9599	91	1·588	127	2·216	163	2·844
20	·3491	56	·9774	92	1·605	128	2·234	164	2·862
21	·3665	57	·9948	93	1·623	129	2·251	165	2·879
22	·3840	58	1·012	94	1·640	130	2·269	166	2·897
23	·4014	59	1·030	95	1·658	131	2·286	167	2·914
24	·4189	60	1·047	76	1·675	132	2·304	168	2·932
25	·4363	61	1·064	97	1·693	133	2·321	169	2·945
26	·4538	62	1·082	98	1·710	134	2·338	170	2·967
27	·4712	63	1·099	99	1·728	135	2·356	171	2·984
28	·4887	64	1·117	100	1·745	136	2·373	172	3·001
29	·5061	65	1·134	101	1·762	137	2·391	173	3·019
30	·5236	66	1·152	102	1·780	138	2·408	174	3·036
31	·5410	67	1·169	103	1·797	139	2·426	175	3·054
32	·5585	68	1·187	104	1·815	140	2·443	176	3·071
33	·5760	69	1·204	105	1·832	141	2·460	177	3·089
34	·5934	70	1·222	106	1·850	142	2·478	178	3·106
35	·6109	71	1·239	107	1·867	143	2·495	179	3·124
36	·6283	72	1·256	108	1·884	144	2·513	180	3·141

NATURAL SINES, TANGENTS, &c.

	Sine.	Tang.	Cotang.	Cosine.	
0 Deg.					
0′	·0000000	·000000	Infinite	1·000000	60
1	·0002909	·000291	3437·746	1·000000	59
2	·0005818	·000582	1718·873	·9999998	58
3	·0008727	·000872	1145·915	·9999996	57
4	·0011636	·001163	859·4363	·9999993	56
5	·0014544	·001454	687·5488	·9999989	55
6	·0017453	·001745	572·9572	·9999985	54
7	·0020362	·002036	491·1060	·9999979	53
8	·0023271	·002327	429·7175	·9999973	52
9	·0026180	·002618	381·9709	·9999966	51
10	·0029089	·002908	343·7737	·9999958	50
11	·0031998	·003199	312·5213	·9999949	49
12	·0034907	·003490	286·4777	·9999939	48
13	·0037815	·003781	264·4408	·9999928	47
14	·0040724	·004072	245·5519	·9999917	46
15	·0043633	·004363	229·1816	·9999905	45
16	·0046542	·004654	214·8576	·9999892	44
17	·0049451	·004945	202·2187	·9999878	43
18	·0052360	·005236	190·9841	·9999863	42
19	·0055268	·005526	180·9322	·9999847	41
20	·0058177	·005817	171·8854	·9999831	40
21	·0061086	·006108	163·7001	·9999813	39
22	·0063995	·006399	156·2590	·9999795	38
23	·0066904	·006690	149·4650	·9999776	37
24	·0069813	·006981	143·2371	·9999756	36
25	·0072721	·007272	137·5075	·9999736	35
26	·0075630	·007563	132·2185	·9999714	34
27	·0078539	·007854	127·3213	·9999692	33
28	·0081448	·008145	122·7739	·9999668	32
29	·0084357	·008436	118·5401	·9999644	31
30	·0087265	·008726	114·5886	·9999619	30
31	·0090174	·009017	110·8920	·9999593	29
32	·0093083	·009308	107·4264	·9999567	28
33	·0095992	·009599	104·1709	·9999539	27
34	·0098900	·009890	101·1069	·9999511	26
35	·0101809	·010181	98·21794	·9999482	25
36	·0104718	·010472	95·48947	·9999452	24
37	·0107627	·010763	92·90848	·9999421	23
38	·0110535	·011054	90·46333	·9999389	22
39	·0113444	·011345	88·14357	·9999357	21
40	·0116353	·011636	85·93979	·9999323	20
41	·0119261	·011927	83·84350	·9999289	19
42	·0122170	·012217	81·84704	·9999254	18
43	·0125079	·012508	79·94343	·9999218	17
44	·0127987	·012799	78·12634	·9999181	16
45	·0130896	·013090	76·39000	·9999143	15
46	·0133805	·013381	74·72916	·9999105	14
47	·0136713	·013672	73·13899	·9999065	13
48	·0139622	·013963	71·61507	·9999025	12
49	·0142530	·014254	70·15334	·9998984	11
50	·0145439	·014545	68·75008	·9998942	10
51	·0148348	·014836	67·40185	·9998900	9
52	·0151256	·015127	66·10547	·9998856	8
53	·0154165	·015418	64·85800	·9998812	7
54	·0157073	·015709	63·65674	·9998766	6
55	·0159982	·016000	62·49915	·9998720	5
56	·0162890	·016291	61·38290	·9998673	4
57	·0165799	·016582	60·30582	·9998625	3
58	·0168707	·016873	59·26587	·9998577	2
59	·0171616	·017164	58·26117	·9998527	1
60	·0174524	·017455	57·28996	·9998477	0′
					89 Deg.
	Cosine.	Cotan.	Tang.	Sine.	

	Sine.	Tang.	Cotang.	Cosine.	
1 Deg.					
0′	·0174524	·017455	57·28996	·9998477	60
5	·0189066	·018910	52·88211	·9998213	55
10	·0203608	·020365	49·10388	·9997927	50
15	·0218149	·021820	45·82935	·9997620	45
20	·0232690	·023275	42·96407	·9997292	40
25	·0247230	·024730	40·43583	·9996943	35
30	·0261769	·026185	38·18845	·9996573	30
35	·0276309	·027641	36·17759	·9996182	25
40	·0290847	·029097	34·36777	·9995770	20
45	·0305385	·030552	32·73026	·9995336	15
50	·0319922	·032008	31·24157	·9994881	10
55	·0334459	·033464	29·88229	·9994405	5
60	·0348995	·034920	28·63625	·9993908	0′
					88 Deg.
2 Deg.					
0′	·0348955	·034920	28·63625	·9993908	60
5	·0363530	·036377	27·48985	·9993390	55
10	·0378065	·037833	26·43160	·9992851	50
15	·0392598	·039290	25·45170	·9992290	45
20	·0407131	·040746	24·54175	·9991709	40
25	·0421663	·042203	23·69453	·9991106	35
30	·0436194	·043660	22·90376	·9990482	30
35	·0450724	·045118	22·16398	·9989837	25
40	·0465253	·046575	21·47040	·9989171	20
45	·0479781	·048033	20·81882	·9988484	15
50	·0494308	·049491	20·20555	·9987775	10
55	·0508835	·050949	19·62729	·9987046	5
60	·0523360	·052407	19·08113	·9986295	0′
					87 Deg.
3 Deg.					
0′	·0523360	·052407	19·08113	·9986295	60
5	·0537883	·053866	18·56447	·9985524	55
10	·0552406	·055325	18·07497	·9984731	50
15	·0566928	·056784	17·61055	·9983917	45
20	·0581448	·058243	17·16933	·9983082	40
25	·0595967	·059702	16·74961	·9982225	35
30	·0610485	·061162	16·34985	·9981348	30
35	·0625002	·062622	15·96866	·9980450	25
40	·0639517	·064082	15·60478	·9979530	20
45	·0654031	·065543	15·25705	·9978589	15
50	·0668544	·067004	14·92441	·9977627	10
55	·0683055	·068465	14·60591	·9976645	5
60	·0697565	·069926	14·30066	·9975641	0′
					86 Deg.
4 Deg.					
0′	·0697565	·069926	14·30066	·9975641	60
5	·0712073	·071388	14·00785	·9974615	55
10	·0726580	·072850	13·72673	·9973569	50
15	·0741085	·074312	13·45662	·9972502	45
20	·0755589	·075775	13·19688	·9971413	40
25	·0770091	·077238	12·94692	·9970304	35
30	·0784591	·078701	12·70620	·9969173	30
35	·0799090	·080165	12·47422	·9968022	25
40	·0813587	·081629	12·25050	·9966849	20
45	·0828082	·083093	12·03462	·9965655	15
50	·0842576	·084558	11·82616	·9964440	10
55	·0857067	·086023	11·62476	·9963204	5
60	·0871557	·087488	11·43005	·9961947	0′
					85 Deg.
	Cosine.	Cotan.	Tang.	Sine.	

NATURAL SINES, TANGENTS, &c. (*continued*).

	Sine.	Tang.	Cotang.	Cosine.	
5 Deg.					
0′	·0871557	·087488	11·43005	·9961947	60
5	·0886046	·088954	11·24171	·9960669	55
10	·0900532	·090420	11·05943	·9959370	50
15	·0915016	·091887	10·88292	·9958049	45
20	·0929499	·093354	10·71191	·9956708	40
25	·0943979	·094821	10·54615	·9955345	35
30	·0958458	·096289	10·38539	·9953962	30
35	·0972934	·097757	10·22942	·9952557	25
40	·0987408	·099225	10·07803	·9951132	20
45	·1001881	·100694	9·931008	·9949685	15
50	·1016351	·102164	9·788173	·9948217	10
55	·1030819	·103634	9·649347	·9946729	5
60	·1045285	·105104	9·514364	·9945219	0′
					84 Deg.
6 Deg.					
0′	·1045285	[illegible]5104	9·514364	·9945219	60
5	·1059748	·106575	9·383066	·9943688	55
10	·1074210	·108046	9·255303	·9942136	50
15	·1088669	·109517	9·130934	·9940563	45
20	·1103126	·110989	9·009826	·9938969	40
25	·1117580	·112462	8·891850	·9937355	35
30	·1132032	·113935	8·776887	·9935719	30
35	·1146482	·115409	8·664822	·9934062	25
40	·1160929	·116883	8·555546	·9932384	20
45	·1175374	·118357	8·448957	·9930685	15
50	·1189816	·119832	8·344955	·9928965	10
55	·1204256	·121308	8·243448	·9927224	5
60	·1218693	·122784	8·144346	·9925462	0′
					83 Deg.
7 Deg.					
0′	·1218693	·122784	8·144346	·9925462	60
5	·1233128	·124261	8·017564	·9923679	55
10	·1247560	·125738	7·953022	·9921874	50
15	·1261990	·127216	7·860642	·9920049	45
20	·1276416	·128694	7·770350	·9918204	40
25	·1290841	·130173	7·682076	·9916337	35
30	·1305262	·131652	7·595754	·9914449	30
35	·1319681	·133132	7·511317	·9912540	25
40	·1334096	·134612	7·428706	·9910610	20
45	·1348509	·136094	7·347861	·9908659	15
50	·1362919	·137575	7·268725	·9906687	10
55	·1377327	·139058	7·191245	·9904694	5
60	·1391731	·140540	7·115369	·9902681	0′
					82 Deg.
8 Deg.					
0′	·1391731	·140540	7·115369	·9902681	60
5	·1406132	·142024	7·041048	·9900646	55
10	·1420531	·143508	6·968233	·9898590	50
15	·1434926	·144993	6·896879	·9896514	45
20	·1449319	·146478	6·826943	·9894416	40
25	·1463708	·147964	6·758382	·9892298	35
30	·1478094	·149451	6·691156	·9890159	30
35	·1492477	·150938	6·625225	·9887998	25
40	·1506857	·152426	6·560553	·9885817	20
45	·1521234	·153914	6·497104	·9883615	15
50	·1535607	·155404	6·[illegible]34842	·9881392	10
55	·1549978	·156893	6·373735	·9879148	5
60	·1564345	·158384	6·313751	·9876883	0′
					81 Deg.
	Cosine.	Cotan.	Tang.	Sine.	

	Sine.	Tang.	Cotang.	Cosine.	
9 Deg.					
0′	·1564345	·158384	6·313751	·9876883	60
5	·1578708	·159875	6·254858	·9874598	55
10	·1593069	·161367	6·197027	·9872291	50
15	·1607426	·162860	6·140230	·9869964	45
20	·1621779	·164353	6·084438	·9867615	40
25	·1636129	·165847	6·029624	·9865246	35
30	·1650476	·167342	5·975764	·9862856	30
35	·1664819	·168838	5·922832	·9860445	25
40	·1679159	·170334	5·870804	·9858013	20
45	·1693495	·171831	5·819657	·9855561	15
50	·1707828	·173329	5·769368	·9853087	10
55	·1722156	·174827	5·719917	·9850593	5
60	·1736482	·176327	5·671281	·9848078	0′
					80 Deg.
10 Deg.					
0′	·1736482	·176327	5·671281	·9848078	60
5	·1750803	·177827	5·623442	·9845542	55
10	·1765121	·179327	5·576378	·9842985	50
15	·1779435	·180829	5·530072	·9840407	45
20	·1793746	·182331	5·484505	·9837808	40
25	·1808052	·183835	5·439659	·9835189	35
30	·1822355	·185339	5·395517	·9832549	30
35	·1836654	·186843	5·352062	·9829888	25
40	·1850949	·188349	5·309279	·9827206	20
45	·1865240	·189855	5·267151	·9824504	15
50	·1879528	·191363	5·225664	·9821781	10
55	·1893811	·192871	5·184803	·9819037	5
60	·1908090	·194380	5·144554	·9816272	0′
					79 Deg.
11 Deg.					
0′	·1908090	·194380	5·144554	·9816272	60
5	·1922365	·195890	5·104902	·9813486	55
10	·1936636	·197400	5·065835	·9810680	50
15	·1950903	·198912	5·027339	·9807853	45
20	·1965166	·200424	4·989402	·9805005	40
25	·1979425	·201938	4·952012	·9802136	35
30	·1993679	·203452	4·915157	·9799247	30
35	·2007930	·204967	4·878824	·9796337	25
40	·2022176	·206483	4·843004	·9793406	20
45	·2036418	·208000	4·807685	·9790455	15
50	·2050655	·209518	4·772856	·9787483	10
55	·2064888	·211036	4·738508	·9784490	5
60	·2079117	·212556	4·704630	·9781476	0′
					78 Deg.
12 Deg.					
0′	·2079117	·212556	4·704630	·9781476	60
5	·2093341	·214077	4·671212	·9778441	55
10	·2107561	·215598	4·638245	·9775386	50
15	·2121777	·217121	4·605720	·9772311	45
20	·2135988	·218644	4·573628	·9769215	40
25	·2150194	·220169	4·541960	·9766098	35
30	·2164396	·221694	4·510708	·9762960	30
35	·2178593	·223221	4·479863	·9759802	25
40	·2192786	·224748	4·449418	·9756623	20
45	·2206974	·226276	4·419364	·9753423	15
50	·2221158	·227806	4·389694	·9750203	10
55	·2235337	·229336	4·360400	·9746962	5
60	·2249511	·230868	4·331475	·9743701	0′
					77 Deg.
	Cosine.	Cotan.	Tang.	Sine.	

NATURAL SINES, TANGENTS, &c. (*continued*).

	Sine.	Tang.	Cotang.	Cosine.	
13 Deg.					
0′	·2249511	·230868	4·331475	·9743701	60
5	·2263680	·232400	4·302913	·9740419	55
10	·2277844	·233934	4·274706	·9737116	50
15	·2292004	·235468	4·246848	·9733792	45
20	·2306159	·237004	4·219331	·9730449	40
25	·2320309	·238541	4·192151	·9727084	35
30	·2334454	·240078	4·165299	·9723699	30
35	·2348594	·241617	4·138771	·9720294	25
40	·2362729	·243157	4·112561	·9716867	20
45	·2376859	·244698	4·086662	·9713421	15
50	·2390984	·246240	4·061070	·9709953	10
55	·2405104	·247783	4·035777	·9706466	5
60	·2419219	·249328	4·010780	·9702957	0′
					76 Deg.
14 Deg.					
0′	·2419219	·249328	4·010780	·9702957	60
5	·2433329	·250873	3·986073	·9699428	55
10	·2447433	·252420	3·961651	·9695879	50
15	·2461533	·253967	3·937509	·9692309	45
20	·2475627	·255516	3·913642	·9688719	40
25	·2489716	·257066	3·890044	·9685108	35
30	·2503800	·258617	3·866713	·9681476	30
35	·2517879	·260169	3·843642	·9677825	25
40	·2531952	·261723	3·820828	·9674152	20
45	·2546019	·263278	3·798266	·9670459	15
50	·2560082	·264833	3·775951	·9666746	10
55	·2574139	·266390	3·753881	·9663012	5
60	·2588190	·267949	3·732050	·9659258	0′
					75 Deg.
15 Deg.					
0′	·2588190	·267949	3·732050	·9659258	60
5	·2602237	·269508	3·710455	·9655484	55
10	·2616277	·271069	3·689092	·9651689	50
15	·2630312	·272631	3·667957	·9647873	45
20	·2644342	·274194	3·647046	·9644037	40
25	·2658366	·275758	3·626356	·9640181	35
30	·2672384	·277324	3·605883	·9636305	30
35	·2686396	·278891	3·585624	·9632408	25
40	·2700403	·280459	3·565574	·9628490	20
45	·2714404	·282029	3·545732	·9624552	15
50	·2728400	·283599	3·526093	·9620594	10
55	·2742390	·285172	3·506655	·9616616	5
60	·2756374	·286745	3·487414	·9612617	0′
					74 Deg.
16 Deg.					
0′	·2756374	·286745	3·487414	·9612617	60
5	·2770352	·288320	3·468367	·9608598	55
10	·2784324	·289896	3·449512	·9604558	50
15	·2798290	·291473	3·430844	·9600499	45
20	·2812251	·293052	3·412362	·9596418	40
25	·2826205	·294632	3·394063	·9592318	35
30	·2840153	·296213	3·375943	·9588197	30
35	·2854096	·297796	3·358000	·9584056	25
40	·2868032	·299380	3·340232	·9579895	20
45	·2881963	·300965	3·322636	·9575714	15
50	·2895887	·302552	3·305209	·9571512	10
55	·2909805	·304141	3·287948	·9567290	5
60	·2923717	·305730	3·270852	·9563048	0′
					73 Deg.
	Cosine.	Cotan.	Tang.	Sine.	

	Sine.	Tang.	Cotang.	Cosine.	
17 Deg.					
0′	·2923717	·305730	3·270852	·9563048	60
5	·2937623	·307321	3·253918	·9558785	55
10	·2951522	·308914	3·237143	·9554502	50
15	·2965416	·310508	3·220526	·9550199	45
20	·2979303	·312103	3·204063	·9545876	40
25	·2993184	·313700	3·187754	·9541533	35
30	·3007058	·315298	3·171594	·9537170	30
35	·3020926	·316898	3·155584	9532786	25
40	·3034788	·318499	3·139719	·9528382	20
45	·3048643	·320102	3·123999	·9523958	15
50	·3062492	·321706	3·108421	·9519514	10
55	·3076334	·323312	3·092983	·9515050	5
60	·3090170	·324919	3·077683	·9510565	0′
					72 Deg.
18 Deg.					
0′	·3090170	·324919	3·077683	·9510565	60
5	·3103999	·326528	3·062520	·9506061	55
10	·3117822	·328138	3·047491	·9501536	50
15	·3131638	·329750	3·032595	·9496991	45
20	·3145448	·331363	3·017830	·9492426	40
25	·3159250	·332978	3·003193	·9487842	35
30	·3173047	·334595	2·988685	·9483237	30
35	·3186836	·336213	2·974301	·9478612	25
40	·3200619	·337833	2·960042	·9473966	20
45	·3214395	·339454	2·945905	·9469301	15
50	·3228164	·341077	2·931888	·9464616	10
55	·3241926	·342701	2·917990	·9459911	5
60	·3255682	·344327	2·904210	·9455186	0′
					71 Deg.
19 Deg.					
0′	·3255682	·344327	2·904210	·9455186	60
5	·3269430	·345955	2·890546	·9450441	55
10	·3283172	·347584	2·876997	·9445675	50
15	·3296906	·349215	2·863560	·9440890	45
20	·3310634	·350848	2·850234	·9436085	40
25	·3324355	·352482	2·837019	·9431260	35
30	·3338069	·354118	2·823912	·9426415	30
35	·3351775	·355756	2·810913	·9421550	25
40	·3365475	·357395	2·798019	·9416665	20
45	·3379167	·359036	2·785230	·9411760	15
50	·3392852	·360679	2·772544	·9406835	10
55	·3406531	·362324	2·759960	·9401891	5
60	·3420201	·363970	2·747477	·9396926	0′
					70 Deg.
20 Deg.					
0′	·3420201	·363970	2·747477	·9396926	60
5	·3433865	·365618	2·735093	·9391942	55
10	·3447521	·367268	2·722807	·9386938	50
15	·3461171	·368919	2·710618	·9381913	45
20	·3474812	·370572	2·698525	·9376869	40
25	·3488447	·372227	2·686526	·9371806	35
30	·3502074	·373884	2·674621	·9366722	30
35	·3515693	·375543	2·662808	·9361618	25
40	·3529306	·377203	2·651086	·9356495	20
45	·3542910	·378866	2·639454	·9351352	15
50	·3556508	·380530	2·627912	·9346189	10
55	·3570097	·382196	2·616457	·9341007	5
60	·3583679	·383864	2·605089	·9335804	0′
					69 Deg.
	Cosine.	Cotan.	Tang.	Sine.	

NATURAL SINES, TANGENTS, &c. (*continued*).

	Sine.	Tang.	Cotang.	Cosine.	
21 Deg.					
0′	·3583679	·383864	2·605089	·9335804	60
5	·3597254	·385533	2·593806	·9330582	55
10	·3610821	·387205	2·582609	·9325340	50
15	·3624380	·388878	2·571495	·9320079	45
20	·3637932	·390554	2·560464	·9314797	40
25	·3651476	·392231	2·549516	·9309496	35
30	·3665012	·393910	2·538647	·9304176	30
35	·3678541	·395591	2·527859	·9298835	25
40	·3692061	·397274	2·517150	·9293475	20
45	·3705574	·398959	2·506519	·9288096	15
50	·3719079	·400646	2·495966	·9282696	10
55	·3732577	·402335	2·485488	·9277277	5
60	·3746066	·404026	2·475086	·9271839	0′
					68 Deg.
22 Deg.					
0′	·3746066	·404026	2·475086	·9271839	60
5	·3759547	·405719	2·464759	·9266380	55
10	·3773021	·407413	2·454506	·9260902	50
15	·3786486	·409110	2·444325	·9255405	45
20	·3799944	·410809	2·434217	·9249888	40
25	·3813393	·412510	2·424180	·9244351	35
30	·3826834	·414213	2·414213	·9238795	30
35	·3840268	·415918	2·404316	·9233220	25
40	·3853693	·417625	2·394488	·9227624	20
45	·3867110	·419334	2·384729	·9222010	15
50	·3880518	·421046	2·375037	·9216375	10
55	·3893919	·422759	2·365411	·9210722	5
60	·3907311	·424474	2·355852	·9205049	0′
					67 Deg.
23 Deg.					
0′	·3907311	·424474	2·355852	·9205049	60
5	·3920695	·426192	2·346358	·9199356	55
10	·3934071	·427912	2·336928	·9193644	50
15	·3947439	·429633	2·327563	·9187912	45
20	·3960798	·431357	2·318260	·9182161	40
25	·3974148	·433084	2·309020	·9176391	35
30	·3987491	·434812	2·299842	·9170601	30
35	·4000825	·436542	2·290725	·9164791	25
40	·4014150	·438275	2·281669	·9158963	20
45	·4027467	·440010	2·272672	·9153115	15
50	·4040775	·441747	2·263735	·9147247	10
55	·4054075	·443487	2·254857	·9141361	5
60	·4067366	·445228	2·246036	·9135455	0′
					66 Deg.
24 Deg.					
0′	·4067366	·445228	2·246036	·9135455	60
5	·4080649	·446972	2·237273	·9129529	55
10	·4093923	·448718	2·228567	·9123584	50
15	·4107189	·450467	2·219917	·9117620	45
20	·4120445	·452217	2·211323	·9111637	40
25	·4133693	·453970	2·202784	·9105635	35
30	·4146932	·455726	2·194299	·9099613	30
35	·4160163	·457483	2·185869	·9093572	25
40	·4173385	·459243	2·177492	·9087511	20
45	·4186597	·461006	2·169167	·9081432	15
50	·4199801	·462771	2·160895	·9075333	10
55	·4212996	·464538	2·152675	·9069215	5
60	·4226183	·466307	2·144506	·9063078	0′
					65 Deg.
	Cosine.	Cotan.	Tang.	Sine.	

	Sine.	Tang.	Cotang.	Cosine.	
25 Deg.					
0′	·4226183	·466307	2·144506	·9063078	60
5	·4239360	·468079	2·136389	·9056922	55
10	·4252528	·469853	2·128321	·9050746	50
15	·4265687	·471630	2·120303	·9044551	45
20	·4278838	·473409	2·112334	·9038338	40
25	·4291979	·475191	2·104415	·9032105	35
30	·4305111	·476975	2·096543	·9025853	30
35	·4318234	·478762	2·088720	·9019582	25
40	·4331348	·480551	2·080943	·9013292	20
45	·4344453	·482342	2·073214	·9006982	15
50	·4357548	·484136	2·065531	·9000654	10
55	·4370634	·485933	2·057895	·8994307	5
60	·4383711	·487732	2·050303	·8987940	0′
					64 Deg.
26 Deg.					
0′	·4383711	·487732	2·050303	·8987940	60
5	·4396779	·489534	2·042757	·8981555	55
10	·4409838	·491338	2·035256	·8975151	50
15	·4422887	·493145	2·027799	·8968727	45
20	·4435927	·494954	2·020386	·8962285	40
25	·4448957	·496766	2·013016	·8955824	35
30	·4461978	·498581	2·005689	·8949344	30
35	·4474990	·500398	1·998405	·8942844	25
40	·4487992	·502218	1·991163	·8936326	20
45	·4500984	·504041	1·983963	·8929789	15
50	·4513967	·505866	1·976805	·8923234	10
55	·4526941	·507694	1·969687	·8916659	5
60	·4539905	·509525	1·962610	·8910065	0′
					63 Deg.
27 Deg.					
0′	·4539905	·509525	1·962610	·8910065	60
5	·4552859	·511358	1·955573	·8903453	55
10	·4565804	·513195	1·948577	·8896822	50
15	·4578739	·515033	1·941620	·8890171	45
20	·4591665	·516875	1·934702	·8883503	40
25	·4604580	·518719	1·927822	·8876815	35
30	·4617486	·520567	1·920982	·8870108	30
35	·4630382	·522417	1·914179	·8863383	25
40	·4643269	·524269	1·907414	·8856639	20
45	·4656145	·526125	1·900687	·8849876	15
50	·4669012	·527983	1·893997	·8843095	10
55	·4681869	·529845	1·887343	·8836295	5
60	·4694716	·531709	1·880726	·8829476	0′
					62 Deg.
28 Deg.					
0′	·4694716	·531709	1·880726	·8829476	60
5	·4707553	·533576	1·874145	·8822638	55
10	·4720380	·535446	1·867600	·8815782	50
15	·4733197	·537319	1·861090	·8808907	45
20	·4746004	·539195	1·854615	·8802014	40
25	·4758801	·541074	1·848176	·8795102	35
30	·4771588	·542955	1·841770	·8788171	30
35	·4784364	·544840	1·835399	·8781222	25
40	·4797131	·546728	1·829062	·8774254	20
45	·4809888	·548618	1·822759	·8767268	15
50	·4822634	·550512	1·816489	·8760263	10
55	·4835370	·552409	1·810252	·8753239	5
60	·4848096	·554309	1·804047	·8746197	0′
					61 Deg.
	Cosine.	Cotan.	Tang.	Sine.	

NATURAL SINES, TANGENTS, &c. (*continued*).

	Sine.	Tang.	Cotang.	Cosine.	
29 Deg.					
0′	·4848096	·554309	1·804047	·8746197	60
5	·4860812	·556211	1·797875	·8739137	55
10	·4873517	·558117	1·791736	·8732058	50
15	·4886212	·560026	1·785628	·8724960	45
20	·4898897	·561939	1·779552	·8717844	40
25	·4911572	·563854	1·773507	·8710710	35
30	·4924236	·565772	1·767494	·8703557	30
35	·4936889	·567694	1·761511	·8696386	25
40	·4949532	·569619	1·755559	·8689196	20
45	·4962165	·571547	1·749637	·8681988	15
50	·4974787	·573478	1·743745	·8674762	10
55	·4987399	·575412	1·737883	·8667517	5
60	·5000000	·577350	1·732050	·8660254	0′
					60 Deg.
30 Deg.					
0′	·5000000	.577350	1·732050	·8660254	60
5	·5012591	·579291	1·726247	·8652973	55
10	·5025170	·581235	1·720473	·8645673	50
15	·5037740	·583182	1·714728	·8638355	45
20	·5050298	·585133	1·709011	·8631019	40
25	·5062846	·587087	1·703323	·8623664	35
30	·5075384	·589045	1·697663	·8616292	30
35	·5087910	·591005	1·692030	·8608901	25
40	·5100426	·592969	1·686426	·8601491	20
45	·5112931	·594937	1·680848	·8594064	15
50	·5125425	·596908	1·675298	·8586619	10
55	·5137908	·598882	1·669775	·8579155	5
60	·5150381	·600860	1·664279	·8571673	0′
					59 Deg.
31 Deg.					
0′	·5150381	·600860	1·664279	·8571673	60
5	·5162842	·602841	1·658809	·8564173	55
10	·5175293	·604826	1·653366	·8556655	50
15	·5187733	·606814	1·647949	·8549119	45
20	·5200161	·608806	1·642557	·8541564	40
25	·5212579	·610801	1·637191	·8533992	35
30	·5224986	·612800	1·631851	·8526402	30
35	·5237381	·614803	1·626536	·8518793	25
40	·5249766	·616809	1·621246	·8511167	20
45	·5262139	·618818	1·615982	·8503522	15
50	·5274502	·620832	1·610741	·8495860	10
55	·5286853	·622848	1·605526	·8488179	5
60	·5299193	·624860	1·600334	·8480481	0′
					58 Deg.
32 Deg.					
0′	·5299193	·624869	1·600334	·8480481	60
5	·5311521	·626893	1·595167	·8472765	55
10	·5323839	·628921	1·590023	·8465030	50
15	·5336145	·630953	1·584904	·8457278	45
20	·5348440	·632988	1·579807	·8449508	40
25	·5360724	·635027	1·574735	·8441720	35
30	·5372996	·637070	1·569685	·8433914	30
35	·5385257	·639116	1·564659	·8426091	25
40	·5397507	·641167	1·559655	·8418249	20
45	·5409745	·643221	1·554674	·8410390	15
50	·5421971	·645279	1·549715	·8402513	10
55	·5434187	·647341	1·544779	·8394618	5
60	·5446390	·649407	1·539865	·8386706	0′
					57 Deg.
	Cosine.	Cotan.	Tang.	Sine.	

	Sine.	Tang.	Cotang.	Cosine.	
33 Deg.					
0′	·5446390	·649407	1·539865	·8386706	60
5	·5458583	·651477	1·534972	·8378775	55
10	·5470763	·653551	1·530102	·8370827	50
15	·5482932	·655628	1·525253	·8362862	45
20	·5495090	·657710	1·520426	·8354878	40
25	·5507236	·659796	1·515620	·8346877	35
30	·5519370	·661885	1·510835	·8338858	30
35	·5531492	·663979	1·506071	·8330822	25
40	·5543603	·666076	1·501328	·8322768	20
45	·5555702	·668178	1·496605	·8314696	15
50	·5567790	·670284	1·491903	·8306607	10
55	·5579865	·672394	1·487222	·8298500	5
60	·5591929	·674508	1·482561	·8290376	0′
					56 Deg.
34 Deg.					
0′	·5591929	·674508	1·482561	·8290376	60
5	·5603981	·676626	1·477919	·8282234	55
10	·5616021	·678749	1·473298	·8274074	50
15	·5628049	·680875	1·468696	·8265897	45
20	·5640066	·683006	1·464114	·8257703	40
25	·5652070	·685141	1·459552	·8249491	35
30	·5664062	·687281	1·455009	·8241262	30
35	·5676043	·689424	1·450485	·8233015	25
40	·5688011	·691572	1·445980	·8224751	20
45	·5699968	·693724	1·441494	·8216469	15
50	·5711912	·695881	1·437026	·8208170	10
55	·5723844	·698042	1·432578	·8199854	5
60	·5735764	·700207	1·428148	·8191520	0′
					55 Deg.
35 Deg.					
0′	·5735764	·700207	1·428148	·8191520	60
5	·5747672	·702377	1·423736	·8183169	55
10	·5759568	·704551	1·419342	·8174801	50
15	·5771452	·706730	1·414967	·8166416	45
20	·5783323	·708913	1·410609	·8158013	40
25	·5795183	·711100	1·406270	·8149593	35
30	·5807030	·713293	1·401948	·8141155	30
35	·5818864	·715489	1·397644	·8132701	25
40	·5830687	·717691	1·393357	·8124229	20
45	·5842497	·719897	1·389087	·8115740	15
50	·5854294	·722107	1·384835	·8107284	10
55	·5866080	·724322	1·380600	·8098710	5
60	·5877853	·726542	1·376381	·8090170	0′
					54 Deg.
36 Deg.					
0′	·5877853	·726542	1·376381	·8090170	60
5	·5889613	·728767	1·372180	·8081612	55
10	·5901361	·730996	1·367995	·8073038	50
15	·5913096	·733230	1·363827	·8064446	45
20	·5924819	·735469	1·359676	·8055837	40
25	·5936530	·737712	1·355541	·8047211	35
30	·5948228	·739961	1·351422	·8038569	30
35	·5959913	·742214	1·347319	·8029909	25
40	·5971586	·744472	1·343233	·8021232	20
45	·5983246	·746735	1·339162	·8012538	15
50	·5994893	·749003	1·335107	·8003827	10
55	·6006528	·751276	1·331068	·7995100	5
60	·6018150	·753554	1·327044	·7986355	0′
					53 Deg.
	Cosine.	Cotan.	Tang.	Sine.	

NATURAL SINES, TANGENTS, &c. (*continued*).

	Sine.	Tang.	Cotang.	Cosine.	
37 Deg.					
0′	·6018150	·753554	1·327044	·7986355	60
5	·6029760	·755836	1·323036	·7977594	55
10	·6041356	·758124	1·319044	·7968815	50
15	·6052940	·760417	1·315066	·7960020	45
20	·6064511	·762715	1·311104	·7951208	40
25	·6076069	·765018	1·307157	·7942379	35
30	·6087614	·767327	1·303225	·7933533	30
35	·6099147	·769640	1·299308	·7924671	25
40	·6110666	·771958	1·295405	·7915792	20
45	·6122173	·774282	1·291517	·7906896	15
50	·6133666	·776611	1·287644	·7897983	10
55	·6145147	·778946	1·283786	·7889054	5
60	·6156615	·781285	1·279941	·7880108	0′
					52 Deg.
38 Deg.					
0′	·6156615	·781285	1·279941	·7880108	60
5	·6168069	·783630	1·276111	·7871145	55
10	·6179511	·785980	1·272295	·7862165	50
15	·6190939	·788336	1·268494	·7853169	45
20	·6202355	·790697	1·264706	·7844157	40
25	·6213757	·793064	1·260932	·7835127	35
30	·6225146	·795435	1·257172	·7826082	30
35	·6236522	·797813	1·253426	·7817019	25
40	·6247885	·800196	1·249693	·7807940	20
45	·6259235	·802584	1·245974	·7798845	15
50	·6270571	·804979	1·242268	·7789733	10
55	·6281894	·807378	1·238576	·7780604	5
60	·6293204	·809784	1·234897	·7771460	0′
					51 Deg.
39 Deg.					
0′	·6293204	·809784	1·234897	·7771460	60
5	·6304500	·812195	1·231231	·7762298	55
10	·6315784	·814611	1·227578	·7753121	50
15	·6327053	·817034	1·223938	·7743926	45
20	·6338310	·819462	1·220312	·7734716	40
25	·6349553	·821896	1·216698	·7725489	35
30	·6360782	·824336	1·213097	·7716246	30
35	·6371998	·826782	1·209508	·7706986	25
40	·6383201	·829233	1·205932	·7697710	20
45	·6394390	·831691	1·202369	·7688418	15
50	·6405566	·834154	1·198818	·7679110	10
55	·6416728	·836624	1·195279	·7669785	5
60	·6427876	·839099	1·191753	·7660444	0′
					50 Deg.
40 Deg.					
0′	·6427876	·839099	1·191753	·7660444	60
5	·6439011	·841581	1·188239	·7651087	55
10	·6450132	·844068	1·184737	·7641714	50
15	·6461240	·846562	1·181247	·7632325	45
20	·6472334	·849062	1·177769	·7622919	40
25	·6483414	·851568	1·174303	·7613497	35
30	·6494480	·854080	1·170849	·7604060	30
35	·6505533	·856599	1·167407	·7594606	25
40	·6516572	·859124	1·163976	·7585136	20
45	·6527598	·861655	1·160557	·7575650	15
50	·6538609	·864192	1·157149	·7566148	10
55	·6549607	·866736	1·153753	·7556630	5
60	·6560590	·869286	1·150368	·7547096	0′
					49 Deg.
	Cosine.	Cotan.	Tang.	Sine.	

	Sine.	Tang.	Cotang.	Cosine.	
41 Deg.					
0′	·6560590	·869286	1·150368	·7547096	60
5	·6571560	·871843	1·146994	·7537546	55
10	·6582516	·874406	1·143632	·7527980	50
15	·6593458	·876976	1·140281	·7518398	45
20	·6604386	·879552	1·136941	·7508800	40
25	·6615300	·882135	1·133612	·7499187	35
30	·6626200	·884725	1·130294	·7489557	30
35	·6637087	·887321	1·126987	·7479912	25
40	·6647959	·889924	1·123690	·7470251	20
45	·6658817	·892534	1·120405	·7460574	15
50	·6669661	·895150	1·117130	·7450881	10
55	·6680490	·897773	1·113866	·7441173	5
60	·6691306	·900404	1·110612	·7431448	0′
					48 Deg.
42 Deg.					
0′	·6691306	·900404	1·110612	·7431448	60
5	·6702108	·903041	1·107369	·7421708	55
10	·6712895	·905685	1·104136	·7411953	50
15	·6723668	·908336	1·100914	·7402181	45
20	·6734427	·910994	1·097702	·7392394	40
25	·6745172	·913659	1·094500	·7382592	35
30	·6755902	·916331	1·091308	·7372773	30
35	·6766618	·919010	1·088126	·7362940	25
40	·6777320	·921696	1·084955	·7353090	20
45	·6788007	·924390	1·081793	·7343225	15
50	·6798681	·927091	1·078642	·7333345	10
55	·6809339	·929799	1·075500	·7323449	5
60	·6819984	·932515	1·072368	·7313537	0′
					47 Deg.
43 Deg.					
0′	·6819984	·932515	1·072368	·7313537	60
5	·6830613	·935238	1·069246	·7303610	55
10	·6841229	·937968	1·066134	·7293668	50
15	·6851830	·940706	1·063031	·7283710	45
20	·6862416	·943451	1·059938	·7273736	40
25	·6872988	·946204	1·056854	·7263748	35
30	·6883546	·948964	1·053780	·7253744	30
35	·6894089	·951732	1·050715	·7243724	25
40	·6904617	·954508	1·047659	·7233690	20
45	·6915131	·957291	1·044613	·7223640	15
50	·6925630	·960082	1·041576	·7213574	10
55	·6936114	·962881	1·038548	·7203494	5
60	·6946584	·965688	1·035530	·7193398	0′
					46 Deg.
44 Deg.					
0′	·6946584	·965688	1·035530	·7193398	60
5	·6957039	·968503	1·032520	·7183287	55
10	·6967479	·971326	1·029520	·7173161	50
15	·6977905	·974156	1·026528	·7163019	45
20	·6988315	·976995	1·023546	·7152863	40
25	·6998711	·979842	1·020572	·7142691	35
30	·7009093	·982697	1·017607	·7132504	30
35	·7019459	·985560	1·014651	·7122303	25
40	·7029811	·988431	1·011703	·7112086	20
45	·7040147	·991311	1·008764	·7101854	15
50	·7050469	·994199	1·005834	·7091607	10
55	·7060776	·997095	1·002913	·7081345	5
60	·7071068	1·00000	1·000000	·7071068	0′
					45 Deg.
	Cosine.	Cotan.	Tang.	Sine.	

LOGARITHMIC SINES, TANGENTS, &c

	Sine.	Tang.	Cotang.	Cosine.	
0 Deg.					
0'	Inf Neg	Inf. Neg.	Infinite	10·00000	60
1	6·46373	6·46373	13·53627	10·00000	59
2	6·76476	6·76476	13·23524	10·00000	58
3	6·94085	6·94085	13·05915	10·00000	57
4	7·06579	7·06579	12·93421	10·00000	56
5	7·16270	7·16270	12·83730	10·00000	55
6	7·24188	7·24188	12·75812	10·00000	54
7	7·30882	7·30882	12·69118	10·00000	53
8	7·36682	7·36682	12·63318	10·00000	52
9	7·41797	7·41797	12·58203	10·00000	51
10	7·46373	7·46373	12·53627	10·00000	50
11	7·50512	7·50512	12·49488	10·00000	49
12	7·54291	7·54291	12·45709	10·00000	48
13	7·57767	7·57767	12·42233	10·00000	47
14	7·60985	7·60986	12·39014	10·00000	46
15	7·63982	7·63982	12·36018	10·00000	45
16	7·66784	7·66785	12·33215	10·00000	44
17	7·69417	7·69418	12·30582	9·99999	43
18	7·71900	7·71900	12·28100	9·99999	42
19	7·74248	7·74248	12·25752	9·99999	41
20	7·76475	7·76476	12·23524	9·99999	40
21	7·78594	7·78595	12·21405	9·99999	39
22	7·80615	7·80615	12·19385	9·99999	38
23	7·82545	7·82546	12·17454	9·99999	37
24	7·84393	7·84394	12·15606	9·99999	36
25	7·86166	7·86167	12·13833	9·99999	35
26	7·87870	7·87871	12·12129	9·99999	34
27	7·89509	7·89510	12·10490	9·99999	33
28	7·91088	7·91089	12·08911	9·99999	32
29	7·92612	7·92613	12·07387	9·99998	31
30	7·94084	7·94086	12·05914	9·99998	30
31	7·95508	7·95510	12·04490	9·99998	29
32	7·96887	7·96889	12·03111	9·99998	28
33	7·98223	7·98225	12·01775	9·99998	27
34	7·99520	7·99522	12·00478	9·99998	26
35	8·00779	8·00781	11·99219	9·99998	25
36	8·02002	8·02004	11·97996	9·99998	24
37	8·03192	8·03194	11·96806	9·99997	23
38	8·04350	8·04353	11·95647	9·99997	22
39	8·05478	8·05481	11·94519	9·99997	21
40	8·06578	8·06581	11·93419	9·99997	20
41	8·07650	8·07653	11·92347	9·99997	19
42	8·08696	8·08700	11·91300	9·99997	18
43	8·09718	8·09722	11·90278	9·99997	17
44	8·10717	8·10720	11·89280	9·99996	16
45	8·11693	8·11696	11·88304	9·99996	15
46	8·12647	8·12651	11·87349	9·99996	14
47	8·13581	8·13585	11·86415	9·99996	13
48	8·14495	8·14500	11·85500	9·99996	12
49	8·15391	8·15395	11·84605	9·99996	11
50	8·16268	8·16273	11·83727	9·99995	10
51	8·17128	8·17133	11·82867	9·99995	9
52	8·17971	8·17976	11·82024	9·99995	8
53	8·18798	8·18804	11·81196	9·99995	7
54	8·19610	8·19616	11·80384	9·99995	6
55	8·20407	8·20413	11·79587	9·99994	5
56	8·21189	8·21195	11·78805	9·99994	4
57	8·21958	8·21964	11·78036	9·99994	3
58	8·22713	8·22720	11·77280	9·99994	2
59	8·23456	8·23462	11·76538	9·99994	1
60	8·24186	8·24192	11·75808	9·99993	0'
					89 Deg.
	Cosine.	Cotang.	Tang.	Sine.	

	Sine.	Tang.	Cotang.	Cosine.	
1 Deg.					
0'	8·24186	8·24192	11·75808	9·99993	60
5	8·27661	8·27669	11·72331	9·99992	55
10	8·30879	8·30888	11·69112	9·99991	50
15	8·33875	8·33886	11·66114	9·99990	45
20	8·36678	8·36689	11·63311	9·99988	40
25	8·39310	8·39323	11·60677	9·99987	35
30	8·41792	8·41807	11·58193	9·99985	30
35	8·44139	8·44156	11·55844	9·99983	25
40	8·46366	8·46385	11·53615	9·99982	20
45	8·48485	8·48505	11·51495	9·99980	15
50	8·50504	8·50527	11·49473	9·99978	10
55	8·52434	8·52459	11·47541	9·99976	5
60	8·54282	8·54308	11·45692	9·99974	0'
					88 Deg.
2 Deg.					
0'	8·54282	8·54308	11·45692	9·99974	60
5	8·56054	8·56083	11·43917	9·99971	55
10	8·57757	8·57788	11·42212	9·99969	50
15	8·59395	8·59428	11·40572	9·99967	45
20	8·60973	8·61009	11·38991	9·99964	40
25	8·62497	8·62535	11·37465	9·99961	35
30	8·63968	8·64009	11·35991	9·99959	30
35	8·65391	8·65435	11·34565	9·99956	25
40	8·66769	8·66816	11·33184	9·99953	20
45	8·68104	8·68154	11·31846	9·99950	15
50	8·69400	8·69453	11·30547	9·99947	10
55	8·70658	8·70714	11·29286	9·99944	5
60	8·71880	8·71940	11·28060	9·99940	0'
					87 Deg.
3 Deg.					
0'	8·71880	8·71940	11·28060	9·99940	60
5	8·73069	8·73132	11·26868	9·99937	55
10	8·74226	8·74292	11·25708	9·99934	50
15	8·75353	8·75423	11·24577	9·99930	45
20	8·76451	8·76525	11·23475	9·99926	40
25	8·77522	8·77600	11·22400	9·99923	35
30	8·78568	8·78649	11·21351	9·99919	30
35	8·79588	8·79673	11·20327	9·99915	25
40	8·80585	8·80674	11·19326	9·99911	20
45	8·81560	8·81653	11·18347	9·99907	15
50	8·82513	8·82610	11·17390	9·99903	10
55	8·83446	8·83547	11·16453	9·99898	5
60	8·84358	8·84464	11·15536	9·99894	0'
					86 Deg.
4 Deg.					
0'	8·84358	8·84464	11·15536	9·99894	60
5	8·85252	8·85363	11·14637	9·99890	55
10	8·86128	8·86243	11·13757	9·99885	50
15	8·86987	8·87106	11·12894	9·99880	45
20	8·87829	8·87953	11·12047	9·99876	40
25	8·88654	8·88783	11·11217	9·99871	35
30	8·89464	8·89598	11·10402	9·99866	30
35	8·90260	8·90399	11·09601	9·99861	25
40	8·91040	8·91185	11·08815	9·99856	20
45	8·91807	8·91957	11·08043	9·99851	15
50	8·92561	8·92716	11·07284	9·99845	10
55	8·93301	8·93462	11·06538	9·99840	5
60	8·94030	8·94195	11·05805	9·99834	0'
					85 Deg.
	Cosine	Cotang.	Tang.	Sine.	

LOGARITHMIC SINES, TANGENTS, &c. (*continued*).

	Sine.	Tang.	Cotang.	Cosine.	
5 Deg.					
0′	8·94030	8·94195	11·05805	9·99834	60
5	8·94746	8·94917	11·05083	9·99829	55
10	8·95450	8·95627	11·04373	9·99823	50
15	8·96143	8·96325	11·03675	9·99817	45
20	8·96825	8·97013	11·02987	9·99812	40
25	8·97496	8·97691	11·02309	9·99806	35
30	8·98157	8·98358	11·01642	9·99800	30
35	8·98808	8·99015	11·00985	9·99793	25
40	8·99450	8·99662	11·00338	9·99787	20
45	9·00082	9·00301	10·99699	9·99781	15
50	9·00704	9·00930	10·99070	9·99775	10
55	9·01318	9·01550	10·98450	9·99768	5
60	9·01923	9·02162	10·97838	9·99761	0′
					84 Deg.
6 Deg.					
0′	9·01923	9·02162	10·97838	9·99761	60
5	9·02520	9·02766	10·97234	9·99755	55
10	9·03109	9·03361	10·96639	9·99748	50
15	9·03690	9·03948	10·96052	9·99741	45
20	9·04262	9·04528	10·95472	9·99734	40
25	9·04828	9·05101	10·94899	9·99727	35
30	9·05386	9·05666	10·94334	9·99720	30
35	9·05937	9·06224	10·93776	9·99713	25
40	9·06481	9·06775	10·93225	9·99705	20
45	9·07018	9·07320	10·92680	9·99698	15
50	9·07548	9·07858	10·92142	9·99690	10
55	9·08072	9·08389	10·91611	9·99683	5
60	9·08589	9·08914	10·91086	9·99675	0′
					83 Deg.
7 Deg.					
0′	9·08589	9·08914	10·91086	9·99675	60
5	9·09101	9·09434	10·90566	9·99667	55
10	9·09606	9·09947	10·90053	9·99659	50
15	9·10106	9·10454	10·89546	9·99651	45
20	9·10599	9·10956	10·89044	9·99643	40
25	9·11087	9·11452	10·88548	9·99635	35
30	9·11570	9·11943	10·88057	9·99627	30
35	9·12047	9·12428	10·87572	9·99618	25
40	9·12519	9·12909	10·87091	9·99610	20
45	9·12985	9·13384	10·86616	9·99601	15
50	9·13447	9·13854	10·86146	9·99593	10
55	9·13904	9·14320	10·85680	9·99584	5
60	9·14356	9·14780	10·85220	9·99575	0′
					82 Deg.
8 Deg.					
0′	9·14356	9·14780	10·85220	9·99575	60
5	9·14803	9·15236	10·84764	9·99566	55
10	9·15245	9·15688	10·84312	9·99557	50
15	9·15683	9·16135	10·83865	9·99548	45
20	9·16116	9·16577	10·83423	9·99539	40
25	9·16545	9·17016	10·82984	9·99530	35
30	9·16970	9·17450	10·82550	9·99520	30
35	9·17391	9·17880	10·82120	9·99511	25
40	9·17807	9·18306	10·81694	9·99501	20
45	9·18220	9·18728	10·81272	9·99492	15
50	9·18628	9·19146	10·80854	9·99482	10
55	9·19033	9·19561	10·80439	9·99472	5
60	9·19433	9·19971	10·80029	9·99462	0′
					81 Deg.
	Cosine.	**Cotang.**	**Tang.**	**Sine.**	

	Sine.	Tang.	Cotang.	Cosine.	
9 Deg.					
0′	9·19433	9·19971	10·80029	0·99462	60
5	9·19830	9·20378	10·79622	9·99452	55
10	9·20223	9·20782	10·79218	9·99442	50
15	9·20613	9·21182	10·78818	9·99432	45
20	9·20999	9·21578	10·78422	9·99421	40
25	9·21382	9·21971	10·78029	9·99411	35
30	9·21761	9·22361	10·77639	9·99400	30
35	9·22137	9·22747	10·77253	9·99390	25
40	9·22509	9·23130	10·76870	9·99379	20
45	9·22878	9·23510	10·76490	9·99368	15
50	9·23244	9·23887	10·76113	9·99357	10
55	9·23607	9·24261	10·75739	9·99346	5
60	9·23967	9·24632	10·75368	9·99335	0′
					80 Deg
10 Deg.					
0′	9·23967	9·24632	10·75368	9·99335	60
5	9·24324	9·25000	10·75000	9·99324	55
10	9·24677	9·25365	10·74635	9·99313	50
15	9·25028	9·25727	10·74273	9·99301	45
20	9·25376	9·26086	10·73914	9·99290	40
25	9·25721	9·26443	10·73557	9·99278	35
30	9·26063	9·26797	10·73203	9·99267	30
35	9·26403	9·27148	10·72852	9·99255	25
40	9·26739	9·27496	10·72504	9·99243	20
45	9·27073	9·27842	10·72158	9·99231	15
50	9·27405	9·28186	10·71814	9·99219	10
55	9·27734	9·28527	10·71473	9·99207	5
60	9·28060	9·28865	10·71135	9·99195	0′
					79 Deg.
11 Deg.					
0′	9·28060	9·28865	10·71135	9·99195	60
5	9·28384	9·29201	10·70799	9·99182	55
10	9·28705	9·29535	10·70465	9·99170	50
15	9·29024	9·29866	10·70134	9·99157	45
20	9·29340	9·30195	10·69805	9·99145	40
25	9·29654	9·30522	10·69478	9·99132	35
30	9·29966	9·30846	10·69154	9·99119	30
35	9·30275	9·31168	10·68832	9·99106	25
40	9·30582	9·31489	10·68511	9·99093	20
45	9·30887	9·31806	10·68194	9·99080	15
50	9·31189	9·32122	10·67878	9·99067	10
55	9·31490	9·32436	10·67564	9·99054	5
60	9·31788	9·32747	10·67253	9·99040	0′
					78 Deg.
12 Deg.					
0′	9·31788	9·32747	10·67253	9·99040	60
5	9·32084	9·33057	10·66943	9·99027	55
10	9·32378	9·33365	10·66635	9·99013	50
15	9·32670	9·33670	10·66330	9·99000	45
20	9·32960	9·33974	10·66026	9·98986	40
25	9·33248	9·34276	10·65724	9·98972	35
30	9·33534	9·34576	10·65424	9·98958	30
35	9·33818	9·34874	10·65126	9·98944	25
40	9·34100	9·35170	10·64830	9·98930	20
45	9·34380	9·35464	10·64536	9·98916	15
50	9·34658	9·35757	10·64243	9·98901	10
55	9·34934	9·36047	10·63953	9·98887	5
60	9·35209	9·36336	10·63664	9·98872	0′
					77 Deg.
	Cosine.	**Cotang.**	**Tang.**	**Sine.**	

Logarithmic Sines, Tangents, &c. (*continued*).

	Sine.	Tang.	Cotang.	Cosine.	
13 Deg.					
0′	9·35209	9·36336	10·63664	9·98872	60
5	9·35481	9·36624	10·63376	9·98858	55
10	9·35752	9·36909	10·63091	9·98843	50
15	9·36022	9·37193	10·62807	9·98828	45
20	9·36289	9·37476	10·62524	9·98813	40
25	9·36555	9·37756	10·62244	9·98798	35
30	9·36819	9·38035	10·61965	9·98783	30
35	9·37081	9·38313	10·61687	9·98768	25
40	9·37341	9·38589	10·61411	9·98753	20
45	9·37600	9·38863	10·61137	9·98737	15
50	9·37858	9·39136	10·60864	9·98722	10
55	9·38113	9·39407	10·60593	9·98706	5
60	9·38368	9·39677	10·60323	9·98690	0′
					76 Deg.
14 Deg.					
0′	9·38368	9·39677	10·60323	9·98690	60
5	9·38620	9·39945	10·60055	9·98675	55
10	9·38871	9·40212	10·59788	9·98659	50
15	9·39121	9·40478	10·59522	9·98643	45
20	9·39369	9·40742	10·59258	9·98627	40
25	9·39615	9·41005	10·58995	9·98610	35
30	9·39860	9·41266	10·58734	9·98594	30
35	9·40103	9·41526	10·58474	9·98578	25
40	9·40346	9·41784	10·58216	9·98561	20
45	9·40586	9·42041	10·57959	9·98545	15
50	9·40825	9·42297	10·57703	9·98528	10
55	9·41063	9·42552	10·57448	9·98511	5
60	9·41300	9·42805	10·57195	9·98494	0′
					75 Deg.
15 Deg.					
0′	9·41300	9·42805	10·57195	9·98494	60
5	9·41535	9·43057	10·56943	9·98477	55
10	9·41768	9·43308	10·56692	9·98460	50
15	9·42001	9·43558	10·56442	9·98443	45
20	9·42232	9·43806	10·56194	9·98426	40
25	9·42461	9·44053	10·55947	9·98409	35
30	9·42690	9·44299	10·55701	9·98391	30
35	9·42917	9·44544	10·55456	9·98373	25
40	9·43143	9·44787	10·55213	9·98356	20
45	9·43367	9·45029	10·54971	9·98338	15
50	9·43591	9·45271	10·54729	9·98320	10
55	9·43813	9·45511	10·54489	9·98302	5
60	9·44034	9·45750	10·54250	9·98284	0′
					74 Deg.
16 Deg.					
0′	9·44034	9·45750	10·54250	9·98284	60
5	9·44253	9·45987	10·54013	9·98266	55
10	9·44472	9·46224	10·53776	9·98248	50
15	9·44689	9·46460	10·53540	9·98229	45
20	9·44905	9·46694	10·53306	9·98211	40
25	9·45120	9·46928	10·53072	9·98192	35
30	9·45334	9·47160	10·52840	9·98174	30
35	9·45547	9·47392	10·52608	9·98155	25
40	9·45758	9·47622	10·52378	9·98136	20
45	9·45969	9·47852	10·52148	9·98117	15
50	9·46178	9·48080	10·51920	9·98098	10
55	9·46386	9·48307	10·51693	9·98079	5
60	9·46594	9·48534	10·51466	9·98060	0′
					73 Deg.
	Cosine.	Cotang.	Tang.	Sine.	

	Sine.	Tang.	Cotang.	Cosine.	
17 Deg.					
0′	9·46594	9·48534	10·51466	9·98060	60
5	9·46800	9·48759	10·51241	9·98040	55
10	9·47005	9·48984	10·51016	9·98021	50
15	9·47209	9·49207	10·50793	9·98001	45
20	9·47411	9·49430	10·50570	9·97982	40
25	9·47613	9·49652	10·50348	9·97962	35
30	9·47814	9·49872	10·50128	9·97942	30
35	9·48014	9·50092	10·49908	9·97922	25
40	9·48213	9·50311	10·49689	9·97902	20
45	9·48411	9·50529	10·49471	9·97882	15
50	9·48607	9·50746	10·49254	9·97861	10
55	9·48803	9·50962	10·49038	9·97841	5
60	9·48998	9·51178	10·48822	9·97821	0′
					72 Deg.
18 Deg.					
0′	9·48998	9·51178	10·48822	9·97821	60
5	9·49192	9·51392	10·48608	9·97800	55
10	9·49385	9·51606	10·48394	9·97779	50
15	9·49577	9·51819	10·48181	9·97759	45
20	9·49768	9·52031	10·47969	9·97738	40
25	9·49958	9·52242	10·47758	9·97717	35
30	9·50148	9·52452	10·47548	9·97696	30
35	9·50336	9·52661	10·47339	9·97674	25
40	9·50523	9·52870	10·47130	9·97653	20
45	9·50710	9·53078	10·46922	9·97632	15
50	9·50896	9·53285	10·46715	9·97610	10
55	9·51080	9·53492	10·46508	9·97589	5
60	9·51264	9·53697	10·46303	9·97567	0′
					71 Deg.
19 Deg.					
0′	9·51264	9·53697	10·46303	9·97567	60
5	9·51447	9·53902	10·46098	9·97545	55
10	9·51629	9·54106	10·45894	9·97523	50
15	9·51811	9·54309	10·45691	9·97501	45
20	9·51991	9·54512	10·45488	9·97479	40
25	9·52171	9·54714	10·45286	9·97457	35
30	9·52350	9·54915	10·45085	9·97435	30
35	9·52527	9·55115	10·44885	9·97412	25
40	9·52705	9·55315	10·44685	9·97390	20
45	9·52881	9·55514	10·44486	9·97367	15
50	9·53056	9·55712	10·44288	9·97344	10
55	9·53231	9·55910	10·44090	9·97322	5
60	9·53405	9·56107	10·43893	9·97299	0′
					70 Deg.
20 Deg.					
0′	9·53405	9·56107	10·43893	9·97299	60
5	9·53578	9·56303	10·43697	9·97276	55
10	9·53751	9·56498	10·43502	9·97252	50
15	9·53922	9·56693	10·43307	9·97229	45
20	9·54093	9·56887	10·43113	9·97206	40
25	9·54263	9·57081	10·42919	9·97182	35
30	9·54433	9·57274	10·42726	9·97159	30
35	9·54601	9·57466	10·42534	9·97135	25
40	9·54769	9·57658	10·42342	9·97111	20
45	9·54936	9·57849	10·42151	9·97087	15
50	9·55102	9·58039	10·41961	9·97063	10
55	9·55268	9·58229	10·41771	9·97039	5
60	9·55433	9·58418	10·41582	9·97015	0′
					69 Deg.
	Cosine.	Cotang.	Tang.	Sine.	

LOGARITHMIC SINES, TANGENTS, &c. (*continued*).

	Sine.	Tang.	Cotang.	Cosine.	
21 Deg.					
0′	9·55433	9·58418	10·41582	9·97015	60
5	9·55597	9·58606	10·41394	9·96991	55
10	9·55761	9·58794	10·41206	9·96966	50
15	9·55923	9·58981	10·41019	9·96942	45
20	9·56085	9·59168	10·40832	9·96917	40
25	9·56247	9·59354	10·40646	9·96893	35
30	9·56408	9·59540	10·40460	9·96868	30
35	9·56568	9·59725	10·40275	9·96843	25
40	9·56727	9·59909	10·40091	9·96818	20
45	9·56886	9·60093	10·39907	9·96793	15
50	9·57044	9·60276	10·39724	9·96767	10
55	9·57201	9·60459	10·39541	9·96742	5
60	9·57358	9·60641	10·39359	9·96717	0′
					68 Deg.
22 Deg.					
0′	9·57358	9·60641	10·39359	9·96717	60
5	9·57514	9·60823	10·39177	9·96691	55
10	9·57669	9·61004	10·38996	9·96665	50
15	9·57824	9·61184	10·38816	9·96640	45
20	9·57978	9·61364	10·38636	9·96614	40
25	9·58131	9·61544	10·38456	9·96588	35
30	9·58284	9·61722	10·38278	9·96562	30
35	9·58436	9·61901	10·38099	9·96535	25
40	9·58588	9·62079	10·37921	9·96509	20
45	9·58739	9·62256	10·37744	9·96483	15
50	9·58889	9·62433	10·37567	9·96456	10
55	9·59039	9·62609	10·37391	9·96429	5
60	9·59188	9·62785	10·37215	9·96403	0′
					67 Deg.
23 Deg.					
0′	9·59188	9·62785	10·37215	9·96403	60
5	9·59336	9·62961	10·37039	9·96376	55
10	9·59484	9·63135	10·36865	9·96349	50
15	9·59632	9·63310	10·36690	9·96322	45
20	9·59778	9·63484	10·36516	9·96294	40
25	9·59924	9·63657	10·36343	9·96267	35
30	9·60070	9·63830	10·36170	9·96240	30
35	9·60215	9·64003	10·35997	9·96212	25
40	9·60359	9·64175	10·35825	9·96185	20
45	9·60503	9·64346	10·35654	9·96157	15
50	9·60646	9·64517	10·35483	9·96129	10
55	9·60789	9·64688	10·35312	9·96101	5
60	9·60931	9·64858	10·35142	9·96073	0′
					66 Deg.
24 Deg.					
0′	9·60931	9·64858	10·35142	9·96073	60
5	9·61073	9·65028	10·34972	9·96045	55
10	9·61214	9·65197	10·34803	9·96017	50
15	9·61354	9·65366	10·34634	9·95988	45
20	9·61494	9·65535	10·34465	9·95960	40
25	9·61634	9·65703	10·34297	9·95931	35
30	9·61773	9·65870	10·34130	9·95902	30
35	9·61911	9·66038	10·33962	9·95873	25
40	9·62049	9·66204	10·33796	9·95844	20
45	9·62186	9·66371	10·33629	9·95815	15
50	9·62323	9·66537	10·33463	9·95786	10
55	9·62459	9·66702	10·33298	9·95757	5
60	9·62595	9·66867	10·33133	9·95728	0′
					65 Deg.
	Cosine.	Cotang.	Tang.	Sine.	

	Sine.	Tang.	Cotang.	Cosine.	
25 Deg.					
0′	9·62595	9·66867	10·33133	9·95728	60
5	9·62730	9·67032	10·32968	9·95698	55
10	9·62865	9·67196	10·32804	9·95668	50
15	9·62999	9·67360	10·32640	9·95639	45
20	9·63133	9·67524	10·32476	9·95609	40
25	9·63266	9·67687	10·32313	9·95579	35
30	9·63398	9·67850	10·32150	9·95549	30
35	9·63531	9·68012	10·31988	9·95519	25
40	9·63662	9·68174	10·31826	9·95488	20
45	9·63794	9·68336	10·31664	9·95458	15
50	9·63924	9·68497	10·31503	9·95427	10
55	9·64054	9·68658	10·31342	9·95397	5
60	9·64184	9·68818	10·31182	9·95366	0′
					64 Deg.
26 Deg.					
0′	9·64184	9·68818	10·31182	9·95366	60
5	9·64313	9·68978	10·31022	9·95335	55
10	9·64442	9·69138	10·30862	9·95304	50
15	9·64571	9·69298	10·30702	9·95273	45
20	9·64698	9·69457	10·30543	9·95242	40
25	9·64826	9·69615	10·30385	9·95211	35
30	9·64953	9·69774	10·30226	9·95179	30
35	9·65079	9·69932	10·30068	9·95148	25
40	9·65205	9·70089	10·29911	9·95116	20
45	9·65331	9·70247	10·29753	9·95084	15
50	9·65456	9·70404	10·29596	9·95052	10
55	9·65580	9·70560	10·29440	9·95020	5
60	9·65705	9·70717	10·29283	9·94988	0′
					63 Deg.
27 Deg.					
0′	9·65705	9·70717	10·29283	9·94988	60
5	9·65828	9·70873	10·29127	9·94956	55
10	9·65952	9·71028	10·28972	9·94923	50
15	9·66075	9·71184	10·28816	9·94891	45
20	9·66197	9·71339	10·28661	9·94858	40
25	9·66319	9·71493	10·28507	9·94826	35
30	9·66441	9·71648	10·28352	9·94793	30
35	9·66562	9·71802	10·28198	9·94760	25
40	9·66682	9·71955	10·28045	9·94727	20
45	9·66803	9·72109	10·27891	9·94694	15
50	9·66922	9·72262	10·27738	9·94660	10
55	9·67042	9·72415	10·27585	9·94627	5
60	9·67161	9·72567	10·27433	9·94593	0′
					62 Deg.
28 Deg.					
0′	9·67161	9·72567	10·27433	9·94593	60
5	9·67280	9·72720	10·27280	9·94560	55
10	9·67398	9·72872	10·27128	9·94526	
15	9·67515	9·73023	10·26977	9·94492	
20	9·67633	9·73175	10·26825	9·94458	0
25	9·67750	9·73326	10·26674	9·94424	35
30	9·67866	9·73476	10·26524	9·94390	30
35	9·67982	9·73627	10·26373	9·94355	25
40	9·68098	9·73777	10·26223	9·94321	20
45	9·68213	9·73927	10·26073	9·94286	15
50	9·68328	9·74077	10·25923	9·94252	10
55	9·68443	9·74226	10·25774	9·94217	5
60	9·68557	9·74375	10·25625	9·94182	0′
					61 Deg.
	Cosine.	Cotang.	Tang.	Sine.	

LOGARITHMIC SINES, TANGENTS, &c. (*continued*).

	Sine.	Tang.	Cotang.	Cosine.	
29 Deg.					
0′	9·68557	9·74375	10·25625	9·94182	60
5	9·68671	9·74524	10·25476	9·94147	55
10	9·68784	9·74673	10·25327	9·94112	50
15	9·68897	9·74821	10·25179	9·94076	45
20	9·69010	9·74969	10·25031	9·94041	40
25	9·69122	9·75117	10·24883	9·94005	35
30	9·69234	9·75264	10·24736	9·93970	30
35	9·69345	9·75411	10·24589	9·93934	25
40	9·69456	9·75558	10·24442	9·93898	20
45	9·69567	9·75705	10·24295	9·93862	15
50	9·69677	9·75852	10·24148	9·93826	10
55	9·69787	9·75998	10·24002	9·93789	5
60	9·69897	9·76144	10·23856	9·93753	0′
					60 Deg.
30 Deg.					
0′	9·69897	9·76144	10·23856	9·93753	60
5	9·70006	9·76290	10·23710	9·93717	55
10	9·70115	9·76435	10·23565	9·93680	50
15	9·70224	9·76580	10·23420	9·93643	45
20	9·70332	9·76725	10·23275	9·93606	40
25	9·70439	9·76870	10·23130	9·93569	35
30	9·70547	9·77015	10·22985	9·93532	30
35	9·70654	9·77159	10·22841	9·93495	25
40	9·70761	9·77303	10·22697	9·93457	20
45	9·70867	9·77447	10·22553	9·93420	15
50	9·70973	9·77591	10·22409	9·93382	10
55	9·71079	9·77734	10·22266	9·93344	5
60	9·71184	9·77877	10·22123	9·93307	0′
					59 Deg.
31 Deg.					
0′	9·71184	9·77877	10·22123	9·93307	60
5	9·71289	9·78020	10·21980	9·93269	55
10	9·71393	9·78163	10·21837	9·93230	50
15	9·71498	9·78306	10·21694	9·93192	45
20	9·71602	9·78446	10·21552	9·93154	40
25	9·71705	9·78590	10·21410	9·93115	35
30	9·71809	9·78732	10·21268	9·93077	30
35	9·71911	9·78874	10·21126	9·93038	15
40	9·72014	9·79015	10·20985	9·92999	20
45	9·72116	9·79156	10·20844	9·92960	15
50	9·72218	9·79297	10·20703	9·92921	10
55	9·72320	9·79438	10·20562	9·92881	5
60	9·72421	9·79579	10·20421	9·92842	0′
					58 Deg.
32 Deg.					
0	9·72421	9·79579	10·20421	9·92842	60
5	9·72522	9·79719	10·20281	9·92803	55
10	9·72622	9·79860	10·20140	9·92763	50
15	9·72723	9·80000	10·20000	9·92723	45
20	9·72823	9·80140	10·19860	9·92683	40
25	9·72922	9·80279	10·19721	9·92643	35
30	9·73022	9·80419	10·19581	9·92603	30
35	9·73121	9·80558	10·19442	9·92563	25
40	9·73219	9·80697	10·19303	9·92522	20
45	9·73318	9·80836	10·19164	9·92482	15
50	9·73416	9·80975	10·19025	9·92441	10
55	9·73513	9·81113	10·18887	9·92400	5
60	9·73611	9·81252	10·18748	9·92359	0′
					57 Deg.
	Cosine.	Cotang.	Tang.	Sine.	

	Sine.	Tang.	Cotang.	Cosine.	
33 Deg.					
0′	9·73611	9·81252	10·18748	9·92359	60
5	9·73708	9·81390	10·18610	9·92318	55
10	9·73805	9·81528	10·18472	9·92277	50
15	9·73901	9·81666	10·18334	9·92235	45
20	9·73997	9·81803	10·18197	9·92194	40
25	9·74093	9·81941	10·18059	9·92152	35
30	9·74189	9·82078	10·17922	9·92111	30
35	9·74284	9·82215	10·17785	9·92069	25
40	9·74379	9·82352	10·17648	9·92027	20
45	9·74474	9·82489	10·17511	9·91985	15
50	9·74568	9·82626	10·17374	9·91942	10
55	9·74662	9·82762	10·17238	9·91900	5
60	9·74756	9·82899	10·17101	9·91857	0′
					56 Deg.
34 Deg.					
0′	9·74756	9·82899	10·17101	9·91857	60
5	9·74850	9·83035	10·16965	9·91815	55
10	9·74943	9·83171	10·16829	9·91772	50
15	9·75036	9·83307	10·16693	9·91729	45
20	9·75128	9·83442	10·16558	9·91686	40
25	9·75221	9·83578	10·16422	9·91643	35
30	9·75313	9·83713	10·16287	9·91599	30
35	9·75405	9·83849	10·16151	9·91556	25
40	9·75496	9·83984	10·16016	9·91512	20
45	9·75587	9·84119	10·15881	9·91469	15
50	9·75678	9·84254	10·15746	9·91425	10
55	9·75769	9·84388	10·15612	9·91381	5
60	9·75859	9·84523	10·15477	9·91336	0′
					55 Deg.
35 Deg.					
0′	9·75859	9·84523	10·15477	9·91336	60
5	9·75949	9·84657	10·15343	9·91292	55
10	9·76039	9·84791	10·15209	9·91248	50
15	9·76129	9·84925	10·15075	9·91203	45
20	9·76218	9·85059	10·14941	9·91158	40
25	9·76307	9·85193	10·14807	9·91114	35
30	9·76395	9·85327	10·14673	9·91069	30
35	9·76484	9·85460	10·14540	9·91023	25
40	9·76572	9·85594	10·14406	9·90978	20
45	9·76660	9·85727	10·14273	9·90933	15
50	9·76747	9·85860	10·14140	9·90887	10
55	9·76835	9·85993	10·14007	9·90842	5
60	9·76922	9·86126	10·13874	9·90796	0′
					54 Deg.
36 Deg.					
0′	9·76922	9·86126	10·13874	9·90796	60
5	9·77009	9·86259	10·13741	9·90750	55
10	9·77095	9·86392	10·13608	9·90704	50
15	9·77181	9·86524	10·13476	9·90657	45
20	9·77268	9·86656	10·13344	9·90611	40
25	9·77353	9·86789	10·13211	9·90565	35
30	9·77439	9·86921	10·13079	9·90518	30
35	9·77524	9·87053	10·12947	9·90471	25
40	9·77609	9·87185	10·12815	9·90424	20
45	9·77694	9·87317	10·12683	9·90377	15
50	9·77778	9·87448	10·12552	9·90330	10
55	9·77862	9·87580	10·12420	9·90282	5
60	9·77946	9·87711	10·12289	9·90235	0′
					53 Deg.
	Cosine.	Cotang.	Tang.	Sine.	

LOGARITHMIC SINES, TANGENTS, &C. (*continued*).

	Sine.	Tang.	Cotang.	Cosine.	
37 Deg.					
0′	9·77916	9·87711	10·12289	9·90235	60
5	9·78030	9·87843	10·12157	9·90187	55
10	9·78113	9·87974	10·12026	9·90139	50
15	9·78197	9·88105	10·11895	9·90091	45
20	9·78280	9·88236	10·11764	9·90043	40
25	9·78362	9·88367	10·11633	9·89995	35
30	9·78445	9·88498	10·11502	9 89947	30
35	9·78527	9·88629	10·11371	9·89898	25
40	9·78609	9·88759	10·11241	9·89849	20
45	9·78691	9·88890	10·11110	9·89801	15
50	9·78772	9·89020	10·10980	9·89752	10
55	9·78853	9·89151	10·10849	9·89702	5
60	9·78934	9·89281	10.10719	9·89653	0′
					52 Deg.
38 Deg.					
0′	9·78934	9·89281	10·10719	9·89653	60
5	9·79015	9·89411	10·10589	9·89604	55
10	9·79095	9·89541	10·10459	9·89554	50
15	9·79176	9·89671	10·10329	9·89501	45
20	9·79256	9·89801	10·10199	9·89455	40
25	9·79335	9·89931	10·10069	9·89405	35
30	9·79415	9·90061	10·09939	9·89354	30
35	9·79494	9·90190	10·09810	9·89304	25
40	9·79573	9·90320	10·09680	9·89251	20
45	9·79652	9·90449	10·09551	9·89203	15
50	9·79731	9·90578	10·09422	9·89152	10
55	9·79809	9·90708	10·09292	9·89101	5
60	9·79887	9·90837	10·09163	9·89050	0′
					51 Deg.
39 Deg.					
0′	9·79887	9·90837	10·09163	9·89050	60
5	9·79965	9·90966	10·09034	9·88999	55
10	9·80043	9·91095	10·08905	9·88948	50
15	9·80120	9·91224	10·08776	9·88896	45
20	9·80197	9·91353	10·08647	9·88844	40
25	9·80274	9·91482	10·08518	9·88793	35
30	9·80351	9·91610	10·08390	9·88741	30
35	9·80428	9·91739	10·08261	9·88688	25
40	9·80504	9·91868	10·08132	9·88636	20
45	9·80580	9·91996	10·08004	9·88584	15
50	9·80656	9·92125	10·07875	9·88531	10
55	9·80731	9·92253	10·07747	9·88478	5
60	9·80807	9·92381	10·07619	9·88425	0′
					50 Deg.
40 Deg.					
0	9·80807	9·92381	10·07619	9·88425	60
5	9·80882	9·92510	10·07490	9·88372	55
10	9·80957	9·92638	10·07362	9·88319	50
15	9·81032	9·92766	10·07234	9·88266	45
20	9·81106	9·92894	10·07106	9·88212	40
25	9·81180	9·93022	10·06978	9·88158	35
30	9·81254	9·93150	10·06850	9·88105	30
35	9·81328	9·93278	10·06722	9·88051	25
40	9 81402	9·93406	10·06594	9·87996	20
45	9·81475	9·93533	10·06467	9·87942	15
50	9·81549	9·93661	10·06339	9·87887	10
55	9·81622	9·93789	10·06211	9·87833	5
60	9·81694	9·93916	10·06084	9·87778	0′
					49 Deg.
	Cosine.	Cotang.	Tang.	Sine.	

	Sine.	Tang.	Cotang.	Cosine.	
41 Deg.					
0′	9·81694	9·93916	10·06084	9·27778	60
5	9·81767	9·94044	10·05956	9·87723	55
10	9·81839	9·94171	10·05829	9·87668	50
15	9·81911	9·94299	10·05701	9·87613	45
20	9·81983	9·94426	10·05574	9·87557	40
25	9·82055	9·94554	10·05446	9·87501	35
30	9·82126	9·94681	10·05319	9·87446	30
35	9·82198	9·94808	10·05192	9·87390	25
40	9·82269	9·94935	10·05065	9·87334	20
45	9·82340	9·95062	10·04938	9·87277	15
50	9·82410	9·95190	10·04810	9·87221	10
55	9·82481	9·95317	10·04683	9·87164	5
60	9·82551	9·95444	10·04556	9·87107	0′
					48 Deg.
42 Deg.					
0′	9·82551	9·95444	10·04556	9·87107	60
5	9·82621	9·95571	10·04429	9·87050	55
10	9·82691	9·95698	10·04302	9·86993	50
15	9·82761	9·95825	10·04175	9·86936	45
20	9·82830	9·95952	10·04048	9·86879	40
25	9·82899	9·96078	10·03922	9·86821	35
30	9·82968	9·96205	10·03795	9·86763	30
35	9·83037	9·96332	10·03668	9·86705	25
40	9·83106	9·96459	10·03541	9·86647	20
45	9·83174	9·96586	10·03414	9·86589	15
50	9·83242	9·96712	10·03288	9·86530	10
55	9·83310	9·96839	10·03161	9·86472	5
60	9·83378	9·96966	10·03034	9·86413	0′
					47 Deg.
43 Deg.					
0′	9·83378	9·96966	10·03034	9·86413	60
5	9·83446	9·97092	10·02908	9·86354	55
10	9·83513	9·97219	10·02781	9·86295	50
15	9·83581	9·97345	10·02655	9·86235	45
20	9 83648	9·97472	10·02528	9·86176	40
25	9·83715	9·97598	10·02402	9·86116	35
30	9·83781	9·97725	10·02275	9·86056	30
35	9·83848	9·97851	10·02149	9·85996	25
40	9·83914	9·97978	10·02022	9·85936	20
45	9·83980	9·98104	10·01896	9·85876	15
50	9·84046	9·98231	10·01769	9·85815	10
55	9·84112	9·98357	10·01643	9·85754	5
60	9·84177	9·98484	10·01516	9·85692	0′
					46 Deg.
44 Deg.					
0′	9·84177	9·98484	10·01516	9·85693	60
5	9·84242	9·98610	10·01390	9·85632	55
10	9·84308	9·98737	10 01263	9·85571	50
15	9·84373	9·98863	10·01137	9·85510	45
20	9·84437	9·98989	10·01011	9·85448	40
25	9·84502	9·99116	10·00884	9·85386	35
30	9·84566	9·99242	10·00758	9·85324	30
35	9·84630	9·99368	10·00632	9·85262	25
40	9·84694	9·99495	10·00505	9·85200	20
45	9·84758	9·99621	10·00379	9·85137	15
50	9·84822	9·99747	10·00253	9·85074	10
55	9·84885	9·99874	10·00126	9·85012	5
60	9·84949	10·00000	10·00000	9·84949	0′
					45 Deg.
	Cosine.	Cotang.	Tang.	Sine.	

SQUARES—CUBES—ROOTS—RECIPROCALS.

SQUARES—CUBES—ROOTS—RECIPROCALS.

TABLE OF SQUARES, CUBES, SQUARE AND CUBE ROOTS, AND RECIPROCALS, OF NUMBERS FROM 1 TO 1,000.

Number.	Squares.	Cubes.	$\sqrt{\text{Roots.}}$	$\sqrt[3]{\text{Roots.}}$	Reciprocals.	Number.	Squares.	Cubes.	$\sqrt{\text{Roots.}}$	$\sqrt[3]{\text{Roots.}}$	Reciprocals.
1	1	1	1·0000000	1·0000000	1·000000000	81	6561	531441	9·0000000	4·3267487	·012345679
2	4	8	1·4142136	1·2599210	·500000000	82	6724	551368	9·0553851	4·3444815	·012195122
3	9	27	1·7320508	1·4422496	·333333333	83	6889	571787	9·1104336	4·3620707	·012048193
4	16	64	2·0000000	1·5874011	·250000000	84	7056	592704	9·1651514	4·3795191	·011904762
5	25	125	2·2360680	1·7099759	·200000000	85	7225	614125	9·2195445	4·3968296	·011764706
6	36	216	2·4494897	1·8171206	·166666666	86	7396	636056	9·2736185	4·4140049	·011627907
7	49	343	2·6457513	1·9129312	·142857143	87	7569	658503	9·3273791	4·4310476	·011494253
8	64	512	2·8284271	2·0000000	·125000000	88	7744	681472	9·3808315	4·4479692	·011363636
9	81	729	3·0000000	2·0800837	·111111111	89	7921	704969	9·4339811	4·4647451	·011235955
10	100	1000	3·1622777	2·1544347	·100000000	90	8100	729000	9·4868330	4·4814047	·011111111
11	121	1331	3·3166248	2·2239801	·090909091	91	8281	753571	9·5393920	4·4979414	·010989011
12	144	1728	3·4641016	2·2894286	·083333333	92	8464	778688	9·5916630	4·5143574	·010869565
13	169	2197	3·6055513	2·3513347	·076923077	93	8649	804357	9·6436508	4·5306549	·010752688
14	196	2744	3·7416574	2·4101422	·071428571	94	8836	830584	9·6953597	4·5468359	·010638298
15	225	3375	3·8729833	2·4662121	·066666667	95	9025	857375	9·7467943	4·5629026	·010526316
16	256	4096	4·0000000	2·5198421	·062500000	96	9216	884736	9·7979590	4·5788570	·010416667
17	289	4913	4·1231056	2·5712816	·058823529	97	9409	912673	9·8488578	4·5947009	·010309278
18	324	5832	4·2426407	2·6207414	·055555556	98	9604	941192	9·8994949	4·6104363	·010204082
19	361	6859	4·3588989	2·6684016	·052631579	99	9801	970299	9·9498744	4·6260650	·010101010
20	400	8000	4·4721360	2·7144177	·050000000	100	10000	1000000	10·0000000	4·6415888	·010000000
21	441	9261	4·5825757	2·7589243	·047619048	101	10201	1030301	10·0498756	4·6570095	·009900990
22	484	10648	4·6904158	2·8020393	·045454545	102	10404	1061208	10·0995049	4·6723287	·009803922
23	529	12167	4·7958315	2·8438670	·043478261	103	10609	1092727	10·1488916	4·6875482	·009708738
24	576	13824	4·8989795	2·8844991	·041666667	104	10816	1124864	10·1980390	4·7026694	·009615385
25	625	15625	5·0000000	2·9240177	·040000000	105	11025	1157625	10·2469508	4·7176940	·009523810
26	676	17576	5·0990195	2·9624960	·038461538	106	11236	1191016	10·2956301	4·7326235	·009433962
27	729	19683	5·1961524	3·0000000	·037037037	107	11449	1225043	10·3440804	4·7474594	·009345794
28	784	21952	5·2915026	3·0365889	·035714286	108	11664	1259712	10·3923048	4·7622032	·009259259
29	841	24389	5·3851648	3·0723168	·034482759	109	11881	1295029	10·4403065	4·7768562	·009174312
30	900	27000	5·4772256	3·1072325	·033333333	110	12100	1331000	10·4880885	4·7914199	·009090909
31	961	29791	5·5677644	3·1413806	·032258065	111	12321	1367631	10·5356538	4·8058995	·009009009
32	1024	32768	5·6568542	3·1748021	·031250000	112	12544	1404928	10·5830052	4·8202845	·008928571
33	1089	35937	5·7445626	3·2075343	·030303030	113	12769	1442897	10·6301458	4·8345881	·008849558
34	1156	39304	5·8309519	3·2396118	·029411765	114	12996	1481544	10·6770783	4·8488076	·008771930
35	1225	42875	5·9160798	3·2710663	·028571429	115	13225	1520875	10·7238053	4·8629442	·008695652
36	1296	46656	6·0000000	3·3019272	·027777778	116	13456	1560896	10·7703296	4·8769990	·008620690

37	1369	50653	6·0827625	3·3322218	·027027027	117	13689	1601613	10·8166538	4·8909732	·008547009
38	1444	54872	6·1644140	3·3619754	·026315789	118	13924	1643032	10·8627805	4·9048681	·008474576
39	1521	59319	6·2449980	3·3912114	·025641026	119	14161	1685159	10·9087121	4·9186847	·008403361
40	1600	64000	6·3245553	3·4199519	·025000000	120	14400	1728000	10·9544512	4·9324242	·008333333
41	1681	68921	6·4031242	3·4482172	·024390244	121	14641	1771561	11·0000000	4·9460874	·008264463
42	1764	74088	6·4807407	3·4760266	·023809524	122	14884	1815848	11·0453610	4·9596757	·008196721
43	1849	79507	6·5574385	3·5033981	·023255814	123	15129	1860867	11·0905365	4·9731898	·008130081
44	1936	85184	6·6332496	3·5303483	·022727273	124	15376	1906624	11·1355287	4·9866310	·008064516
45	2025	91125	6·7082039	3·5568933	·022222222	125	15625	1953125	11·1803399	5·0000000	·008000000
46	2116	97336	6·7823300	3·5830479	·021739130	126	15876	2000376	11·2249722	5·0132979	·007936508
47	2209	103823	6·8556546	3·6088261	·021276600	127	16129	2048383	11·2694277	5·0265257	·007874016
48	2304	110592	6·9282032	3·6342411	·020833333	128	16384	2097152	11·3137085	5·0396842	·007812500
49	2401	117649	7·0000000	3·6593057	·020408163	129	16641	2146689	11·3578167	5·0527743	·007751938
50	2500	125000	7·0710678	3·6840314	·020000000	130	16900	2197000	11·4017543	5·0657970	·007692308
51	2601	132651	7·1414284	3·7084298	·019607843	131	17161	2248091	11·4455231	5·0787531	·007633588
52	2704	140608	7·2111026	3·7325111	·019230769	132	17424	2299968	11·4891253	5·0916434	·007575758
53	2809	148877	7·2801099	3·7562858	·018867925	133	17689	2352637	11·5325626	5·1044687	·007518797
54	2916	157464	7·3484692	3·7797631	·018518519	134	17956	2406104	11·5758369	5·1172299	·007462687
55	3025	166375	7·4161985	3·8029525	·018181818	135	18225	2460375	11·6189500	5·1299278	·007407407
56	3136	175616	7·4833148	3·8258624	·017857143	136	18496	2515456	11·6619038	5·1425632	·007352941
57	3249	185193	7·5498344	3·8485011	·017543860	137	18769	2571353	11·7046999	5·1551367	·007299270
58	3364	195112	7·6157731	3·8708766	·017241379	138	19044	2628072	11·7473401	5·1676493	·007246377
59	3481	205379	7·6811457	3·8929965	·016949153	139	19321	2685619	11·7898261	5·1801015	·007194245
60	3600	216000	7·7459667	3·9148676	·016666667	140	19600	2744000	11·8321596	5·1924941	·007142857
61	3721	226981	7·8102497	3·9364972	·016393443	141	19881	2803221	11·8743421	5·2048279	·007092199
62	3844	238328	7·8740079	3·9578915	·016129032	142	20164	2863288	11·9163753	5·2171034	·007042254
63	3969	250047	7·9372539	3·9790571	·015873016	143	20449	2924207	11·9582607	5·2293215	·006993007
64	4096	262144	8·0000000	4·0000000	·015625000	144	20736	2985984	12·0000000	5·2414828	·006944444
65	4225	274625	8·0622577	4·0207256	·015384615	145	21025	3048625	12·0415946	5·2535879	·006896552
66	4356	287496	8·1240384	4·0412401	·015151515	146	21316	3112136	12·0830460	5·2656374	·006849315
67	4489	300763	8·1853528	4·0615480	·014925373	147	21609	3176523	12·1243557	5·2776321	·006802721
68	4624	314432	8·2462113	4·0816551	·014705882	148	21904	3241792	12·1655251	5·2895725	·006756757
69	4761	328509	8·3066239	4·1015661	·014492754	149	22201	3307949	12·2065556	5·3014592	·006711409
70	4900	343000	8·3666003	4·1212853	·014285714	150	22500	3375000	12·2474487	5·3132928	·006666667
71	5041	357911	8·4261498	4·1408178	·014084517	151	22801	3442951	12·2882057	5·3250740	·006622517
72	5184	373248	8·4852814	4·1601676	·013888889	152	23104	3511808	12·3288280	5·3368033	·006578947
73	5329	389017	8·5440037	4·1793390	·013698630	153	23409	3581577	12·3693169	5·3484812	·006535948
74	5476	405224	8·6023253	4·1983364	·013513514	154	23716	3652264	12·4096736	5·3601084	·006493506
75	5625	421875	8·6602540	4·2171633	·013333333	155	24025	3723875	12·4498996	5·3716854	·006451613
76	5776	438976	8·7177979	4·2358236	·013157895	156	24336	3796416	12·4899960	5·3832126	·006410256
77	5929	456533	8·7749644	4·2543210	·012987013	157	24649	3869893	12·5299641	5·3946907	·006369427
78	6084	474552	8·8317609	4·2726586	·012820513	158	24964	3944312	12·5698051	5·4061202	·006329114
79	6241	493039	8·8881944	4·2908404	·012658228	159	25281	4019679	12·6095202	5·4175015	·006289308
80	6400	512000	8·9442719	4·3088695	·012500000	160	25600	4096000	12·6491106	5·4288352	·006250000

TABLE OF SQUARES, CUBES, SQUARE AND CUBE ROOTS, AND RECIPROCALS (*continued*).

Number.	Squares.	Cubes.	$\sqrt{\text{Roots}}$.	$\sqrt[3]{\text{Roots}}$.	Reciprocals.	Number.	Squares.	Cubes.	$\sqrt{\text{Roots}}$.	$\sqrt[3]{\text{Roots}}$.	Reciprocals.
161	25921	4173281	12·6885775	5·4401218	·006211180	245	60025	14706125	15·6524758	6·2573248	·004081633
162	26244	4251528	12·7279221	5·4513618	·006172840	246	60516	14886936	15·6843871	6·2658266	·004065041
163	26569	4330747	12·7671453	5·4625556	·006134969	247	61009	15069223	15·7162336	6·2743054	·004048583
164	26896	4410944	12·8062485	5·4737037	·006097561	248	61504	15252992	15·7480157	6·2827613	·004032258
165	27225	4492125	12·8452326	5·4848066	·006060606	249	62001	15438249	15·7797338	6·2911946	·004016064
166	27556	4574296	12·8840987	5·4958647	·006024096	250	62500	15625000	15·8113883	6·2996053	·004000000
167	27889	4657463	12·9228480	5·5068784	·005988024	251	63001	15813251	15·8429795	6·3079935	·003984064
168	28224	4741632	12·9614814	5·5178484	·005952381	252	63504	16003008	15·8745079	6·3163596	·003968254
169	28561	4826809	13·0000000	5·5287748	·005917160	253	64009	16194277	15·9059737	6·3247035	·003952569
170	28900	4913000	13·0384048	5·5396583	·005882353	254	64516	16387064	15·9373775	6·3330256	·003937008
171	29241	5000211	13·0766968	5·5504991	·005847953	255	65025	16581375	15·9687194	6·3413257	·003921569
172	29584	5088448	13·1148770	5·5612978	·005813953	256	65536	16777216	16·0000000	6·3496042	·003906250
173	29929	5177717	13·1529464	5·5720546	·005780347	257	66049	16974593	16·0312195	6·3578611	·003891051
174	30276	5268024	13·1909060	5·5827702	·005747126	258	66564	17173512	16·0623784	6·3660968	·003875969
175	30625	5359375	13·2287566	5·5934447	·005714286	259	67081	17373979	16·0934769	6·3743111	·003861004
176	30976	5451776	13·2664992	5·6040787	·005681818	260	67600	17576000	16·1245155	6·3825043	·003846154
177	31329	5545233	13·3041347	5·6146724	·005649718	261	68121	17779581	16·1554944	6·3906765	·003831418
178	31684	5639752	13·3416641	5·6252263	·005617978	262	68644	17984728	16·1864141	6·3988279	·003816794
179	32041	5735339	13·3790882	5·6357408	·005586592	263	69169	18191447	16·2172747	6·4069585	·003802281
180	32400	5832000	13·4164079	5·6462162	·005555556	264	69696	18399744	16·2480768	6·4150687	·003787879
181	32761	5929741	13·4536240	5·6566528	·005524862	265	70225	18609625	16·2788206	6·4231583	·003773585
182	33124	6028568	13·4907376	5·6670511	·005494505	266	70756	18821096	16·3095064	6·4312276	·003759398
183	33489	6128487	13·5277493	5·6774114	·005464481	267	71289	19034163	16·3401346	6·4392767	·003745318
184	33856	6229504	13·5646600	5·6877340	·005434783	268	71824	19248832	16·3707055	6·4473057	·003731343
185	34225	6331625	13·6014705	5·6980192	·005405405	269	72361	19465109	16·4012195	6·4553148	·003717472
186	34596	6434856	13·6381817	5·7082675	·005376344	270	72900	19683000	16·4316767	6·4633041	·003703704
187	34969	6539203	13·6747943	5·7184791	·005347594	271	73441	19902511	16·4620776	6·4712736	·003690037
188	35344	6644672	13·7113092	5·7286543	·005319149	272	73984	20123648	16·4924225	6·4792236	·003676471
189	35721	6751269	13·7477271	5·7387936	·005291005	273	74529	20346417	16·5227116	6·4871541	·003663004
190	36100	6859000	13·7840488	5·7488971	·005263158	274	75076	20570824	16·5529454	6·4950653	·003649635
191	36481	6967871	13·8202750	5·7589652	·005235602	275	75625	20796875	16·5831240	6·5029572	·003636364
192	36864	7077888	13·8564065	5·7689982	·005208333	276	76176	21024576	16·6132477	6·5108300	·003623188
193	37249	7189057	13·8924440	5·7789966	·005181347	277	76729	21253933	16·6433170	6·5186839	·003610108
194	37636	7301384	13·9283883	5·7889604	·005154639	278	77284	21484952	16·6733320	6·5265189	·003597122
195	38025	7414875	13·9642400	5·7988900	·005128205	279	77841	21717639	16·7032931	6·5343351	·003584229
196	38416	7529536	14·0000000	5·8087857	·005102041	280	78400	21952000	16·7332005	6·5421326	·003571429
197	38809	7645373	14·0356688	5·8186479	·005076142	281	78961	22188041	16·7630546	6·5499116	·003558719
198	39204	7762392	14·0712473	5·8284867	·005050505	282	79524	22425768	16·7928556	6·5576722	·003546099
199	39601	7880599	14·1067360	5·8382725	·005025126	283	80089	22665187	16·8226038	6·5654144	·003533569
200	40000	8000000	14·1421356	5·8480355	·005000000	284	80656	22906304	16·8522995	6·5731385	·003521127

201	40401	8120601	14·1774469	5·8577660	·004975124	285	81225	23149125	16·8819430	6·5808443	·003508772
202	40804	8242408	14·2126704	5·8674673	·004950495	286	81796	23393656	16·9115345	6·5885323	·003496503
203	41209	8365427	14·2478068	5·8771307	·004926108	287	82369	23639903	16·9410743	6·5962023	·003484321
204	41616	8489664	14·2828569	5·8867653	·004901961	288	82944	23887872	16·9705627	6·6038545	·003472222
205	42025	8615125	14·3178211	5·8963685	·004878049	289	83521	24137569	17·0000000	6·6114890	·003460208
206	42436	8741816	14·3527001	5·9059406	·004854369	290	84100	24389000	17·0293864	6·6191060	·003448276
207	42849	8869743	14·3874946	5·9154817	·004830918	291	84681	24642171	17·0587221	6·6267054	·003436426
208	43264	8998912	14·4222051	5·9249921	·004807692	292	85264	24897088	17·0880075	6·6342874	·003424658
209	43681	9129329	14·4568323	5·9344721	·004784689	293	85849	25153757	17·1172428	6·6418522	·003412969
210	44100	9261000	14·4913767	5·9439220	·004761905	294	86436	25412184	17·1464282	6·6493998	·003401361
211	44521	9393931	14·5258390	5·9533418	·004739336	295	87025	25672375	17·1755640	6·6569302	·003389831
212	44944	9528128	14·5602198	5·9627320	·004716981	296	87616	25934336	17·2046505	6·6644437	·003378378
213	45369	9663597	14·5945195	5·9720926	·004694836	297	88209	26198073	17·2336879	6·6719403	·003367003
214	45796	9800344	14·6287388	5·9814240	·004672897	298	88804	26463592	17·2626765	6·6794200	·003355705
215	46225	9938375	14·6628783	5·9907264	·004651163	299	89401	26730899	17·2916165	6·6868831	·003344482
216	46656	10077696	14·6969385	6·0000000	·004629630	300	90000	27000000	17·3205081	6·6943295	·003333333
217	47089	10218313	14·7309199	6·0092450	·004608295	301	90601	27270901	17·3493516	6·7017593	·003322259
218	47524	10360232	14·7648231	6·0184617	·004587156	302	91204	27543608	17·3781472	6·7091729	·003311258
219	47961	10503459	14·7986486	6·0276502	·004566210	303	91809	27818127	17·4068952	6·7165700	·003301330
220	48400	10648000	14·8323970	6·0368107	·004545455	304	92416	28094464	17·4355958	6·7239508	·003289474
221	48841	10793861	14·8660687	6·0459435	·004524887	305	93025	28372625	17·4642492	6·7313155	·003278689
222	49284	10941048	14·8996644	6·0550489	·004504505	306	93636	28652616	17·4928557	6·7386641	·003267974
223	49729	11089567	14·9331845	6·0641270	·004484305	307	94249	28934443	17·5214155	6·7459967	·003257329
224	50176	11239424	14·9666295	6·0731779	·004464286	308	94864	29218112	17·5499288	6·7533134	·003246753
225	50625	11390625	15·0000000	6·0822020	·004444444	309	95481	29503629	17·5783958	6·7606143	·003236246
226	51076	11543176	15·0332964	6·0911994	·004424779	310	96100	29791000	17·6068169	6·7678995	·003225806
227	51529	11697083	15·0665192	6·1001702	·004405286	311	96721	30080231	17·6351921	6·7751690	·003215434
228	51984	11852352	15·0996689	6·1091147	·004385965	312	97344	30371328	17·6635217	6·7824229	·003205128
229	52441	12008989	15·1327460	6·1180332	·004366812	313	97969	30664297	17·6918060	6·7896613	·003194888
230	52900	12167000	15·1657509	6·1269257	·004347826	314	98596	30959144	17·7200451	6·7968844	·003184713
231	53361	12326391	15·1986842	6·1357924	·004329004	315	99225	31255875	17·7482393	6·8040921	·003174603
232	53824	12487168	15·2315462	6·1446337	·004310345	316	99856	31554496	17·7763888	6·8112847	·003164557
233	54289	12649337	15·2643375	6·1534495	·004291845	317	100489	31855013	17·8044938	6·8184620	·003154574
234	54756	12812904	15·2970585	6·1622401	·004273504	318	101124	32157432	17·8325545	6·8256242	·003144654
235	55225	12977875	15·3297097	6·1710058	·004255319	319	101761	32461759	17·8605711	6·8327714	·003134796
236	55696	13144256	15·3622915	6·1797466	·004237288	320	102400	32768000	17·8885438	6·8399037	·003125000
237	56169	13312053	15·3948043	6·1884628	·004219409	321	103041	33076161	17·9164729	6·8470213	·003115265
238	56644	13481272	15·4272486	6·1971544	·004201681	322	103684	33386248	17·9443584	6·8541240	·003105590
239	57121	13651919	15·4596248	6·2058218	·004184100	323	104329	33698267	17·9722008	6·8612120	·003095975
240	57600	13824000	15·4919334	6·2144650	·004166667	324	104976	34012224	18·0000000	6·8682855	·003086420
241	58081	13997521	15·5241747	6·2230843	·004149378	325	105625	34328125	18·0277564	6·8753433	·003076923
242	58564	14172488	15·5563492	6·2316797	·004132231	326	106276	34645976	18·0554701	6·8823888	·003067485
243	59049	14348907	15·5884573	6·2402515	·004115226	327	106929	34965783	18·0831413	6·8894188	·003058104
244	59536	14526784	15·6204994	6·2487998	·004098361	328	107584	35287552	18·1107703	6·8964345	·003048780

TABLE OF SQUARES, CUBES, SQUARE AND CUBE ROOTS, AND RECIPROCALS (*continued*).

Number.	Squares.	Cubes.	√ Roots.	∛ Roots.	Reciprocals.	Number.	Squares.	Cubes.	√ Roots.	∛ Roots.	Reciprocals.
329	108241	35611289	18·1383571	6·9034359	·003039514	413	170569	70444997	20·3224014	7·4470343	·002421308
330	108900	35937000	18·1659021	6·9104232	·003030303	414	171396	70957944	20·3469899	7·4530399	·002415469
331	109561	36264691	18·1934054	6·9173964	·003021148	415	172225	71473375	20·3715488	7·4590359	·002409639
332	110224	36594368	18·2208672	6·9243556	·003012048	416	173056	71991296	20·3960781	7·4650223	·002406846
333	110889	36926037	18·2482876	6·9312008	·003003003	417	173889	72511713	20·4205779	7·4709991	·002398082
334	111556	37259704	18·2756669	6·9382321	·002994012	418	174724	73034632	20·4450483	7·4769664	·002392344
335	112225	37595375	18·3030052	6·9451496	·002985075	419	175561	73560059	20·4694895	7·4829242	·002386635
336	112896	37933056	18·3303028	6·9520533	·002976190	420	176400	74088000	20·4939015	7·4888724	·002380952
337	113569	38272753	18·3575598	6·9589434	·002967359	421	177241	74618461	20·5182845	7·4948113	·002375297
338	114244	38614472	18·3847763	6·9658198	·002958580	422	178084	75151448	20·5426386	7·5007406	·002369668
339	114921	38958219	18·4119526	6·9726826	·002949853	423	178929	75686967	20·5669638	7·5066607	·002364066
340	115600	39304000	18·4390889	6·9795321	·002941176	424	179776	76225024	20·5912603	7·5125715	·002358491
341	116281	39651821	18·4661853	6·9863681	·002932551	425	180625	76765625	20·6155281	7·5184730	·002352941
342	116964	40001688	18·4932420	6·9931906	·002923977	426	181476	77308776	20·6397674	7·5243652	·002347418
343	117649	40353607	18·5202592	7·0000000	·002915452	427	182329	77854483	20·6639783	7·5302482	·002341920
344	118336	40707584	18·5472370	7·0067962	·002906977	428	183184	78402752	20·6881609	7·5361221	·002336449
345	119025	41063625	18·5741756	7·0135791	·002898551	429	184041	78953589	20·7123152	7·5419867	·002331002
346	119716	41421736	18·6010752	7·0203490	·002890173	430	184900	79507000	20·7364414	7·5478423	·002325581
347	120409	41781923	18·6279360	7·0271058	·002881844	431	185761	80062991	20·7605395	7·5536888	·002320186
348	121104	42144192	18·6547581	7·0338497	·002873563	432	186624	80621568	20·7846097	7·5595263	·002314815
349	121801	42508549	18·6815417	7·0405806	·002865330	433	187489	81182737	20·8086520	7·5653548	·002309469
350	122500	42875000	18·7082869	7·0472987	·002857143	434	188356	81746504	20·8326667	7·5711743	·002304147
351	123201	43243551	18·7349940	7·0540041	·002849003	435	189225	82312875	20·8566536	7·5769849	·002298851
352	123904	43614208	18·7616630	7·0606967	·002840909	436	190096	82881856	20·8806130	7·5827865	·002293578
353	124609	43986977	18·7882942	7·0673767	·002832861	437	190969	83453453	20·9045450	7·5885793	·002288330
354	125316	44361864	18·8148877	7·0740440	·002824859	438	191844	84027672	20·9284495	7·5943633	·002283105
355	126025	44738875	18·8414437	7·0806988	·002816901	439	192721	84604519	20·9523268	7·6001385	·002277904
356	126736	45118016	18·8679623	7·0873411	·002808989	440	193600	85184000	20·9761770	7·6059049	·002272727
357	127449	45499293	18·8944436	7·0939709	·002801120	441	194481	85766121	21·0000000	7·6116626	·002267574
358	128164	45882712	18·9208879	7·1005885	·002793296	442	195364	86350888	21·0237960	7·6174116	·002262443
359	128881	46268279	18·9472953	7·1071937	·002785515	443	196249	86938307	21·0475652	7·6231519	·002257336
360	129600	46656000	18·9736660	7·1137866	·002777778	444	197136	87528384	21·0713075	7·6288837	·002252252
361	130321	47045881	19·0000000	7·1203674	·002770082	445	198025	88121125	21·0950231	7·6346067	·002247191
362	131044	47437928	19·0262976	7·1269360	·002762431	446	198916	88716536	21·1187121	7·6403213	·002242152
363	131769	47832147	19·0525589	7·1334925	·002754821	447	199809	89314623	21·1423745	7·6460272	·002237136
364	132496	48228544	19·0787840	7·1400370	·002747253	448	200704	89915392	21·1660105	7·6517247	·002232143
365	133225	48627125	19·1049732	7·1465695	·002739726	449	201601	90518849	21·1896201	7·6574138	·002227171
366	133956	49027896	19·1311265	7·1530901	·002732240	450	202500	91125000	21·2132034	7·6630943	·002222222
367	134689	49430863	19·1572441	7·1595988	·002724796	451	203401	91733851	21·2367606	7·6687665	·002217295
368	135424	49836032	19·1833261	7·1660957	·002717391	452	204304	92345408	21·2602916	7·6744303	·002212389

369	136161	50243409	19·2093727	7·1725809	·002710027	453	205209	92959677	21·2837967	7·6800857	·002207506
370	136900	50653000	19·2353841	7·1790544	·002702703	454	206116	93576664	21·3072758	7·6857328	·002202643
371	137641	51064811	19·2613603	7·1855162	·002695418	455	207025	94196375	21·3307290	7·6913717	·002197802
372	138384	51478848	19·2873015	7·1919663	·002688172	456	207936	94818816	21·3541565	7·6970023	·002192982
373	139129	51895117	19·3132079	7·1984050	·002680965	457	208849	95443993	21·3775583	7·7026246	·002188184
374	139876	52313624	19·3390796	7·2048322	·002673797	458	209764	96071912	21·4009346	7·7082388	·002183406
375	140625	52734375	19·3649167	7·2112479	·002666667	459	210681	96702579	21·4242853	7·7138448	·002178649
376	141376	53157376	19·3907194	7·2176522	·002659574	460	211600	97336000	21·4476106	7·7194426	·002173913
377	142129	53582633	19·4164878	7·2240450	·002652520	461	212521	97972181	21·4709106	7·7250325	·002169197
378	142884	54010152	19·4422221	7·2304268	·002645503	462	213444	98611128	21·4941853	7·7306141	·002164502
379	143641	54439939	19·4679223	7·2367972	·002638521	463	214369	99252847	21·5174348	7·7361877	·002159827
380	144400	54872000	19·4935887	7·2431565	·002631579	464	215296	99897344	21·5406592	7·7417532	·002155172
381	145161	55306341	19·5192213	7·2495045	·002624672	465	216225	100544625	21·5638587	7·7473109	·002150538
382	145924	55742968	19·5448203	7·2558415	·002617801	466	217156	101194696	21·5870331	7·7528606	·002145923
383	146689	56181887	19·5703858	7·2621675	·002610966	467	218089	101847563	21·6101828	7·7584023	·002141328
384	147456	56623104	19·5959179	7·2684824	·002604167	468	219024	102503232	21·6333077	7·7639361	·002136752
385	148225	57066625	19·6214169	7·2747864	·002597403	469	219961	103161709	21·6564078	7·7694620	·002132196
386	148996	57512456	19·6468827	7·2810794	·002590674	470	220900	103823000	21·6794834	7·7749801	·002127660
387	149769	57960603	19·6723156	7·2873617	·002583979	471	221841	104487111	21·7025344	7·7804904	·002123142
388	150544	58411072	19·6977156	7·2936330	·002577320	472	222784	105154048	21·7255610	7·7859928	·002118644
389	151321	58863869	19·7230829	7·2998936	·002570694	473	223729	105823817	21·7485632	7·7914875	·002114165
390	152100	59319000	19·7484177	7·3061436	·002564103	474	224676	106496424	21·7715411	7·7969745	·002109705
391	152881	59776471	19·7737199	7·3123828	·002557545	475	225625	107171875	21·7944947	7·8024538	·002105263
392	153664	60236288	19·7989899	7·3186114	·002551020	476	226576	107850176	21·8174242	7·8079254	·002100840
393	154449	60698457	19·8242276	7·3248295	·002544529	477	227529	108531333	21·8403297	7·8133892	·002096486
394	155236	61162984	19·8494332	7·3310369	·002538071	478	228484	109215352	21·8632111	7·8188456	·002092050
395	156025	61629875	19·8746069	7·3372339	·002531646	479	229441	109902239	21·8860686	7·8242942	·002087683
396	156816	62099136	19·8997487	7·3434205	·002525253	480	230400	110592000	21·9089023	7·8297353	·002083333
397	157609	62570773	19·9248588	7·3495966	·002518892	481	231361	111284641	21·9317122	7·8351688	·002079002
398	158404	63044792	19·9499373	7·3557624	·002512563	482	232324	111980168	21·9544984	7·8405949	·002074689
399	159201	63521199	19·9749844	7·3619178	·002506266	483	233289	112678587	21·9772610	7·8460134	·002070393
400	160000	64000000	20·0000000	7·3680630	·002500000	484	234256	113379904	22·0000000	7·8514244	·002066116
401	160801	64481201	20·0249844	7·3741979	·002493766	485	235225	114084125	22·0227155	7·8568281	·002061856
402	161604	64964808	20·0499377	7·3803227	·002487562	486	236196	114791256	22·0454077	7·8622242	·002057613
403	162409	65450827	20·0748599	7·3864373	·002481390	487	237169	115501303	22·0680765	7·8676130	·002053388
404	163216	65939264	20·0997512	7·3925418	·002475248	488	238144	116214272	22·0907220	7·8729944	·002049180
405	164025	66430125	20·1246118	7·3986363	·002469136	489	239121	116930169	22·1133444	7·8783684	·002044990
406	164836	66923416	20·1494417	7·4047206	·002463054	490	240100	117649000	22·1359436	7·8837352	·002040816
407	165649	67419143	20·1742410	7·4107950	·002457002	491	241081	118370771	22·1585198	7·8890946	·002036660
408	166464	67917312	20·1990099	7·4168595	·002450980	492	242064	119095488	22·1810730	7·8944468	·002032520
409	167281	68417929	20·2237484	7·4229142	·002444988	493	243049	119823157	22·2036033	7·8997917	·002028398
410	168100	68921000	20·2484567	7·4289589	·002439024	494	244036	120553784	22·2261108	7·9051294	·002024291
411	168921	69426531	20·2731349	7·4349938	·002433090	495	245025	121287375	22·2485955	7·9104599	·002020202
412	169744	69934528	20·2977831	7·4410189	·002427184	496	246016	122023936	22·2710575	7·9157832	·002016129

TABLE OF SQUARES, CUBES, SQUARE AND CUBE ROOTS, AND RECIPROCALS (*continued*).

Number.	Squares.	Cubes.	√Roots.	∛Roots.	Reciprocals.	Number.	Squares.	Cubes.	√Roots.	∛Roots.	Reciprocals.
497	247009	122763473	22·2934968	7·9210994	·002012072	581	337561	196122941	24·1039416	8·3443410	·001721170
498	248004	123505992	22·3159136	7·9264085	·002008032	582	338724	197137368	24·1246762	8·3491256	·001718213
499	249001	124251499	22·3383079	7·9317104	·002004008	583	339889	198155287	24·1453929	8·3539047	·001715266
500	250000	125000000	22·3606798	7·9370053	·002000000	584	341056	199176704	24·1660919	8·3586784	·001712329
501	251001	125751501	22·3830293	7·9422931	·001996008	585	342225	200201625	24·1867732	8·3634466	·001709402
502	252004	126506008	22·4053565	7·9475739	·001992032	586	343396	201230056	24·2074369	8·3682095	·001706485
503	253009	127263527	22·4276615	7·9528477	·001988072	587	344569	202262003	24·2280829	8·3729668	·001703578
504	254016	128024064	22·4499443	7·9581144	·001984127	588	345744	203297472	24·2487113	8·3777188	·001700680
505	255025	128787625	22·4722051	7·9633743	·001980198	589	346921	204336469	24·2693222	8·3824653	·001697793
506	256036	129554216	22·4944438	7·9686271	·001976285	590	348100	205379000	24·2899156	8·3872065	·001694915
507	257049	130323843	22·5166605	7·9738731	·001972387	591	349281	206425071	24·3104916	8·3919428	·001692047
508	258064	131096512	22·5388553	7·9791122	·001968504	592	350464	207474688	24·3310501	8·3966729	·001689189
509	259081	131872229	22·5610283	7·9843444	·001964637	593	351649	208527857	24·3515913	8·4013981	·001686341
510	260100	132651000	22·5831796	7·9895697	·001960784	594	352836	209584584	24·3721152	8·4061180	·001683502
511	261121	133432831	22·6053091	7·9947883	·001956947	595	354025	210644875	24·3926218	8·4108326	·001680672
512	262144	134217728	22·6274170	8·0000000	·001953125	596	355216	211708736	24·4131112	8·4155419	·001677852
513	263169	135005697	22·6495033	8·0052049	·001949318	597	356409	212776173	24·4335834	8·4202460	·001675042
514	264196	135796744	22·6715681	8·0104032	·001945525	598	357604	213847192	24·4540385	8·4249448	·001672241
515	265225	136590875	22·6936114	8·0155946	·001941748	599	358801	214921799	24·4744765	8·4296383	·001669449
516	266256	137388096	22·7156334	8·0207794	·001937984	600	360000	216000000	24·4948974	8·4343267	·001666667
517	267289	138188413	22·7376341	8·0259574	·001934236	601	361201	217081801	24·5153013	8·4390098	·001663894
518	268324	138991832	22·7596134	8·0311287	·001930502	602	362404	218167208	24·5356883	8·4436877	·001661130
519	269361	139798359	22·7815715	8·0362935	·001926782	603	363609	219256227	24·5560583	8·4483605	·001658375
520	270400	140608000	22·8035085	8·0414515	·001923077	604	364816	220348864	24·5764115	8·4530281	·001655629
521	271441	141420761	22·8254244	8·0466030	·001919386	605	366025	221445125	24·5967478	8·4576906	·001652893
522	272484	142236648	22·8473193	8·0517479	·001915709	606	367236	222545016	24·6170673	8·4623479	·001650165
523	273529	143055667	22·8691933	8·0568862	·001912046	607	368449	223648543	24·6373700	8·4670001	·001647446
524	274576	143877824	22·8910463	8·0620180	·001908397	608	369664	224755712	24·6576560	8·4716471	·001644737
525	275625	144703125	22·9128785	8·0671432	·001904762	609	370881	225866529	24·6779254	8·4762892	·001642036
526	276676	145531576	22·9346899	8·0722620	·001901141	610	372100	226981000	24·6981781	8·4809261	·001639344
527	277729	146363183	22·8564806	8·0773743	·001897533	611	373321	228099131	24·7184142	8·4855579	·001636661
528	278784	147197952	22·9782506	8·0824800	·001893939	612	374544	229220928	24·7386338	8·4901848	·001633987
529	279841	148035889	23·0000000	8·0875794	·001890359	613	375769	230346397	24·7588368	8·4948065	·001631321
530	280900	148877000	23·0217289	8·0926723	·001886792	614	376996	231475544	24·7790234	8·4994233	·001628664
531	281961	149721291	23·0434372	8·0977589	·001883239	615	378225	232608375	24·7991935	8·5040350	·001626016
532	283024	150568768	23·0651252	8·1028390	·001879699	616	379456	233744896	24·8193473	8·5086417	·001623377
533	284089	151419437	23·0867928	8·1079128	·001876173	617	380689	234885113	24·8394847	8·5132435	·001620746
534	285156	152273304	23·1084400	8·1129803	·001872659	618	381924	236029032	24·8596058	8·5178403	·001618123
535	286225	153130375	23·1300670	8·1180414	·001869159	619	383161	237176659	24·8797106	8·5224331	·001615509
536	287296	153990656	23·1516738	8·1230962	·001865672	620	384400	238328000	24·8997992	8·5270189	·001612903

537	288369	154854153	23·1732605	8·1281447	·001862197	621	385641	239483061	24·9198716	8·5316009	·001610306
538	289444	155720872	23·1948270	8·1331870	·001858736	622	386884	240641848	24·9399278	8·5361780	·001607717
539	290521	156590819	23·2163735	8·1382230	·001855288	623	388129	241804367	24·9599679	8·5407501	·001605136
540	291600	157464000	23·2379001	8·1432529	·001851852	624	389376	242970624	24·9799920	8·5453173	·001602564
541	292681	158340421	23·2594067	8·1482765	·001848429	625	390625	244140625	25·0000000	8·5498797	·001600000
542	293764	159220088	23·2808935	8·1532939	·001845018	626	391876	245314376	25·0199920	8·5544372	·001597444
543	294849	160103007	23·3023604	8·1583051	·001841621	627	393129	246491883	25·0399681	8·5589899	·001594896
544	295936	160989184	23·3238076	8·1633102	·001838235	628	394384	247673152	25·0599282	8·5635377	·001592357
545	297025	161878625	23·3452351	8·1683092	·001834862	629	395641	248858189	25·0798724	8·5680807	·001589825
546	298116	162771336	23·3666429	8·1733020	·001831502	630	396900	250047000	25·0998008	8·5726189	·001587302
547	299209	163667323	23·3880311	8·1782888	·001828154	631	398161	251239591	25·1197134	8·5771523	·001584786
548	300304	164566592	23·4093998	8·1832695	·001824818	632	399424	252435968	25·1396102	8·5816809	·001582278
549	301401	165469149	23·4307490	8·1882441	·001821494	633	400689	253636137	25·1594913	8·5862047	·001579779
550	302500	166375000	23·4520788	8·1932127	·001818182	634	401956	254840104	25·1793566	8·5907238	·001577287
551	303601	167284151	23·4733892	8·1981753	·001814882	635	403225	256047875	25·1992063	8·5952380	·001574803
552	304704	168196608	23·4946802	8·2031319	·001811594	636	404496	257259456	25·2190404	8·5997476	·001572327
553	305809	169112377	23·5159520	8·2080825	·001808318	637	405769	258474853	25·2388589	8·6042525	·001569859
554	306916	170031464	23·5372046	8·2130271	·001805054	638	407044	259694072	25·2586619	8·6087526	·001567398
555	308025	170953875	23·5584380	8·2179657	·001801802	639	408321	260917119	25·2784493	8·6132480	·001564945
556	309136	171879616	23·5796522	8·2228985	·001798561	640	409600	262144000	25·2982213	8·6177388	·001562500
557	310249	172808693	23·6008474	8·2278254	·001795332	641	410881	263374721	25·3179778	8·6222248	·001560062
558	311364	173741112	23·6220236	8·2327463	·001792115	642	412164	264609288	25·3377189	8·6267063	·001557632
559	312481	174676879	23·6431808	8·2376614	·001788909	643	413449	265847707	25·3574447	8·6311830	·001555210
560	313600	175616000	23·6643191	8·2425706	·001785714	644	414736	267089984	25·3771551	8·6356551	·001552795
561	314721	176558481	23·6854386	8·2474740	·001782531	645	416025	268336125	25·3968502	8·6401226	·001550388
562	315844	177504328	23·7065392	8·2523715	·001779359	646	417316	269586136	25·4165302	8·6445855	·001547988
563	316969	178453547	23·7276210	8·2572635	·001776199	647	418609	270840023	25·4361947	8·6490437	·001545595
564	318096	179406144	23·7486842	8·2621492	·001773050	648	419904	272097792	25·4558441	8·6534974	·001543210
565	319225	180362125	23·7697286	8·2670294	·001769912	649	421201	273359449	25·4754784	8·6579465	·001540832
566	320356	181321496	23·7907545	8·2719039	·001766784	650	422500	274625000	25·4950976	8·6623911	·001538462
567	321489	182284263	23·8117618	8·2767726	·001763668	651	423801	275894451	25·5147016	8·6668310	·001536098
568	322624	183250432	23·8327506	8·2816355	·001760563	652	425104	277167808	25·5342907	8·6712665	·001533742
569	323761	184220009	23·8537209	8·2864928	·001757469	653	426409	278445077	25·5538647	8·6756974	·001531394
570	324900	185193000	23·8746728	8·2913444	·001754386	654	427716	279726264	25·5734237	8·6801237	·001529052
571	326041	186169411	23·8956063	8·2961903	·001751313	655	429025	281011375	25·5929678	8·6845456	·001526718
572	327184	187149248	23·9165215	8·3010304	·001748252	656	430336	282300416	25·6124969	8·6889630	·001524390
573	328329	188132517	23·9374184	8·3058651	·001745201	657	431649	283593393	25·6320112	8·6933759	·001522070
574	329476	189119224	23·9582971	8·3106941	·001742160	658	432964	284890312	25·6515107	8·6977843	·001519751
575	330625	190109375	23·9791576	8·3155175	·001739130	659	434281	286191179	25·6709953	8·7021882	·001517451
576	331776	191102976	24·0000000	8·3203353	·001736111	660	435600	287496000	25·6904652	8·7065877	·001515152
577	332929	192100033	24·0208243	8·3251475	·001733102	661	436921	288804781	25·7099203	8·7109827	·001512859
578	334084	193100552	24·0416306	8·3299542	·001730104	662	438244	290117528	25·7293607	8·7153734	·001510574
579	335241	194104539	24·0624188	8·3347553	·001727116	663	439569	291434247	25·7487864	8·7197596	·001508296
580	336400	195112000	24·0831891	8·3395509	·001724138	664	440896	292754944	25·7681975	8·7241414	·001506024

TABLE OF SQUARES, CUBES, SQUARE AND CUBE ROOTS, AND RECIPROCALS (*continued*).

Number.	Squares.	Cubes.	√Roots.	∛Roots.	Reciprocals.	Number.	Squares.	Cubes.	√Roots.	∛Roots.	Reciprocals.
665	442225	294079625	25·7875939	8·7285187	·001503759	749	561001	420189749	27·3678644	9·0815631	·001335113
666	443556	295408296	25·8069758	8·7328918	·001501502	750	562500	421875000	27·3861279	9·0856030	·001333333
667	444889	296740963	25·8263431	8·7372604	·001499250	751	564001	423564751	27·4043792	9·0896392	·001331558
668	446224	298077632	25·8456960	8·7416246	·001497006	752	565504	425259008	27·4226184	9·0936719	·001329787
669	447561	299418309	25·8650343	8·7459846	·001494768	753	567009	426957777	27·4408155	9·0977010	·001328021
670	448900	300763000	25·8843582	8·7503401	·001492537	754	568516	428661064	27·4590604	9·1017265	·001326260
671	450241	302111711	25·9036677	8·7546913	·001490313	755	570025	430368875	27·4772633	9·1057485	·001324503
672	451584	303464448	25·9229628	8·7590383	·001488095	756	571536	432081216	27·4954542	9·1097669	·001322751
673	452929	304821217	25·9422435	8·7633809	·001485884	757	573049	433798093	27·5136330	9·1137818	·001321004
674	454276	306182024	25·9615100	8·7677192	·001483680	758	574564	435519512	27·5317998	9·1177931	·001319261
675	455625	307546875	25·9807621	8·7720532	·001481481	759	576081	437245479	27·5499546	9·1218010	·001317523
676	456976	308915776	26·0000000	8·7763830	·001479290	760	577600	438976000	27·5680975	9·1258053	·001315789
677	458329	310288733	26·0192237	8·7807084	·001477105	761	579121	440711081	27·5862284	9·1298061	·001314060
678	459684	311665752	26·0384331	8·7850296	·001474926	762	580644	442450728	27·6043475	9·1338034	·001312336
679	461041	313046839	26·0576284	8·7893466	·001472754	763	582169	444194947	27·6224546	9·1377971	·001310616
680	462400	314432000	26·0768096	8·7936593	·001470588	764	583696	445943744	27·6405499	9·1417874	·001308901
681	463761	315821241	26·0959767	8·7979679	·001468429	765	585225	447697125	27·6586334	9·1457742	·001307190
682	465124	317214568	26·1151297	8·8022721	·001466276	766	586756	449455096	27·6767050	9·1497576	·001305483
683	466489	318611987	26·1342687	8·8065722	·001464129	767	588289	451217663	27·6947648	9·1537375	·001303781
684	467856	320013504	26·1533937	8·8108681	·001461988	768	589824	452984832	27·7128129	9·1577139	·001302083
685	469225	321419125	26·1725047	8·8151598	·001459854	769	591361	454756609	27·7308492	9·1616869	·001300390
686	470596	322828856	26·1916017	8·8194471	·001457726	770	592900	456533000	27·7488739	9·1656565	·001298701
687	471969	324242703	26·2106848	8·8237307	·001455604	771	594441	458314011	27·7668868	9·1696225	·001297017
688	473344	325660672	26·2297541	8·8280099	·001453488	772	595984	460099648	27·7848880	9·1735852	·001295337
689	474721	327082769	26·2488095	8·8322850	·001451379	773	597529	461889917	27·8028775	9·1775445	·001293661
690	476100	328509000	26·2678511	8·8365559	·001449275	774	599076	463684824	27·8208555	9·1815003	·001291990
691	477481	329939371	26·2868789	8·8408227	·001447178	775	600625	465484375	27·8388218	9·1854527	·001290323
692	478864	331373888	26·3058929	8·8450854	·001445087	776	602176	467288576	27·8567766	9·1894018	·001288660
693	480249	332812557	26·3248932	8·8493440	·001443001	777	603729	469097433	27·8747197	9·1933474	·001287001
694	481636	334255384	26·3438797	8·8535985	·001440922	778	605284	470910952	27·8926514	9·1972897	·001285347
695	483025	335702375	26·3628527	8·8578489	·001438849	779	606841	472729139	27·9105715	9·2012286	·001283697
696	484416	337153536	26·3818119	8·8620952	·001436782	780	608400	474552000	27·9284801	9·2051641	·001282051
697	485809	338608873	26·4007576	8·8663375	·001434720	781	609961	476379541	27·9463772	9·2090962	·001280410
698	487204	340068392	26·4196896	8·8705757	·001432665	782	611524	478211768	27·9642629	9·2130250	·001278772
699	488601	341532099	26·4386081	8·8748099	·001430615	783	613089	480048687	27·9821372	9·2169505	·001277139
700	490000	343000000	26·4575131	8·8790400	·001428571	784	614656	481890304	28·0000000	9·2208726	·001275510
701	491401	344472101	26·4764046	8·8832661	·001426534	785	616225	483736625	28·0178515	9·2247914	·001273885
702	492804	345948408	26·4952826	8·8874882	·001424501	786	617796	485587656	28·0356915	9·2287068	·001272265
703	494209	347428927	26·5141472	8·8917063	·001422475	787	619369	487443403	28·0535203	9·2326189	·001270648
704	495616	348913664	26·5329983	8·8959204	·001420455	788	620944	489303872	28·0713377	9·2365277	·001269036

705	497025	350402625	26·5518361	8·9001304	·001418440
706	498436	351895816	26·5706605	8·9043366	·001416431
707	499849	353393243	26·5894716	8·9085387	·001414427
708	501264	354894912	26·6082694	8·9127369	·001412429
709	502681	356400829	26·6270539	8·9169311	·001410437
710	504100	357911000	26·6458252	8·9211214	·001408451
711	505521	359425431	26·6645833	8·9253078	·001406470
712	506944	360944128	26·6833281	8·9294902	·001404494
713	508369	362467097	26·7020598	8·9336687	·001402525
714	509796	363994344	26·7207784	8·9378433	·001400560
715	511225	365525875	26·7394839	8·9420140	·001398601
716	512656	367061696	26·7581763	8·9461809	·001396648
717	514089	368601813	26·7768557	8·9503438	·001394700
718	515524	370146232	26·7955220	8·9545029	·001392758
719	516961	371694959	26·8141754	8·9586581	·001390821
720	518400	373248000	26·8328157	8·9628095	·001388889
721	519841	374805361	26·8514432	8·9669570	·001386963
722	521284	376367048	26·8700577	8·9711007	·001385042
723	522729	377933067	26·8886593	8·9752406	·001383126
724	524176	379503424	26·9072481	8·9793766	·001381215
725	525625	381078125	26·9258240	8·9835089	·001379310
726	527076	382657176	26·9443872	8·9876373	·001377410
727	528529	384240583	26·9629375	8·9917620	·001375516
728	529984	385828352	26·9814751	8·9958829	·001373626
729	531441	387420489	27·0000000	9·0000000	·001371742
730	532900	389017000	27·0185122	9·0041134	·001369863
731	534361	390617891	27·0370117	9·0082229	·001367989
732	535824	392223168	27·0554985	9·0123288	·001366120
733	537289	393832837	27·0739727	9·0164309	·001364256
734	538756	395446904	27·0924344	9·0205293	·001362398
735	540225	397065375	27·1108834	9·0246239	·001360544
736	541696	398688256	27·1293199	9·0287149	·001358696
737	543169	400315553	27·1477149	9·0328021	·001356852
738	544644	401947272	27·1661554	9·0368857	·001355014
739	546121	403583419	27·1845544	9·0409655	·001353180
740	547600	405224000	27·2029140	9·0450419	·001351351
741	549081	406869021	27·2213152	9·0491142	·001349528
742	550564	408518488	27·2396769	9·0531831	·001347709
743	552049	410172407	27·2580263	9·0572482	·001345895
744	553536	411830784	27·2763634	9·0613098	·001344086
745	555025	413493625	27·2946881	9·0653677	·001342282
746	556516	415160936	27·3130006	9·0694220	·001340483
747	558009	416832723	27·3313007	9·0734726	·001338688
748	559504	418508992	27·3495887	9·0775197	·001336898

789	622521	491169069	28·0891438	9·2404333	·001267427
790	624100	493039000	28·1069386	9·2443355	·001265823
791	625681	494913671	28·1247222	9·2482344	·001264223
792	627264	496793088	28·1424946	9·2521300	·001262626
793	628849	498677257	28·1602557	9·2560224	·001261034
794	630436	500566184	28·1780056	9·2599114	·001259446
795	632025	502459875	28·1957444	9·2637973	·001257862
796	633616	504358336	28·2134720	9·2676798	·001256281
797	635209	506261573	28·2311884	9·2715592	·001254705
798	636804	508169592	28·2488938	9·2754352	·001253133
799	638401	510082399	28·2665881	9·2793081	·001251564
800	640000	512000000	28·2842712	9·2831777	·001250000
801	641601	513922401	28·3019434	9·2870444	·001248439
802	643204	515849608	28·3196045	9·2909072	·001246883
803	644809	517781627	28·3372546	9·2947671	·001245330
804	646416	519718464	28·3548938	9·2986239	·001243781
805	648025	521660125	28·3725219	9·3024775	·001242236
806	649636	523606616	28·3901391	9·3063278	·001240695
807	651249	525557943	28·4077454	9·3101750	·001239157
808	652864	527514112	28·4253408	9·3140190	·001237624
809	654481	529475129	28·4429253	9·3178599	·001236094
810	656100	531441000	28·4604989	9·3216975	·001234568
811	657721	533411731	28·4780617	9·3255320	·001233046
812	659344	535387328	28·4956137	9·3293634	·001231527
813	660969	537367797	28·5131549	9·3331916	·001230012
814	662596	539353144	28·5306852	9·3370167	·001228501
815	664225	541343375	28·5482048	9·3408386	·001226994
816	665856	543338496	28·5657137	9·3446575	·001225499
817	667489	545338513	28·5832119	9·3484731	·001223990
818	669124	547343432	28·6006993	9·3522857	·001222494
819	670761	549353259	28·6181760	9·3560952	·001221001
820	672400	551368000	28·6356421	9·3599016	·001219512
821	674041	553387661	28·6530976	9·3637049	·001218027
822	675684	555412248	28·6705424	9·3675051	·001216545
823	677329	557441767	28·6879766	9·3713022	·001215067
824	678976	559476224	28·7054002	9·3750963	·001213592
825	680625	561515625	28·7228132	9·3788873	·001212121
826	682276	563559976	28·7402157	9·3826752	·001210654
827	683929	565609283	28·7576077	9·3864600	·001209190
828	685584	567663552	28·7749891	9·3902419	·001207729
829	687241	569722789	28·7923601	9·3940206	·001206273
830	688900	571787000	28·8097206	9·3977964	·001204819
831	690561	573856191	28·8270706	9·4015691	·001203369
832	692224	575930368	28·8444102	9·4053387	·001201923

TABLE OF SQUARES, CUBES, SQUARE AND CUBE ROOTS, AND RECIPROCALS (*continued*).

Number.	Squares.	Cubes.	√Roots.	∛Roots.	Reciprocals.	Number.	Squares.	Cubes.	√Roots.	∛Roots.	Reciprocals.
833	693889	578009537	28·8617394	9·4091054	·001200480	917	840889	771095213	30·2820079	9·7153051	·001090513
834	695556	580093704	28·8790582	9·4128690	·001199041	918	842724	773620632	30·2985148	9·7188354	·001089325
835	697225	582182875	28·8963666	9·4166297	·001197605	919	844561	776151559	30·3150128	9·7223631	·001088139
836	698896	584277056	28·9136646	9·4203873	·001196172	920	846400	778688000	30·3315018	9·7258883	·001086957
837	700569	586376253	28·9309523	9·4241420	·001194743	921	848241	781229961	30·3479818	9·7294109	·001085776
838	702244	588480472	28·9482297	9·4278936	·001193317	922	850084	783777448	30·3644529	9·7329309	·001084599
839	703921	590589719	28·9654967	9·4316423	·001191895	923	851929	786330467	30·3809151	9·7364484	·001083423
840	705600	592704000	28·9827535	9·4353880	·001190476	924	853776	788889024	30·3973683	9·7399634	·001082251
841	707281	594823321	29·0000000	9·4391307	·001189061	925	855625	791453125	30·4138127	9·7434758	·001081081
842	708964	596947688	29·0172363	9·4428704	·001187648	926	857476	794022776	30·4302481	9·7469857	·001079914
843	710649	599077107	29·0344623	9·4466072	·001186240	927	859329	796597983	30·4466747	9·7504930	·001078749
844	712336	601211584	29·0516781	9·4503410	·001184834	928	861184	799178752	30·4630924	9·7539979	·001077586
845	714025	603351125	29·0688837	9·4540719	·001183432	929	863041	801765089	30·4795013	9·7575002	·001076426
846	715716	605495736	29·0860791	9·4577999	·001182033	930	864900	804357000	30·4959014	9·7610001	·001075269
847	717409	607645423	29·1032644	9·4615249	·001180638	931	866761	806954491	30·5122926	9·7644974	·001074114
848	719104	609800192	29·1204396	9·4652470	·001179245	932	868624	809557568	30·5286750	9·7679922	·001072961
849	720801	611960049	29·1376046	9·4689661	·001177856	933	870489	812166237	30·5450487	9·7714845	·001071811
850	722500	614125000	29·1547595	9·4726824	·001176471	934	872356	814780504	30·5614136	9·7749743	·001070664
851	724201	616295051	29·1719043	9·4763957	·001175088	935	874225	817400375	30·5777697	9·7784616	·001069519
852	725904	618470208	29·1890390	9·4801061	·001173709	936	876096	820025856	30·5941171	9·7819466	·001068376
853	727609	620650477	29·2061637	9·4838136	·001172333	937	877969	822656953	30·6104557	9·7854288	·001067236
854	729316	622835864	29·2232784	9·4875182	·001170960	938	879844	825293672	30·6267857	9·7889087	·001066098
855	731025	625026375	29·2403830	9·4912200	·001169591	939	881721	827936019	30·6431069	9·7923861	·001064963
856	732736	627222016	29·2574777	9·4949188	·001168224	940	883600	830584000	30·6594194	9·7958611	·001063830
857	734449	629422793	29·2745623	9·4986147	·001166861	941	885481	833237621	30·6757233	9·7993336	·001062699
858	736164	631628712	29·2916370	9·5023078	·001165501	942	887364	835896888	30·6920185	9·8028036	·001061571
859	737881	633839779	29·3087018	9·5059980	·001164144	943	889249	838561807	30·7083051	9·8062711	·001060445
860	739600	636056000	29·3257566	9·5096854	·001162791	944	891136	841232384	30·7245830	9·8097362	·001059322
861	741321	638277381	29·3428015	9·5133699	·001161440	945	893025	843908625	30·7408523	9·8131989	·001058201
862	743044	640503928	29·3598365	9·5170515	·001160093	946	894916	846590536	30·7571130	9·8166591	·001057082
863	744769	642735647	29·3768616	9·5207303	·001158749	947	896809	849278123	30·7733651	9·8201169	·001055966
864	746496	644972544	29·3938769	9·5244063	·001157407	948	898704	851971392	30·7896086	9·8235723	·001054852
865	748225	647214625	29·4108823	9·5280794	·001156069	949	900601	854670349	30·8058436	9·8270252	·001053741
866	749956	649461896	29·4278779	9·5317497	·001154734	950	902500	857375000	30·8220700	9·8304757	·001052632
867	751689	651714363	29·4448637	9·5354172	·001153403	951	904401	860085351	30·8382879	9·8339238	·001051525
868	753424	653972032	29·4618397	9·5390818	·001152074	952	906304	862801408	30·8544972	9·8373695	·001050420
869	755161	656234909	29·4788059	9·5427437	·001150748	953	908209	865523177	30·8706981	9·8408127	·001049318
870	756900	658503000	29·4957624	9·5464027	·001149425	954	910116	868250664	30·8868904	9·8442536	·001048218
871	758641	660776311	29·5127091	9·5500589	·001148106	955	912025	870983875	30·9030743	9·8476920	·001047120
872	760384	663054848	29·5296461	9·5537123	·001146789	956	913936	873722816	30·9192497	9·8511280	·001046025

873	762129	665338617	29·5465734	9·5573630	·001145475	957	915849	876467493	30·9354166	9·8545617	·001044932
874	763876	667627624	29·5634910	9·5610108	·001144165	958	917764	879217912	30·9515751	9·8579929	·001043841
875	765625	669921875	29·5803989	9·5646559	·001142857	959	919681	881974079	30·9677251	9·8614218	·001042753
876	767376	672221376	29·5972972	9·5682982	·001141553	960	921600	884736000	30·9838668	9·8648483	·001041667
877	769129	674526133	29·6141858	9·5719377	·001140251	961	923521	887503681	31·0000000	9·8682724	·001040583
878	770884	676836152	29·6310648	9·5755745	·001138952	962	925444	890277128	31·0161248	9·8716941	·001039501
879	772641	679151439	29·6479342	9·5792085	·001137656	963	927369	893056347	31·0322413	9·8751135	·001038422
880	774400	681472000	29·6647939	9·5828397	·001136364	964	929296	895841344	31·0483494	9·8785305	·001037344
881	776161	683797841	29·6816442	9·5864682	·001135074	965	931225	898632125	31·0644491	9·8819451	·001036269
882	777924	686128968	29·6984848	9·5900937	·001133787	966	933156	901428696	31·0805405	9·8853574	·001035197
883	779689	688465387	29·7153159	9·5937169	·001132503	967	935089	904231063	31·0966236	9·8887673	·001034126
884	781456	690807104	29·7321375	9·5973373	·001131222	968	937024	907039232	31·1126984	9·8921749	·001033058
885	783225	693154125	29·7489496	9·6009548	·001129944	969	938961	909853209	31·1287648	9·8955801	·001031992
886	784996	695506456	29·7657521	9·6045696	·001128668	970	940900	912673000	31·1448230	9·8989830	·001030928
887	786769	697864103	29·7825452	9·6081817	·001127396	971	942841	915498611	31·1608729	9·9023835	·001029866
888	788544	700227072	29·7993289	9·6117911	·001126126	972	944784	918330048	31·1769145	9·9057817	·001028807
889	790321	702595369	29·8161030	9·6153977	·001124859	973	946729	921167317	31·1929479	9·9091776	·001027749
890	792100	704969000	29·8328678	9·6190017	·001123596	974	948676	924010424	31·2089731	9·9125712	·001026694
891	793881	707347971	29·8496231	9·6226030	·001122334	975	950625	926859375	31·2249900	9·9159624	·001025641
892	795664	709732288	29·8663690	9·6262016	·001121076	976	952576	929714176	31·2409987	9·9193513	·001024590
893	797449	712121957	29·8831056	9·6297975	·001119821	977	954529	932574833	31·2569992	9·9227379	·001023541
894	799236	714516984	29·8998328	9·6333907	·001118568	978	956484	935441352	31·2729915	9·9261222	·001022495
895	801025	716917375	29·9165506	9·6369812	·001117318	979	958441	938313739	31·2889757	9·9295042	·001021450
896	802816	719323136	29·9332591	9·6405690	·001116071	980	960400	941192000	31·3049517	9·9328839	·001020408
897	804609	721734273	29·9499583	9·6441542	·001114827	981	962361	944076141	31·3209195	9·9362613	·001019168
898	806404	724150792	29·9666481	9·6477367	·001113586	982	964324	946966168	31·3368792	9·9396363	·001018330
899	808201	726572699	29·9833287	9·6513166	·001112347	983	966289	949862087	31·3528308	9·9430092	·001017294
900	810000	729000000	30·0000000	9·6548938	·001111111	984	968256	952763904	31·3687743	9·9463797	·001016260
901	811801	731432701	30·0166621	9·6584684	·001109878	985	970225	955671625	31·3847097	9·9497479	·001015228
902	813604	733870808	30·0333148	9·6620403	·001108647	986	972196	958585256	31·4006369	9·9531138	·001014199
903	815409	736314327	30·0499584	9·6656096	·001107420	987	974169	961504803	31·4165561	9·9564775	·001013171
904	817216	738763264	30·0665928	9·6691762	·001106195	988	976144	964430272	31·4324673	9·9598389	·001012146
905	819025	741217625	30·0832179	9·6727403	·001104972	989	978121	967361669	31·4483704	9·9631981	·001011122
906	820836	743677416	30·0998339	9·6763017	·001103753	990	980100	970299000	31·4642654	9·9665549	·001010101
907	822649	746142643	30·1164407	9·6798604	·001102536	991	982081	973242271	31·4801525	9·9699095	·001009082
908	824464	748613312	30·1330383	9·6834166	·001101322	992	984064	976191488	31·4960315	9·9732619	·001008065
909	826281	751089429	30·1496269	9·6869701	·001100110	993	986049	979146657	31·5119025	9·9766120	·001007049
910	828100	753571000	30·1662063	9·6905211	·001098901	994	988036	982107784	31·5277655	9·9799599	·001006036
911	829921	756058031	30·1827765	9·6940694	·001097695	995	990025	985074875	31·5436206	9·9833055	·001005025
912	831744	758550528	30·1993377	9·6976151	·001096491	996	992016	988047936	31·5594677	9·9866488	·001004016
913	833569	761048497	30·2158899	9·7011583	·001095290	997	994009	991026973	31·5753068	9·9899900	·001003009
914	835396	763551944	30·2324329	9·7046989	·001094092	998	996004	994011992	31·5911380	9·9933289	·001002004
915	837225	766060875	30·2489669	9·7082369	·001092896	999	998001	997002999	31·6069613	9·9966656	·001001001
916	839056	768575296	30·2654919	9·7117723	·001091703	1000	1000000	1000000000	31·6227766	10·0000000	·001000000

TABLE OF FOURTH POWERS.

Numbers.	Fourth Powers.	Numbers.	Fourth Powers.	Numbers.	Fourth Powers.	Numbers.	Fourth Powers.
1·00	1	9·00	6561	17·00	83521	24·75	375232·8164
1·25	2·4414	9·25	7320·9414	17·25	88543·4414	25·00	390625
1.50	5·0625	9·50	8145·0625	17·50	93789·0625	25·25	406485·9414
1·75	9·3789	9·75	9036·8789	17·75	99264·3789	25·50	422825·0625
2·00	16	10·00	10000	18·00	104976	25·75	439651·8789
2·25	25·6289	10·25	11038·1289	18·25	110930·6289	26·00	456976
2·5	39·0625	10·50	12155·0625	18·50	117135·0625	26·25	474807·1289
2·75	57·1914	10·75	13354·6914	18·75	123596·1914	26·50	493155·0625
3·00	81	11·00	14641	19·00	130321	26·75	512029·6914
3·25	111·5664	11·25	16018·0664	19·25	137316·5664	27·00	531441
3·50	150·0625	11·50	17490·0625	19·50	144590·0625	27·25	551399·0664
3·75	197·7539	11·75	19061·2539	19·75	152148·7539	27·50	571914·0625
4·00	256	12·00	20736	20·00	160000	27·75	592996·2539
4·25	326·2539	12·25	22518·7539	20·25	168151·2539	28·00	614656
4·50	410·0625	12·50	24414·0625	20·50	176610·0625	28·25	636903·7539
4·75	509·0064	12·75	26426·5664	20·75	185384·0664	28·50	659750·0625
5·00	625	13·00	28561	21·00	194481	28·75	683205·5664
5·25	759·6914	13·25	30822·1914	21·25	203908·6914	29·00	707281
5·50	915·0625	13·50	33215·0625	21·50	213675·0625	29·25	731987·1914
5·75	1093·1289	13·75	35744·6289	21·75	223788·1289	29·50	757335·0625
6·00	1296	14·00	38416	22·00	234256	29·75	783335·6289
6·25	1525·8789	14·25	41234·3789	22·25	245086·8789	30·00	810000
6·50	1785·0625	14·50	44205·0625	22·50	256289·0625	30·25	837339·3789
6·75	2075·9414	14·75	47333·4414	22·75	267870·9414	30·50	865365·0625
7·00	2401	15·00	50625	23·00	279841	30·75	894088·4414
7·25	2762·8164	15·25	54085·3164	23·25	292207·8164	31·00	923521
7·50	3164·0625	15·50	57720·0625	23·50	304980·0625	31·25	953674·3164
7·75	3607·5039	15·75	61535·0039	23·75	318166·5039	31·50	984560·0625
8·00	4096	16·00	65536	24·00	331776	31·75	1016190·0039
8·25	4632·5039	16·25	69729·0039	24·25	345817·5039	32·00	1048576
8·50	5220·0625	16·50	74120·0625	24·50	360300·0625	32·25	1081730·0039
8·75	5861·8164	16·75	78715·3164				

LOGARITHMS.

COMMON LOGARITHMS OF NUMBERS, FROM 1 TO 9,999.

No.	0	1	2	3	4	5	6	7	8	9	
0	—	000000	301030	477121	602060	698970	778151	845098	903090	954243	
1	000000	041393	079181	113943	146128	176091	204120	230449	255273	278754	
2	301030	322219	342423	361728	380211	397940	414973	431364	447158	462398	
3	477121	491362	505150	518514	531479	544068	556303	568202	579784	591065	
4	602060	612784	623249	633468	643453	653213	662758	672098	681241	690196	
5	698970	707570	716003	724276	732394	740363	748188	755875	763428	770852	
6	778151	785330	792392	799341	806180	812913	819544	826075	832509	838849	
7	845098	851258	857332	863323	869232	875061	880814	886491	892095	897627	
8	903090	908485	913814	919078	924279	929419	934498	939519	944483	949390	
9	954243	959041	963788	968483	973128	977724	982271	986772	991226	995635	
No.	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**D.**
100	000000	000434	000868	001301	001734	002166	002598	003029	003461	003891	432
101	004321	004751	005181	005609	006038	006466	006894	007321	007748	008174	428
102	008600	009026	009451	009876	010300	010724	011147	011570	011993	012415	424
103	012837	013259	013680	014100	014521	014940	015360	015779	016197	016616	419
104	017033	017451	017868	018284	018700	019116	019532	019947	020361	020775	415
105	021189	021603	022016	022428	022841	023252	023664	024075	024486	024896	411
106	025306	025715	026125	026533	026942	027350	027757	028164	028571	028978	408
107	029384	029789	030195	030600	031004	031408	031812	032216	032619	033021	404

COMMON LOGARITHMS OF NUMBERS (*continued*).

No.	0	1	2	3	4	5	6	7	8	9	D.
108	033424	033826	034227	034628	035029	035430	035830	036230	036629	037028	400
109	037426	037825	038223	038620	039017	039414	039811	040207	040602	040998	396
110	041393	041787	042182	042576	042969	043362	043755	044148	044540	044932	393
111	045323	045714	046105	046495	046885	047275	047664	048053	048442	048830	389
112	049218	049606	049993	050380	050766	051153	051538	051924	052309	052694	386
113	053078	053463	053846	054230	054613	054996	055378	055760	056142	056524	383
114	056905	057286	057666	058046	058426	058805	059185	059563	059942	060320	379
115	060698	061075	061452	061829	062206	062582	062958	063333	063709	064083	376
116	064458	064832	065206	065580	065953	066326	066699	067071	067443	067815	373
117	068186	068557	068928	069298	069668	070038	070407	070776	071145	071514	369
118	071882	072250	072617	072985	073352	073718	074085	074451	074816	075182	366
119	075547	075912	076276	076640	077004	077368	077731	078094	078457	078819	363
120	079181	079543	079904	080266	080626	080987	081347	081707	082067	082426	360
121	082785	083144	083503	083861	084219	084576	084934	085291	085647	086004	357
122	086360	086716	087071	087426	087781	088136	088490	088845	089198	089552	354
123	089905	090258	090611	090963	091315	091667	092018	092370	092721	093071	351
124	093422	093772	094122	094471	094820	095169	095518	095866	096215	096562	349
125	096910	097257	097604	097951	098298	098644	098990	099335	099681	100026	346
126	100371	100715	101059	101403	101747	102091	102434	102777	103119	103462	343
127	103804	104146	104487	104828	105169	105510	105851	106191	106531	106871	340
128	107210	107549	107888	108227	108565	108903	109241	109579	109916	110253	338
129	110590	110926	111263	111599	111934	112270	112605	112940	113275	113609	335
130	113943	114277	114611	114944	115278	115611	115943	116276	116608	116940	332
131	117271	117603	117934	118265	118595	118926	119256	119586	119915	120245	330
132	120574	120903	121231	121560	121888	122216	122544	122871	123198	123525	327
133	123852	124178	124504	124830	125156	125481	125806	126131	126456	126781	325
134	127105	127429	127753	128076	128399	128722	129045	129368	129690	130012	322
135	130334	130655	130977	131298	131619	131939	132260	132580	132900	133219	320
136	133539	133858	134177	134496	134814	135133	135451	135769	136086	136403	318
137	136721	137037	137354	137671	137987	138303	138618	138934	139249	139564	316
138	139879	140194	140508	140822	141136	141450	141763	142076	142389	142702	313
139	143015	143327	143639	143951	144263	144574	144885	145196	145507	145818	311
140	146128	146438	146748	147058	147367	147676	147985	148294	148603	148911	309
141	149219	149527	149835	150142	150449	150756	151063	151370	151676	151982	308
142	152288	152594	152900	153205	153510	153815	154120	154424	154728	155032	304
143	155336	155640	155943	156246	156549	156852	157154	157457	157759	158061	302
144	158362	158664	158965	159266	159567	159868	160168	160469	160769	161068	300
145	161368	161667	161967	162266	162564	162863	163161	163460	163758	164055	298
146	164353	164650	164947	165244	165541	165838	166134	166430	166726	167022	296
147	167317	167613	167908	168203	168497	168792	169086	169380	169674	169968	294
148	170262	170555	170848	171141	171434	171726	172019	172311	172603	172895	292
149	173186	173478	173769	174060	174351	174641	174932	175222	175512	175802	290
150	176091	176381	176670	176959	177248	177536	177825	178113	178401	178689	288
151	178977	179264	179552	179839	180126	180413	180699	180986	181272	181558	286
152	181844	182129	182415	182700	182985	183270	183555	183839	184123	184407	284
153	184691	184975	185259	185542	185825	186108	186391	186674	186956	187239	282
154	187521	187803	188084	188366	188647	188928	189209	189490	189771	190051	281
155	190332	190612	190892	191171	191451	191730	192010	192289	192567	192846	279
156	193125	193403	193681	193959	194237	194514	194792	195069	195346	195623	277
157	195900	196176	196453	196729	197005	197281	197556	197832	198107	198382	275
158	198657	198932	199206	199481	199755	200029	200303	200577	200850	201124	274
159	201397	201670	201943	202216	202488	202761	203033	203305	203577	203848	272
160	204120	204391	204663	204934	205204	205475	205746	206016	206286	206556	270
161	206826	207096	207365	207634	207904	208173	208441	208710	208979	209247	269
162	209515	209783	210051	210319	210586	210853	211121	211388	211654	211921	267
163	212188	212454	212720	212986	213252	213518	213783	214049	214314	214579	266
164	214844	215109	215373	215638	215902	216166	216430	216694	216957	217221	264
165	217484	217747	218010	218273	218536	218798	219060	219323	219585	219846	262
166	220108	220370	220631	220892	221153	221414	221675	221936	222196	222456	260
167	222716	222976	223236	223496	223755	224015	224274	224533	224792	225051	259

COMMON LOGARITHMS OF NUMBERS (*continued*).

No.	0	1	2	3	4	5	6	7	8	9	D.
168	225309	225568	225826	226084	226342	226600	226858	227115	227372	227630	257
169	227887	228144	228400	228657	228913	229170	229426	229682	229938	230193	356
170	230449	230704	230960	231215	231470	231724	231979	232234	232488	232742	254
171	232996	233250	233504	233757	234011	234264	234517	234770	235023	235276	253
172	235528	235781	236033	236285	236537	236789	237041	237292	237544	237795	251
173	238046	238297	238548	238799	239049	239299	239550	239800	240050	240300	250
174	240549	240799	241048	241297	241546	241795	242044	242293	242541	242790	249
175	243038	243286	243534	243782	244030	244277	244525	244772	245019	245266	247
176	245513	245759	246006	246252	246499	246745	246991	247237	247482	247728	246
177	247973	248219	248464	248709	248954	249198	249443	249687	249932	250176	244
178	250420	250664	250908	251151	251395	251638	251881	252125	252368	252610	243
179	252853	253096	253338	253580	253822	254064	254306	254548	254790	255031	242
180	255273	255514	255755	255996	256237	256477	256718	256958	257198	257439	240
181	257679	257918	258158	258398	258637	258877	259116	259355	259594	259833	239
182	260071	260310	260548	260787	261025	261263	261501	261739	261976	262214	238
183	262451	262688	262925	263162	263399	263636	263873	264109	264346	264582	236
184	264818	265054	265290	265525	265761	265996	266232	266467	266702	266937	235
185	267172	267406	267641	267875	268110	268344	268578	268812	269046	269279	234
186	269513	269746	269980	270213	270446	270679	270912	271144	271377	271609	233
187	271842	272074	272306	272538	272770	273001	273233	273464	273696	273927	232
188	274158	274389	274620	274850	275081	275311	275542	275772	276002	276232	230
189	276462	276692	276921	277151	277380	277609	277838	278067	278296	278525	229
190	278754	278982	279211	279439	279667	279895	280123	280351	280578	280806	228
191	281033	281261	281488	281715	281942	282169	282396	282622	282849	283075	227
192	283301	283527	283753	283979	284205	284431	284656	284882	285107	285332	225
193	285557	285782	286007	286232	286456	286681	286905	287130	287354	287578	224
194	287802	288026	288249	288473	288696	288920	289143	289366	289589	289812	223
195	290035	290257	290480	290702	290925	291147	291369	291591	291813	292034	222
196	292256	292478	292699	292920	293141	293363	293584	293804	294025	294246	221
197	294466	294687	294907	295127	295347	295567	295787	296007	296226	296446	219
198	296665	296884	297104	297323	297542	297761	297979	298198	298416	298635	218
199	298853	299071	299289	299507	299725	299943	300161	300378	300595	300813	217
200	301030	301247	301464	301681	301898	302114	302331	302547	302764	302980	216
201	303196	303412	303628	303844	304059	304275	304491	304706	304921	305136	215
202	305351	305566	305781	305996	306211	306425	306639	306854	307068	307282	214
203	307496	307710	307924	308137	308351	308564	308778	308991	309204	309417	213
204	309630	309843	310056	310268	310481	310693	310906	311118	311330	311542	212
205	311754	311966	312177	312389	312600	312812	313023	313234	313445	313656	211
206	313867	314078	314289	314499	314710	314920	315130	315340	315551	315760	210
207	315970	316180	316390	316599	316809	317018	317227	317436	317646	317854	209
208	318063	318272	318481	318689	318898	319106	319314	319522	319730	319938	208
209	320146	320354	320562	320769	320977	321184	321391	321598	321805	322012	207
210	322219	322426	322633	322839	323046	323252	323458	323665	323871	324077	206
211	324282	324488	324694	324899	325105	325310	325516	325721	325926	326131	205
212	326336	326541	326745	326950	327155	327359	327563	327767	327972	328176	205
213	328380	328583	328787	328991	329194	329398	329601	329805	330008	330211	203
214	330414	330617	330819	331022	331225	331427	331630	331832	332034	332236	202
215	332438	332640	332842	333044	333246	333447	333649	333850	334051	334253	201
216	334454	334655	334856	335057	335257	335458	335658	335859	336059	336260	200
217	336460	336660	336860	337060	337260	337459	337659	337858	338058	338257	199
218	338456	338656	338855	339054	339253	339451	339650	339849	340047	340246	198
219	340444	340642	340841	341039	341237	341435	341632	341830	342028	342225	197
220	342423	342620	342817	343014	343212	343409	343606	343802	343999	344196	197
221	344392	344589	344785	344981	345178	345374	345570	345766	345962	346157	196
222	346353	346549	346744	346939	347135	347330	347525	347720	347915	348110	195
223	348305	348500	348694	348889	349083	349278	349472	349666	349860	350054	194
224	350248	350442	350636	350829	351023	351216	351410	351603	351796	351989	193
225	352183	352375	352568	352761	352954	353147	353339	353532	353724	353916	192
226	354108	354301	354493	354685	354876	355068	355260	355452	355643	355834	191
227	356026	356217	356408	356599	356790	356981	357172	357363	357554	357744	191

COMMON LOGARITHMS OF NUMBERS (*continued*).

No.	0	1	2	3	4	5	6	7	8	9	D.
228	357935	358125	358316	358506	358696	358886	359076	359266	359456	359646	190
229	359835	360025	360215	360404	360593	360783	360972	361161	361350	361539	189
230	361728	361917	362105	362294	362482	362671	362859	363048	363236	363424	188
231	363612	363800	363988	364176	364363	364551	364739	364926	365113	365301	187
232	365488	365675	365862	366049	366236	366423	366610	366796	366983	367169	187
233	367356	367542	367729	367915	368101	368287	368473	368659	368845	369030	186
234	369216	369401	369587	369772	369958	370143	370328	370513	370698	370883	185
235	371068	371253	371437	371622	371806	371991	372175	372360	372544	372728	184
236	372912	373096	373280	373464	373647	373831	374015	374198	374382	374565	184
237	374748	374932	375115	375298	375481	375664	375846	376029	376212	376394	183
238	376577	376759	376942	377124	377306	377488	377670	377852	378034	378216	182
239	378398	378580	378761	378943	379124	379306	379487	379668	379849	380030	181
240	380211	380392	380573	380754	380934	381115	381296	381476	381656	381837	181
241	382017	382197	382377	382557	382737	382917	383097	383277	383456	383636	179
242	383815	383995	384174	384353	384533	384712	384891	385070	385249	385428	179
243	385606	385785	385964	386142	386321	386499	386677	386856	387034	387212	178
244	387390	387568	387746	387923	388101	388279	388456	388634	388811	388989	177
245	389166	389343	389520	389698	389875	390051	390228	390405	390582	390759	177
246	390935	391112	391288	391464	391641	391817	391993	392169	392345	392521	176
247	392697	392873	393048	393224	393400	393575	393751	393926	394101	394277	175
248	394452	394627	394802	394977	395152	395326	395501	395676	395850	396025	175
249	396199	396374	396548	396722	396896	397071	397245	397419	397592	397766	174
250	397940	398114	398287	398461	398634	398808	398981	399154	399328	399501	173
251	399674	399847	400020	400192	400365	400538	400711	400883	401056	401228	173
252	401401	401573	401745	401917	402089	402261	402433	402605	402777	402949	172
253	403121	403292	403464	403635	403807	403978	404149	404320	404492	404663	171
254	404834	405005	405176	405346	405517	405688	405858	406029	406199	406370	171
255	406540	406710	406881	407051	407221	407391	407561	407731	407901	408070	170
256	408240	408410	408579	408749	408918	409087	409257	409426	409595	409764	169
257	409933	410102	410271	410440	410609	410777	410946	411114	411283	411451	169
258	411620	411788	411956	412124	412293	412461	412629	412796	412964	413132	168
259	413300	413467	413635	413803	413970	414137	414305	414472	414639	414806	167
260	414973	415140	415307	415474	415641	415808	415974	416141	416308	416474	167
261	416641	416807	416973	417139	417306	417472	417638	417804	417970	418135	166
262	418301	418467	418633	418798	418964	419129	419295	419460	419625	419791	165
263	419956	420121	420286	420451	420616	420781	420945	421110	421275	421439	165
264	421604	421768	421933	422097	422261	422426	422590	422754	422918	423082	164
265	423246	423410	423574	423737	423901	424065	424228	424392	424555	424718	163
266	424882	425045	425208	425371	425534	425697	425860	426023	426186	426349	163
267	426511	426674	426836	426999	427161	427324	427486	427648	427811	427973	162
268	428135	428297	428459	428621	428783	428944	429106	429268	429429	429591	162
269	429752	429914	430075	430236	430398	430559	430720	430881	431042	431203	161
270	431364	431525	431685	431846	432007	432167	432328	432488	432649	432809	160
271	432969	433130	433290	433450	433610	433770	433930	434090	434249	434409	160
272	434569	434729	434888	435048	435207	435367	435526	435685	435844	436004	159
273	436163	436322	436481	436640	436799	436957	437116	437275	437433	437592	159
274	437751	437909	438067	438226	438384	438542	438701	438859	439017	439175	158
275	439333	439491	439648	439806	439964	440122	440279	440437	440594	440752	157
276	440909	441066	441224	441381	441538	441695	441852	442009	442166	442323	157
277	442480	442637	442793	442950	443106	443263	443419	443576	443732	443889	156
278	444045	444201	444357	444513	444669	444825	444981	445137	445293	445449	156
279	445604	445760	445915	446071	446226	446382	446537	446692	446848	447003	155
280	447158	447313	447468	447623	447778	447933	448088	448242	448397	448552	155
281	448706	448861	449015	449170	449324	449478	449633	449787	449941	450095	154
282	450249	450403	450557	450711	450865	451018	451172	451326	451479	451633	154
283	451786	451940	452093	452247	452400	452553	452706	452859	453012	453165	153
284	453318	453471	453624	453777	453930	454082	454235	454387	454540	454692	153
285	454845	454997	455150	455302	455454	455606	455758	455910	456062	456214	152
286	456366	456518	456670	456821	456973	457125	457276	457428	457579	457731	152
287	457882	458033	458184	458336	458487	458638	458789	458940	459091	459242	151

COMMON LOGARITHMS OF NUMBERS (*continued*).

No.	0	1	2	3	4	5	6	7	8	9	D.
288	459392	459543	459694	459845	459995	460146	460296	460447	460597	460748	150
289	460898	461048	461198	461348	461499	461649	461799	461948	462098	462248	150
290	462398	462548	462697	462847	462997	463146	463296	463445	463594	463744	149
291	463893	464042	464191	464340	464490	464639	464788	464936	465085	465234	149
292	465383	465532	465680	465829	465977	466126	466274	466423	466571	466719	148
293	466868	467016	467164	467312	467460	467608	467756	467904	468052	468200	148
294	468347	468495	468643	468790	468938	469085	469233	469380	469527	469675	147
295	469822	469969	470116	470263	470410	470557	470704	470851	470998	471145	147
296	471292	471438	471585	471732	471878	472025	472171	472318	472464	472610	146
297	472756	472903	473049	473195	473341	473487	473633	473779	473925	474071	146
298	474216	474362	474508	474653	474799	474944	475090	475235	475381	475526	146
299	475671	475816	475962	476107	476252	476397	476542	476687	476832	476976	145
300	477121	477266	477411	477555	477700	477844	477989	478133	478278	478422	145
301	478566	478711	478855	478999	479143	479287	479431	479575	479719	479863	144
302	480007	480151	480294	480438	480582	480725	480869	481012	481156	481299	144
303	481443	481586	481729	481872	482016	482159	482302	482445	482588	482731	143
304	482874	483016	483159	483302	483445	483587	483730	483872	484015	484157	143
305	484300	484442	484585	484727	484869	485011	485153	485295	485437	485579	142
306	485721	485863	486005	486147	486289	486430	486572	486714	486855	486997	142
307	487138	487280	487421	487563	487704	487845	487986	488127	488269	488410	141
308	488551	488692	488833	488974	489114	489255	489396	489537	489677	489818	141
309	489958	490099	490239	490380	490520	490661	490801	490941	491081	491222	140
310	491362	491502	491642	491782	491922	492062	492201	492341	492481	492621	140
311	492760	492900	493040	493179	493319	493458	493597	493737	493876	494015	139
312	494155	494294	494433	494572	494711	494850	494989	495128	495267	495406	139
313	495544	495683	495822	495960	496099	496238	496376	496515	496653	496791	138
314	496930	497068	497206	497344	497483	497621	497759	497897	498035	498173	138
315	498311	498448	498586	498724	498862	498999	499137	499275	499412	499550	138
316	499687	499824	499962	500099	500236	500374	500511	500648	500785	500922	137
317	501059	501196	501333	501470	501607	501744	501880	502017	502154	502291	137
318	502427	502564	502700	502837	502973	503109	503246	503382	503518	503655	136
319	503791	503927	504063	504199	504335	504471	504607	504743	504878	505014	136
320	505150	505286	505421	505557	505693	505828	505964	506099	506234	506370	136
321	506505	506640	506776	506911	507046	507181	507316	507451	507586	507721	135
322	507856	507991	508126	508260	508395	508530	508664	508799	508934	509068	135
323	509203	509337	509471	509606	509740	509874	510009	510143	510277	510411	134
324	510545	510679	510813	510947	511081	511215	511349	511482	511616	511750	134
325	511883	512017	512151	512284	512418	512551	512684	512818	512951	513084	133
326	513218	513351	513484	513617	513750	513883	514016	514149	514282	514415	133
327	514548	514681	514813	514946	515079	515211	515344	515476	515609	515741	133
328	515874	516006	516139	516271	516403	516535	516668	516800	516932	517064	132
329	517196	517328	517460	517592	517724	517855	517987	518119	518251	518382	132
330	518514	518646	518777	518909	519040	519171	519303	519434	519566	519697	131
331	519828	519959	520090	520221	520353	520484	520615	520745	520876	521007	131
332	521138	521269	521400	521530	521661	521792	521922	522053	522183	522314	131
333	522444	522573	522705	522835	522966	523096	523226	523356	523486	523616	130
334	523746	523876	524006	524136	524266	524396	524526	524656	524785	524915	130
335	525045	525174	525304	525434	525563	525693	525822	525951	526081	526210	129
336	526339	526469	526598	526727	526856	526985	527114	527243	527372	527501	129
337	527630	527759	527888	528016	528145	528274	528402	528531	528660	528788	129
338	528917	529045	529174	529302	529430	529559	529687	529815	529943	530072	128
339	530200	530328	530456	530584	530712	530840	530968	531096	531223	531351	128
340	531479	531607	531734	531862	531990	532117	532245	532372	532500	532627	128
341	532754	532882	533009	533136	533264	533391	533518	533645	533772	533899	127
342	534026	534153	534280	534407	534534	534661	534787	534914	535041	535167	127
343	535294	535421	535547	535674	535800	535927	536053	536179	536306	536432	126
344	536558	536685	536811	536937	537063	537189	537315	537441	537567	537693	126
345	537819	537945	538071	538197	538322	538448	538574	538699	538825	538951	126
346	539076	539202	539327	539452	539578	539703	539829	539954	540079	540204	125
347	540329	540455	540580	540705	540830	540955	541080	541205	541330	541454	125

COMMON LOGARITHMS OF NUMBERS (*continued*).

No.	0	1	2	3	4	5	6	7	8	9	D.
348	541579	541704	541829	541953	542078	542203	542327	542452	542576	542701	125
349	542825	542950	543074	543199	543323	543447	543571	543696	543820	543944	124
350	544068	544192	544316	544440	544564	544688	544812	544936	545060	545183	124
351	545307	545431	545555	545678	545802	545925	546049	546172	546296	546419	124
352	546543	546666	546789	546913	547036	547159	547282	547405	547529	547652	123
353	547775	547898	548021	548144	548267	548389	548512	548635	548758	548881	123
354	549003	549126	549249	549371	549494	549616	549739	549861	549984	550106	123
355	550228	550351	550473	550595	550717	550840	550962	551084	551206	551328	122
356	551450	551572	551694	551816	551938	552060	552181	552303	552425	552547	122
357	552668	552790	552911	553033	553155	553276	553398	553519	553640	553762	121
358	553883	554004	554126	554247	554368	554489	554610	554731	554852	554973	121
359	555094	555215	555336	555457	555578	555699	555820	555940	556061	556182	121
360	556303	556423	556544	556664	556785	556905	557026	557146	557267	557387	120
361	557507	557627	557748	557868	557988	558108	558228	558349	558469	558589	120
362	558709	558829	558948	559068	559188	559308	559428	559548	559667	559787	120
363	559907	560026	560146	560265	560385	560504	560624	560743	560863	560982	119
364	561101	561221	561340	561459	561578	561698	561817	561936	562055	562174	119
365	562293	562412	562531	562650	562769	562887	563006	563125	563244	563362	119
366	563481	563600	563718	563837	563955	564074	564192	564311	564429	564548	119
367	564666	564784	564903	565021	565139	565257	565376	565494	565612	565730	118
368	565848	565966	566084	566202	566320	566437	566555	566673	566791	566909	118
369	567026	567144	567262	567379	567497	567614	567732	567849	567967	568084	118
370	568202	568319	568436	568554	568671	568788	568905	569023	569140	569257	117
371	569374	569491	569608	569725	569842	569959	570076	570193	570309	570426	117
372	570543	570660	570776	570893	571010	571126	571243	571359	571476	571592	117
373	571709	571825	571942	572058	572174	572291	572407	572523	572639	572755	116
374	572872	572988	573104	573220	573336	573452	573568	573684	573800	573915	116
375	574031	574147	574263	574379	574494	574610	574726	574841	574957	575072	116
376	575188	575303	575419	575534	575650	575765	575880	575996	576111	576226	115
377	576341	576457	576572	576687	576802	576917	577032	577147	577262	577377	115
378	577492	577607	577722	577836	577951	578066	578181	578295	578410	578525	115
379	578639	578754	578868	578983	579097	579212	579326	579441	579555	579669	114
380	579784	579898	580012	580126	580241	580355	580469	580583	580697	580811	114
381	580925	581039	581153	581267	581381	581495	581608	581722	581836	581950	114
382	582063	582177	582291	582404	582518	582631	582745	582858	582972	583085	114
383	583199	583312	583426	583539	583652	583765	583879	583992	584105	584218	113
384	584331	584444	584557	584670	584783	584896	585009	585122	585235	585348	113
385	585461	585574	585686	585799	585912	586024	586137	586250	586362	586475	113
386	586587	586700	586812	586925	587037	587149	587262	587374	587486	587599	112
387	587711	587823	587935	588047	588160	588272	588384	588496	588608	588720	112
388	588832	588944	589056	589167	589279	589391	589503	589615	589726	589838	112
389	589950	590061	590173	590284	590396	590507	590619	590730	590842	590953	112
390	591065	591176	591287	591399	591510	591621	591732	591843	591955	592066	111
391	592177	592288	592399	592510	592621	592732	592843	592954	593064	593175	111
392	593286	593397	593508	593618	593729	593840	593950	594061	594171	594282	111
393	594393	594503	594614	594724	594834	594945	595055	595165	595276	595386	110
394	595496	595606	595717	595827	595937	596047	596157	596267	596377	596487	110
395	596597	596707	596817	596927	597037	597146	597256	597366	597476	597586	110
396	597695	597805	597914	598024	598134	598243	598353	598462	598572	598681	110
397	598791	598900	599009	599119	599228	599337	599446	599556	599665	599774	109
398	599883	599992	600101	600210	600319	600428	600537	600646	600765	600864	109
399	600973	601082	601191	601299	601408	601517	601625	601734	601843	601951	109
400	602060	602169	602277	602386	602494	602603	602711	602819	602928	603036	108
401	603144	603253	603361	603469	603577	603686	603794	603902	604010	604118	108
402	604226	604334	604442	604550	604658	604766	604874	604982	605089	605197	108
403	605305	605413	605521	605628	605736	605844	605951	606059	606166	606274	108
404	606381	606489	606596	606704	606811	606919	607026	607133	607241	607348	107
405	607455	607562	607669	607777	607884	607991	608098	608205	608312	608419	107
406	608526	608633	608740	608847	608954	609061	609167	609274	609381	609488	107
407	609594	609701	609808	609914	610021	610128	610234	610341	610447	610554	107

Common Logarithms of Numbers (*continued*).

No.	0	1	2	3	4	5	6	7	8	9	D.
408	610660	610767	610873	610979	611086	611192	611298	611405	611511	611617	106
409	611723	611829	611936	612042	612148	612254	612360	612466	612572	612678	106
410	612784	612890	612996	613102	613207	613313	613419	613525	613630	613736	106
411	613842	613947	614053	614159	614264	614370	614475	614581	614686	614792	106
412	614897	615003	615108	615213	615319	615424	615529	615634	615740	615845	105
413	615950	616055	616160	616265	616370	616476	616581	616686	616790	616895	105
414	617000	617105	617210	617315	617420	617525	617629	617734	617839	617943	105
415	618048	618153	618257	618362	618466	618571	618676	618780	618884	618989	105
416	619093	619198	619302	619406	619511	619615	619719	619824	619928	620032	104
417	620136	620240	620344	620448	620552	620656	620760	620864	620968	621072	104
418	621176	621280	621384	621488	621592	621695	621799	621903	622007	622110	104
419	622214	622318	622421	622525	622628	622732	622835	622939	623042	623146	104
420	623249	623353	623456	623559	623663	623766	623869	623973	624076	624179	103
421	624282	624385	624488	624591	624695	624798	624901	625004	625107	625210	103
422	625312	625415	625518	625621	625724	625827	625929	626032	626135	626238	103
423	626340	626443	626546	626648	626751	626853	626956	627058	627161	627263	103
424	627366	627468	627571	627673	627775	627878	627980	628082	628185	628287	102
425	628389	628491	628593	628695	628797	628900	629002	629104	629206	629308	102
426	629410	629512	629613	629715	629817	629919	630021	630123	630224	630326	102
427	630428	630530	630631	630733	630835	630936	631038	631139	631241	631342	102
428	631444	631545	631647	631748	631849	631951	632052	632153	632255	632356	101
429	632457	632559	632660	632761	632862	632963	633064	633165	633266	633367	101
430	633468	633569	633670	633771	633872	633973	634074	634175	634276	634376	101
431	634477	634578	634679	634779	634880	634981	635081	635182	635283	635383	101
432	635484	635584	635685	635785	635886	635986	636087	636187	636287	636388	100
433	636488	636588	636688	636789	636889	636989	637089	637189	637290	637390	100
434	637490	637590	637690	637790	637890	637990	638090	638190	638290	638389	100
435	638489	638589	638689	638789	638888	638988	639088	639188	639287	639387	100
436	639486	639586	639686	639785	639885	639984	640084	640183	640283	640382	100
437	640481	640581	640680	640779	640879	640978	641077	641177	641276	641375	99
438	641474	641573	641672	641771	641871	641970	642069	642168	642267	642366	99
439	642465	642563	642662	642761	642860	642959	643058	643156	643255	643354	99
440	643453	643551	643650	643749	643847	643946	644044	644143	644242	644340	98
441	644439	644537	644636	644734	644832	644931	645029	645127	645226	645324	98
442	645422	645521	645619	645717	645815	645913	646011	646110	646208	646306	98
443	646404	646502	646600	646698	646796	646894	646992	647089	647187	647285	98
444	647383	647481	647579	647676	647774	647872	647969	648067	648165	648262	98
445	648360	648458	648555	648653	648750	648848	648945	649043	649140	649237	97
446	649335	649432	649530	649627	649724	649821	649919	650016	650113	650210	97
447	650308	650405	650502	650599	650696	650793	650890	650987	651084	651181	97
448	651278	651375	651472	651569	651666	651762	651859	651956	652053	652150	97
449	652246	652343	652440	652536	652633	652730	652826	652923	653019	653116	97
450	653213	653309	653405	653502	653598	653695	653791	653888	653984	654080	96
451	654177	654273	654369	654465	654562	654658	654754	654850	654946	655042	96
452	655138	655235	655331	655427	655523	655619	655715	655810	655906	656002	96
453	656098	656194	656290	656386	656482	656577	656673	656769	656864	656960	96
454	657056	657152	657247	657343	657438	657534	657629	657725	657820	657916	96
455	658011	658107	658202	658298	658393	658488	658584	658679	658774	658870	95
456	658965	659060	659155	659250	659346	659441	659536	659631	659726	659821	95
457	659916	660011	660106	660201	660296	660391	660486	660581	660676	660771	95
458	660865	660960	661055	661150	661245	661339	661434	661529	661623	661718	95
459	661813	661907	662002	662096	662191	662286	662380	662475	662569	662663	94
460	662758	662852	662947	663041	663135	663230	663324	663418	663512	663607	94
461	663701	663795	663889	663983	664078	664172	664266	664360	664454	664548	94
462	664642	664734	664830	664924	665018	665112	665206	665299	665393	665487	94
463	665581	665675	665769	665862	665956	666050	666143	666237	666331	666424	94
464	666518	666612	666705	666799	666892	666986	667079	667173	667266	667360	94
465	667453	667546	667640	667733	667826	667920	668013	668106	668199	668293	93
466	668386	668479	668572	668665	668759	668852	668945	669038	669131	669224	93
467	669317	669410	669503	669596	669689	669782	669875	669967	670060	670153	93

COMMON LOGARITHMS OF NUMBERS (*continued*).

No.	0	1	2	3	4	5	6	7	8	9	D.
468	670246	670339	670431	670524	670617	670710	670802	670895	670988	671080	93
469	671173	671265	671358	671451	671543	671636	671728	671821	671913	672005	92
470	672098	672190	672283	672375	672467	672560	672652	672744	672836	672929	92
471	673021	673113	673205	673297	673390	673482	673574	673666	673758	673850	92
472	673942	674034	674126	674218	674310	674402	674494	674586	674677	674769	92
473	674861	674953	675045	675137	675228	675320	675412	675503	675595	675687	92
474	675778	675870	675962	676053	676145	676236	676328	676419	676511	676602	92
475	676694	676785	676876	676968	677059	677151	677242	677333	677424	677516	91
476	677607	677698	677789	677881	677972	678063	678154	678245	678336	678427	91
477	678518	678609	678700	678791	678882	678973	679064	679155	679246	679337	91
478	679428	679519	679610	679700	679791	679882	679973	680063	680154	680245	91
479	680336	680426	680517	680607	680698	680789	680879	680970	681060	681151	91
480	681241	681332	681422	681513	681603	681693	681784	681874	681964	682055	90
481	682145	682235	682326	682416	682506	682596	682686	682777	682867	682957	90
482	683047	683137	683227	683317	683407	683497	683587	683677	683767	683857	90
483	683947	684037	684127	684217	684307	684396	684486	684576	684666	684756	90
484	684845	684935	685025	685114	685204	685294	685383	685473	685563	685652	90
485	685742	685831	685921	686010	686100	686189	686279	686368	686458	686547	90
486	686636	686726	686815	686904	686994	687083	687172	687261	687351	687440	89
487	687529	687618	687707	687796	687886	687975	688064	688153	688242	688331	89
488	688420	688509	688598	688687	688776	688865	688953	689042	689131	689220	89
489	689309	689398	689486	689575	689664	689753	689841	689930	690019	690107	89
490	690196	690285	690373	690462	690550	690639	690728	690816	690905	690993	89
491	691081	691170	691258	691347	691435	691524	691612	691700	691789	691877	88
492	691965	692053	692142	692230	692318	692406	692494	692583	692671	692759	88
493	692847	692935	693023	693111	693199	693287	693375	693463	693551	693639	88
494	693727	693815	693903	693991	694078	694166	694254	694342	694430	694517	88
495	694605	694693	694781	694868	694956	695044	695131	695219	695307	695394	88
496	695482	695569	695657	695744	695832	695919	696007	696094	696182	696269	87
497	696356	696444	696531	696618	696706	696793	696880	696968	697055	697142	87
498	697229	697317	697404	697491	697578	697665	697752	697839	697926	698014	87
499	698101	698188	698275	698362	698449	698535	698622	698709	698796	698883	87
500	698970	699057	699144	699231	699317	699404	699491	699578	699664	699751	87
501	699838	699924	700011	700098	700184	700271	700358	700444	700531	700617	87
502	700704	700790	700877	700963	701050	701136	701222	701309	701395	701482	86
503	701568	701654	701741	701827	701913	701999	702086	702172	702258	702344	86
504	702431	702517	702603	702689	702775	702861	702947	703033	703119	703205	86
505	703291	703377	703463	703549	703635	703721	703807	703893	703979	704065	86
506	704151	704236	704322	704408	704494	704579	704665	704751	704837	704922	86
507	705008	705094	705179	705265	705350	705436	705522	705607	705693	705778	86
508	705864	705949	706035	706120	706206	706291	706376	706462	706547	706632	85
509	706718	706803	706888	706974	707059	707144	707229	707315	707400	707485	85
510	707570	707655	707740	707826	707911	707996	708081	708166	708251	708336	85
511	708421	708506	708591	708676	708761	708846	708931	709015	709100	709185	85
512	709270	709355	709440	709524	709609	709694	709679	709863	709948	710033	85
513	710117	710202	710287	710371	710456	710540	710625	710710	710794	710879	86
514	710963	711048	711132	711217	711301	711385	711470	711554	711639	711723	84
515	711807	711892	711976	712060	712144	712229	712313	712397	712481	712566	84
516	712650	712734	712818	712902	712986	713070	713154	713238	713323	713407	84
517	713491	713575	713659	713742	713826	713910	713994	714078	714162	714246	84
518	714330	714414	714497	714581	714665	714749	714833	714916	715000	715084	84
519	715167	715251	715335	715418	715502	715586	715669	715753	715836	715920	84
520	716003	716087	716170	716254	716337	716421	716504	716588	716671	716754	83
521	716838	716921	717004	717088	717171	717254	717338	717421	717504	717587	83
522	717671	717754	717837	717920	718003	718086	718169	718253	718336	718419	83
523	718502	718585	718668	718751	718834	718917	719000	719083	719165	719248	83
524	719331	719414	719497	719580	719663	719745	719828	719911	719994	720077	83
525	720159	720242	720325	720407	720490	720573	720655	720738	720821	720903	83
526	720986	721068	721151	721233	721316	721398	721481	721563	721646	721728	83
527	721811	721893	721975	722058	722140	722222	722305	722387	722469	722552	82

COMMON LOGARITHMS OF NUMBERS (*continued*).

No.	0	1	2	3	4	5	6	7	8	9	D.
528	722634	722716	722798	722881	722963	723045	723127	723209	723291	723374	82
529	723456	723538	723620	723702	723784	723866	723948	724030	724112	724194	82
530	724276	724358	724440	724522	724604	724685	724767	724849	724931	725013	82
531	725095	725176	725258	725340	725422	725503	725585	725667	725748	725830	82
532	725912	725993	726075	726156	726238	726320	726401	726483	726564	726646	82
533	726727	726809	726890	726972	727053	727134	727216	727297	727379	727460	81
534	727541	727623	727704	727785	727866	727948	728029	728110	728191	728273	81
535	728354	728435	728516	728597	728678	728759	728841	728922	729003	729084	81
536	729165	729246	729327	729408	729489	729570	729651	729732	729813	729893	81
537	729974	730055	730136	730217	730298	730378	730459	730540	730621	730702	81
538	730782	730863	730944	731024	731105	731186	731266	731347	731428	731508	81
539	731589	731669	731750	731830	731911	731991	732072	732152	732233	732313	81
540	732394	732474	732555	732635	732715	732796	732876	732956	733037	733117	80
541	733197	733278	733358	733438	733518	733598	733679	733759	733839	733919	80
542	733999	734079	734160	734240	734320	734400	734480	734560	734640	734720	80
543	734800	734880	734960	735040	735120	735200	735279	735359	735439	735519	89
544	735599	735679	735759	735838	735918	735998	736078	736157	736237	736317	80
545	736397	736476	736556	736635	736715	736795	736874	736954	737034	737113	80
546	737193	737272	737352	737431	737511	737590	737670	737749	737829	737908	79
547	737987	738067	738146	738225	738305	738384	738463	738543	738622	738701	79
548	738781	738860	738939	739018	739097	739177	739256	739335	739414	739493	79
549	739572	739651	739731	739810	739889	739968	740047	740126	740205	740284	79
550	740363	740442	740521	740600	740678	740757	740836	740915	740994	741073	79
551	741152	741230	741309	741388	741467	741546	741624	741703	741782	741860	79
552	741939	742018	742096	742175	742254	742332	742411	742489	742568	742647	79
553	742725	742804	742882	742961	743039	743118	743196	743275	743353	743431	78
554	743510	743588	743667	743745	743823	743902	743980	744058	744136	744215	78
555	744293	744371	744449	744528	744606	744684	744762	744840	744919	744997	78
556	745075	745153	745231	745309	745387	745465	745543	745621	745699	745777	78
557	745855	745933	746011	746089	746167	746245	746323	746401	746479	746556	78
558	746634	746712	746790	746868	746945	747023	747101	747179	747256	747334	78
559	747412	747489	747567	747645	747722	747800	747878	747955	748033	748110	78
560	748188	748266	748343	748421	748498	748576	748653	748731	748808	748885	77
561	748963	749040	749118	749195	749272	749350	749427	749504	749582	749659	77
562	749736	749814	749891	749968	750045	750123	750200	750277	750354	750431	77
563	750508	750586	750663	750740	750817	750894	750971	751048	751125	751202	77
564	751279	751356	751433	751510	751587	751664	751741	751818	751895	751972	77
565	752048	752125	752202	752279	752356	752433	752509	752586	752663	752740	77
566	752816	752893	752970	753047	753123	753200	753277	753353	753430	753506	77
567	753583	753660	753736	753813	753889	753966	754042	754119	754195	754272	77
568	754348	754425	754501	754578	754654	754730	754807	754883	754960	755036	76
569	755112	755189	755265	755341	755417	755494	755570	755646	755722	755799	76
570	755875	755951	756027	756103	756180	756256	756332	756408	756484	756560	76
571	756636	756712	756788	756864	756940	757016	757092	757168	757244	757320	76
572	757396	757472	757548	757624	757700	757775	757851	757927	758003	758079	76
573	758155	758230	758306	758382	758458	758533	758609	758685	758761	758836	76
574	758912	758988	759063	759139	759214	759290	759366	759441	759517	759592	76
575	759668	759743	759819	759894	759970	760045	760121	760196	760272	760347	75
576	760422	760498	760573	760649	760724	760799	760875	760950	761025	761101	75
577	761176	761251	761326	761402	761477	761552	761627	761702	761778	761853	75
578	761928	762003	762078	762153	762228	762303	762378	762453	762529	762604	75
579	762679	762754	762829	762904	762978	763053	763128	763203	763278	763353	75
580	763428	763503	763578	763653	763727	763802	763877	763952	764027	764101	75
581	764176	764251	764326	764400	764475	764550	764624	764699	764774	764848	75
582	764923	764998	765072	765147	765221	765296	765370	765445	765520	765594	75
583	765669	765743	765818	765892	765966	766041	766115	766190	766264	766338	74
584	766413	766487	766562	766636	766710	766785	766859	766933	767007	767082	74
585	767156	767230	767304	767379	767453	767527	767601	767675	767749	767823	74
586	767898	767972	768046	768120	768194	768268	768342	768416	768490	768564	74
587	768638	768712	768786	768860	768934	769008	769082	769156	769230	769303	74

COMMON LOGARITHMS OF NUMBERS (*continued*).

No.	0	1	2	3	4	5	6	7	8	9	D.
588	769377	769451	769525	769599	769673	769746	769820	769894	769968	770042	74
589	770115	770189	770263	770336	770410	770484	770557	770631	770705	770778	74
590	770852	770926	770999	771073	771146	771220	771293	771367	771440	771514	74
591	771587	771661	771734	771808	771881	771955	772028	772102	772175	772248	73
592	772322	772395	772468	772542	772615	772688	772762	772835	772908	772981	73
593	773055	773128	773201	773274	773348	773421	773494	773567	773640	773713	73
594	773786	773860	773933	774006	774079	774152	774225	774298	774371	774444	73
595	774517	774590	774663	774736	774809	774882	774955	775028	775100	775173	73
596	775246	775319	775392	775465	775538	775610	775683	775756	775829	775902	73
597	775974	776047	776120	776193	776265	776338	776411	776483	776556	776629	73
598	776701	776774	776846	776919	776992	777064	777137	777209	777282	777354	73
599	777427	777499	777572	777644	777717	777789	777862	777934	778006	778079	72
600	778151	778224	778296	778368	778441	778513	778585	778658	778730	778802	72
601	778874	778947	779019	779091	779163	779236	779308	779380	779452	779524	72
602	779596	779669	779741	779813	779885	779957	780029	780101	780173	780245	72
603	780317	780389	780461	780533	780605	780677	780749	780821	780893	780965	72
604	781037	781109	781181	781253	781324	781396	781468	781540	781612	781684	72
605	781755	781827	781899	781971	782042	782114	782186	782258	782329	782401	72
606	782473	782544	782616	782688	782759	782831	782902	782974	783046	783117	72
607	783189	783260	783332	783403	783475	783546	783618	783689	783761	783832	71
608	783904	783975	784046	784118	784189	784261	784332	784403	784475	784546	71
609	784617	784689	784760	784831	784902	784974	785045	785116	785187	785259	71
610	785330	785401	785472	785543	785615	785686	785757	785828	785899	785970	71
611	786041	786112	786183	786254	786325	786396	786467	786538	786609	786680	71
612	786751	786822	786893	786964	787035	787106	787177	787248	787319	787390	71
613	787460	787531	787602	787673	787744	787815	787885	787956	788027	788098	71
614	788168	788239	788310	788381	788451	788522	788593	788663	788734	788804	71
615	788875	788946	789016	789087	789157	789228	789299	789369	789440	789510	71
616	789581	789651	789722	789792	789863	789933	790004	790074	790144	790215	70
617	790285	790356	790426	790496	790567	790637	790707	790778	790858	790918	70
618	790988	791059	791129	791199	791269	791340	791410	791480	791550	791620	70
619	791691	791761	791831	791901	791971	792041	792111	792181	792252	792322	70
620	792392	792462	792532	792602	792672	792742	792812	792882	792952	793022	70
621	793092	793162	793231	793301	793371	793441	793511	793581	793651	793721	70
622	793790	793860	793930	794000	794070	794139	794209	794279	794349	794418	70
623	794488	794558	794627	794697	794767	794836	794906	794976	795045	795115	70
624	795185	795254	795324	795393	795463	795532	795602	795672	795741	795811	70
625	795880	795949	796019	796088	796158	796227	796297	796366	796436	796505	69
626	796574	796644	796713	796782	796852	796921	796990	797060	797129	797198	69
627	797268	797337	797406	797475	797545	797614	797683	797752	797821	797890	69
628	797960	798029	798098	798167	798236	798305	798374	798443	798513	798582	69
629	798651	798720	798789	798858	798927	798996	799065	799134	799203	799272	69
630	799341	799409	799478	799547	799616	799685	799754	799823	799892	799961	69
631	800029	800098	800167	800236	800305	800373	800442	800511	800580	800648	69
632	800717	800786	800854	800923	800992	801061	801129	801198	801266	801335	69
633	801404	801472	801541	801609	801678	801747	801815	801884	801952	802021	69
634	802089	802158	802226	802295	802363	802432	802500	802568	802637	802705	69
635	802774	802842	802910	802979	803047	803116	803184	803252	803321	803389	68
636	803457	803525	803594	803662	803730	803798	803867	803935	804003	804071	68
637	804139	804208	804276	804344	804412	804480	804548	804616	804685	804753	68
638	804821	804889	804957	805025	805093	805161	805229	805297	805365	805433	68
639	805501	805569	805637	805705	805773	805841	805908	805976	806044	806112	68
640	806180	806248	806316	806384	806451	806519	806587	806655	806723	806790	68
641	806858	806926	806994	807061	807129	807197	807264	807332	807400	807467	68
642	807535	807603	807670	807738	807806	807873	807941	808008	808076	808143	68
643	808211	808279	808346	808414	808481	808549	808616	808684	808751	808818	67
644	808886	808953	809021	809088	809156	809223	809290	809358	809425	809492	67
645	809560	809627	809694	809762	809829	809896	809964	810031	810098	810165	67
646	810233	810300	810367	810434	810501	810569	810636	810703	810770	810837	67
647	810904	810971	811039	811106	811173	811240	811307	811374	811441	811508	67

COMMON LOGARITHMS OF NUMBERS (*continued*).

No.	0	1	2	3	4	5	6	7	8	9	D.
648	811575	811642	811709	811776	811843	811910	811977	812044	812111	812178	67
649	812245	812312	812379	812445	812512	812579	812646	812713	812780	812847	67
650	812913	812980	813047	813114	813181	813247	813314	813381	813448	813514	67
651	813581	813648	813714	813781	813848	813914	813981	814048	814114	814181	67
652	814248	814314	814381	814447	814514	814581	814647	814714	814780	814847	67
653	814913	814980	815046	815113	815179	815246	815312	815378	815445	815511	66
654	815578	815844	815711	815777	815843	815910	815976	816042	816109	816175	66
655	816241	816308	816374	816440	816506	816573	816639	816705	816771	816838	66
656	816904	816970	817036	817102	817169	817235	817301	817367	817433	817499	66
657	817565	817631	817698	817764	817830	817896	817962	818028	818094	818160	66
658	818226	818292	818358	818424	818490	818556	818622	818688	818754	818820	66
659	818885	818951	819017	819083	819149	819215	819281	819346	819412	819478	66
660	819544	819610	819676	819741	819807	819873	819939	820004	820070	820136	66
661	820201	820267	820333	820399	820464	820530	820595	820661	820727	820792	66
662	820858	820924	820989	821055	821120	821186	821251	821317	821382	821448	66
663	821514	821579	821645	821710	821775	821841	821906	821972	822037	822103	65
664	822168	822233	822299	822364	822430	822495	822560	822626	822691	822756	65
665	822822	822887	822952	823018	823083	823148	823213	823279	823344	823409	65
666	823474	823539	823605	823670	823735	823800	823865	823930	823996	824061	65
667	824126	824191	824256	824321	824386	824451	824516	824581	824646	824711	65
668	824776	824841	824906	824971	825036	825101	825166	825231	825296	825361	65
669	825426	825491	825556	825621	825686	825751	825815	825880	825945	826010	65
670	826075	826140	826204	826269	826334	826399	826464	826528	826593	826658	65
671	826723	826787	826852	826917	826981	827046	827111	827175	827240	827305	65
672	827369	827434	827499	827563	827628	827692	827757	827821	827886	827951	65
673	828015	828080	828144	828209	828273	828338	828402	828467	828531	828595	64
674	828660	828724	828789	828853	828918	828982	829046	829111	829175	829239	64
675	829304	829368	829432	829497	829561	829625	829690	829754	829818	829882	64
676	829947	830011	830075	830139	830204	830268	830332	830396	830460	830525	64
677	830589	830653	830717	830781	830845	830909	830973	831037	831102	831166	64
678	831230	831294	831358	831422	831486	831550	831614	831678	831742	831806	64
679	831870	831934	831998	832062	832126	832189	832253	832317	832381	832445	64
680	832509	832573	832637	832700	832764	832828	832892	832956	833020	833083	64
681	833147	833211	833275	833338	833402	833466	833530	833593	833657	833721	64
682	833784	833848	833912	833975	834039	834103	834166	834230	834294	834357	64
683	834421	834484	834548	834611	834675	834739	834802	834866	834929	834993	64
684	835056	835120	835183	835247	835310	835373	835437	835500	835564	835627	63
685	835691	835754	835817	835881	835944	836007	836071	836134	836197	836261	63
686	836324	836387	836451	836514	836577	836641	836704	836767	836830	836894	63
687	836957	837020	837083	837146	837210	837273	837336	837399	837462	837525	63
688	837588	837652	837715	837778	837841	837904	837967	838030	838093	838156	63
689	838219	838282	838345	838408	838471	838534	838597	838660	838723	838786	63
690	838849	838912	838975	839038	839101	839164	839227	839289	839352	839415	63
691	839478	839541	839604	839667	839729	839792	839855	839918	839981	840043	63
692	840106	840169	840232	840294	840357	840420	840482	840545	840608	840671	63
693	840733	840796	840859	840921	840984	841046	841109	841172	841234	841297	63
694	841359	841422	841485	841547	841610	841672	841735	841797	841860	841922	63
695	841985	842047	842110	842172	842235	842297	842360	842422	842484	842547	62
696	842609	842672	842734	842796	842859	842921	842983	843046	843108	843170	62
697	843233	843295	843357	843420	843482	843544	843606	843669	843731	843793	62
698	843855	843918	843980	844042	844104	844166	844229	844291	844353	844415	62
699	844477	844539	844601	844664	844726	844788	844850	844912	844974	845036	62
700	845098	845160	845222	845284	845346	845408	845470	845532	845594	845656	62
701	845718	845780	845842	845904	845966	846028	846090	846151	846213	846275	62
702	846337	846399	846461	846523	846585	846646	846708	846770	846832	846894	62
703	846955	847017	847079	847141	847202	847264	847326	847388	847449	847511	62
704	847573	847634	847696	847758	847819	847881	847943	848004	848066	848128	62
705	848189	848251	848312	848374	848435	848497	848559	848620	848682	848743	62
706	848805	848866	848928	848989	849051	849112	849174	849235	849297	849358	61
707	849419	849481	849542	849604	849665	849726	849788	849849	849911	849972	61

COMMON LOGARITHMS OF NUMBERS (*continued*).

No.	0	1	2	3	4	5	6	7	8	9	D.
708	850033	850095	850156	850217	850279	850340	850401	850462	850524	850585	61
709	850646	850707	850769	850830	850891	850952	851014	851075	851136	851197	61
710	851258	851320	851381	851442	851503	851564	851625	851686	851747	851809	61
711	851870	851931	851992	852053	852114	852175	852236	852297	852358	852419	61
712	852480	852541	852602	852663	852724	852785	852846	852907	852968	853029	61
713	853090	853150	853211	853272	853333	853394	853455	853516	853577	853637	61
714	853698	853759	853820	853881	853941	854002	854063	854124	854185	854245	61
715	854306	854367	854428	854488	854549	854610	854670	854731	854792	854852	61
716	854913	854974	855034	855095	855156	855216	855277	855337	855398	855459	61
717	855519	855580	855640	855701	855761	855822	855882	855943	856003	856064	61
718	856124	856185	856245	856306	856366	856427	856487	856548	856608	856668	60
719	856729	856789	856850	856910	856970	857031	857091	857152	857212	857272	60
720	857332	857393	857453	857513	857574	857634	857694	857755	857815	857875	60
721	857935	857995	858056	858116	858176	858236	858297	858357	858417	858477	60
722	858537	858597	858657	858718	858778	858838	858898	858958	659018	859078	60
723	859138	859198	859258	859318	859379	859439	859499	859559	859619	859679	60
724	859739	859799	859859	859918	859978	860038	860098	860158	860218	860278	60
725	860338	860398	860458	860518	860578	860637	860697	860757	860817	860877	60
726	860937	860996	861056	861116	861176	861236	861295	861355	861415	861475	60
727	861534	861594	861654	861714	861773	861833	861893	861952	862012	862072	60
728	862131	862191	862251	862310	862370	862430	862489	862549	862608	862668	60
729	862728	862787	862847	862906	862966	863025	863085	863144	863204	863263	60
730	863323	863382	863442	863501	863561	863620	863680	863739	863799	863858	59
731	863917	863977	864036	864096	864155	864214	864274	864333	864392	864452	59
732	864511	864570	864630	864689	864748	864808	864867	864926	864985	865045	59
733	865104	865163	865222	865282	865341	865400	865459	865519	865578	865637	59
734	865696	865755	865814	865874	865933	865992	866051	866110	866169	866228	59
735	866287	866346	866405	866465	866524	866583	866642	866701	866760	866819	59
736	866878	866937	866996	867055	867114	867173	867232	867291	867350	867409	59
737	867467	867526	867585	867644	867703	867762	867821	867880	867939	867998	59
738	868056	868115	868174	868233	868292	868350	868409	868468	868527	868586	59
739	868644	868703	868762	868821	868879	868938	868997	869056	869114	869173	59
740	869232	869290	869349	869408	869466	869525	869585	869642	869701	869760	59
741	869818	869877	869935	869994	870053	870111	870170	870228	870287	870345	59
742	870404	870462	870521	870579	870638	870696	870755	870813	870872	870930	58
743	870989	871047	871106	871164	871223	871281	871339	871398	871456	871515	58
744	871573	871631	871690	871748	871806	871865	871923	871981	872040	872098	58
745	872156	872215	872273	872331	872389	872448	872506	872564	872622	872681	58
746	872739	872797	872855	872913	872972	873030	873088	873146	873204	873262	58
747	873321	873379	873437	873495	873553	873611	873669	873727	873785	873844	58
748	873902	873960	874018	874076	874134	874192	874250	874208	874366	874424	58
749	874482	874540	874598	874656	874714	874772	874830	874888	874945	875003	58
750	875061	875119	875177	875235	875293	875351	875409	875466	875524	875582	58
751	875640	875698	875756	875813	875871	875929	875987	876045	876102	876160	58
752	876218	876276	876333	876391	876449	876507	876564	876622	876680	876737	58
753	876795	876853	876910	876968	877026	877083	877141	877199	877256	877314	58
754	877371	877429	877487	877544	877602	877659	877717	877774	877832	877889	58
755	877947	878004	878062	878119	878177	878234	878292	878349	878407	878464	57
756	878522	878579	878637	878694	878752	878809	878866	878924	878981	879039	57
757	879096	879153	879211	879268	879325	879383	879440	879497	879555	879612	57
758	879669	879726	879784	879841	879898	879956	880013	880070	880127	880185	57
759	880242	880299	880356	880413	880471	880528	880585	880642	880699	880756	57
760	880814	880871	880928	880985	881042	881099	881156	881213	881271	881328	57
761	881385	881442	881499	881556	881613	881670	881727	881784	881841	881898	57
762	881955	882012	882069	882126	882183	882240	882297	882354	882411	882468	57
763	882525	882581	882638	882695	882752	882809	882866	882923	882980	883037	57
764	883093	883150	883207	883264	883321	883377	883434	883491	883548	883605	57
765	883661	883718	883775	883832	883888	883945	884002	884059	884115	884172	57
766	884229	884285	884342	884399	884455	884512	884569	884625	884682	884739	57
767	884795	884852	884909	884965	885022	885078	885135	885192	885248	885305	57

COMMON LOGARITHMS OF NUMBERS (*continued*).

No.	0	1	2	3	4	5	6	7	8	9	D.
768	885361	885418	885474	885531	885587	885644	885700	885757	885813	885870	57
769	885926	885983	886039	886096	886152	886209	886265	886321	886378	886434	56
770	886491	886547	886604	886660	886716	886773	886829	886885	886942	886996	56
771	887054	887111	887167	887223	887280	887336	887392	887449	887505	887561	56
772	887617	887674	887730	887786	887842	887898	887955	888011	888067	888123	56
773	888179	888236	888292	888348	888404	888460	888516	888573	888629	888685	56
774	888741	888797	888853	888909	888965	889021	889077	889134	889190	889246	56
775	889302	889358	889414	889470	889526	889582	889638	889694	889750	889806	56
776	889862	889918	889974	890030	890086	890141	890197	890253	890309	890365	56
777	890421	890477	890533	890589	890645	890700	890756	890812	890868	890924	56
778	890980	891035	891091	891147	891203	891259	891314	891370	891426	891482	56
779	891537	891593	891649	891705	891760	891816	891872	891928	891983	892039	56
780	892095	892150	892206	892262	892317	892373	892429	892484	892540	892595	56
781	892651	892707	892762	892818	892873	892929	892985	893040	893096	893151	56
782	893207	893262	893318	893373	893429	893484	893540	893595	893651	893706	56
783	893762	893817	893873	893928	893984	894039	894094	894150	894205	894261	55
784	894316	894371	894427	894482	894538	894593	894648	894704	894759	894814	55
785	894870	894925	894980	895036	895091	895146	895201	895257	895312	895367	55
786	895423	895478	895533	895588	895644	895699	895754	895809	895864	895920	55
787	895975	896030	896085	896140	896195	896251	896306	896361	896416	896471	55
788	896526	896581	896636	896692	896747	896802	896857	896912	896967	897022	55
789	897077	897132	897187	897242	897297	897352	897407	897462	897517	897572	55
790	897627	897682	897737	897792	897847	897902	897957	898012	898067	898122	55
791	898176	898231	898286	898341	898396	898451	898506	898561	898615	898670	55
792	898725	898780	898835	898890	898944	898999	899054	899109	899164	899218	55
793	899273	899328	899383	899437	899492	899547	899602	899656	899711	899766	55
794	899821	899875	899930	899985	900039	900094	900149	900203	900258	900312	55
795	900367	900422	900476	900531	900586	900640	900695	900749	900804	900859	55
796	900913	900968	901022	901077	901131	901186	901240	901295	901349	901404	55
797	901458	901513	901567	901622	901676	901731	901785	901840	901894	901948	54
798	902003	902057	902112	902166	902221	902275	902329	902384	902438	902492	54
799	902547	902601	902655	902710	902764	902818	902873	902927	902981	903036	54
800	903090	903144	903199	903253	903307	903361	903416	903470	903524	903578	54
801	903633	903687	903741	903795	903849	903904	903958	904012	904066	904120	54
802	904174	904229	904283	904337	904391	904445	904499	904553	904607	904661	54
803	904716	904770	904824	904878	904932	904986	905040	905094	905148	905202	54
804	905256	905310	905364	905418	905472	905526	905580	905634	905688	905742	54
805	905796	905850	905904	905958	906012	906066	906119	906173	906227	906281	54
806	906335	906389	906443	906497	906551	906604	906658	906712	906766	906820	54
807	906874	906927	906981	907035	907089	907143	907196	907250	907304	907358	54
808	907411	907465	907519	907573	907626	907680	907734	907787	907841	907895	54
809	907949	908002	908056	908110	908163	908217	908270	908324	908378	908431	54
810	908485	908539	908592	908646	908699	908753	908807	908860	908914	908967	54
811	909021	909074	909128	909181	909235	909289	909342	909396	909449	909503	54
812	909556	909610	909663	909716	909770	909823	909877	909930	909984	910037	53
813	910091	910144	910197	910251	910304	910358	910411	910464	910518	910571	53
814	910624	910678	910731	910784	910838	910891	910944	910998	911051	911104	53
815	911158	911211	911264	911317	911371	911424	911477	911530	911584	911637	53
816	911690	911743	911797	911850	911903	911956	912009	912063	912116	912169	53
817	912222	912275	912328	912381	912435	912488	912541	912594	912647	912700	53
818	912753	912806	912859	912913	912966	913019	913072	913125	913178	913231	54
819	913284	913337	913390	913443	913496	913549	913602	913655	913708	913761	53
820	913814	913867	913920	913973	914026	914079	914132	914184	914237	914290	53
821	914343	914396	914449	914502	914555	914608	914660	914713	914766	914819	53
822	914872	914925	914977	915030	915083	915136	915189	915241	915294	915347	53
823	915400	915453	915505	915558	915611	915664	915716	915769	915822	915875	53
824	915927	915980	916033	916085	916138	916191	916243	916296	916349	916401	53
825	916454	916507	916559	916612	916664	916717	916770	916822	916875	916927	53
826	916980	917033	917085	917138	917190	917243	917295	917348	917400	917453	53
827	917506	917558	917611	917663	917716	917768	917820	917873	917925	917978	52

COMMON LOGARITHMS OF NUMBERS (*continued*).

No.	0	1	2	3	4	5	6	7	8	9	D.
828	918030	918083	918135	918188	918240	918293	918345	918397	918450	918502	52
829	918555	918607	918659	918712	918764	918816	918869	918921	918973	919026	52
830	919078	919130	919183	919235	919287	919340	919392	919444	919496	919549	52
831	919601	919653	919706	919758	919810	919862	919914	919967	920019	920071	52
832	920123	920176	920228	920280	920332	920384	920436	920489	920541	920593	52
833	920645	920697	920749	920801	920853	920906	920958	921010	921062	921114	52
834	921166	921218	921270	921322	921374	921426	921478	921530	921582	921634	52
835	921686	921738	921790	921842	921894	921946	921998	922050	922102	922154	52
836	922206	922258	922310	922362	922414	922466	922518	922570	922622	922674	52
837	922725	922777	922829	922881	922933	922985	923037	923089	923140	923192	52
838	923244	923296	923348	923399	923451	923503	923555	923607	923658	923710	52
839	923762	923814	923865	923917	923969	924021	924072	924124	924176	924228	52
840	924279	924331	924383	924434	924486	924538	924589	924641	924693	924744	52
841	924796	924848	924899	924951	925003	925054	925106	925157	925209	925261	52
842	925312	925364	925415	925467	925518	925570	925621	925673	925725	925776	52
843	925828	925879	925931	925982	926034	926085	926137	926188	926240	926291	51
844	926342	926394	926445	926497	926548	926600	926651	926702	926754	926805	51
845	926857	926908	926959	927011	927062	927114	927165	927216	927268	927319	51
846	927370	927422	927473	927524	927576	927627	927678	927730	927781	927832	51
847	927883	927935	927986	928037	928088	928140	928191	928242	928293	928345	51
848	928396	928447	928498	928549	928601	928652	928703	928754	928805	928857	51
849	928908	928959	929010	929061	929112	929163	929215	929266	929317	929368	51
850	929419	929470	929521	929572	929623	929674	929725	929776	929827	929879	51
851	929930	929981	930032	930083	930134	930185	930236	930287	930338	930389	51
852	930440	930491	930542	930592	930643	930694	930745	930796	930847	930898	51
853	930949	931000	931051	931102	931153	931204	931254	931305	931356	931407	51
854	931458	931509	931560	931610	931661	931712	931763	931814	931865	931915	51
855	931966	932017	932068	932118	932169	932220	932271	932322	932372	932423	51
856	932474	932524	932575	932626	932677	932727	932778	932829	932879	932930	51
857	932981	933031	933082	933133	933183	933234	933285	933335	933386	933437	51
858	933487	933538	933589	933639	933699	933740	933791	933841	933892	933943	51
859	933993	934044	934094	934145	934195	934246	934296	934347	934397	934448	51
860	934498	934549	934599	934650	934700	934751	934801	934852	934902	934953	50
861	935003	935054	935104	935154	935205	935255	935306	935356	935406	935457	50
862	935507	935558	935608	935658	935709	935759	935809	935860	935910	935960	50
863	936011	936061	936111	936162	936212	936262	936313	936363	936413	936463	50
864	936514	936564	936614	936665	936715	936765	936815	936865	936916	936966	50
865	937016	937066	937117	937167	937217	937267	937317	937367	937418	937468	50
866	937518	937568	937618	937668	937718	937769	937819	937869	937919	937969	50
867	938019	938069	938119	938169	938219	938269	938320	938370	938420	938470	50
868	938520	938570	938620	938670	938720	938770	938820	938870	938920	938970	50
869	939020	939070	939120	939170	939220	939270	939320	939369	939419	939469	50
870	939519	939569	939619	939669	939719	939769	939819	939869	939918	939968	50
871	940018	940068	940118	940168	940218	940267	940317	940367	940417	940467	50
872	940516	940566	940616	940666	940716	940765	940815	940865	940915	940964	50
873	941014	941064	941114	941163	941213	941263	941313	941362	941412	941462	50
874	941511	941561	941611	941660	941710	941760	941809	941859	941909	941958	50
875	942008	942058	942107	942157	942207	942256	942306	942355	942405	942455	50
876	942504	942554	942603	942653	942702	942752	942801	942851	942901	942950	50
877	943000	943049	943099	943148	943198	943247	943297	943346	943396	943445	49
878	943495	943544	943593	943643	943692	943742	943791	943841	943890	943939	49
879	943989	944038	944088	944137	944186	944236	944285	944335	944384	944433	49
880	944483	944532	944581	944631	944680	944729	944779	944828	944877	944927	49
881	944976	945025	945074	945124	945173	945222	945272	945321	945370	945419	49
882	845469	945518	945567	945616	945665	945715	945764	945813	945862	945912	49
883	945961	946010	946059	946108	946157	946207	946256	946305	946354	946403	49
884	946452	946501	946551	946600	946649	946698	946747	946796	946845	946894	49
885	946943	946992	947041	947090	947140	947189	947238	947287	947336	947385	49
886	947434	947483	947532	947581	947630	947679	947728	947777	947826	947875	49
887	947924	947973	948022	948070	948119	948168	948217	948266	948315	948364	49

COMMON LOGARITHMS OF NUMBERS (*continued*).

No.	0	1	2	3	4	5	6	7	8	9	D.
888	948413	948462	948511	948560	948609	948657	948706	948755	948804	948853	49
889	948902	948951	948999	949048	949097	949146	949195	949244	949292	949341	49
890	949390	949439	949488	949536	949585	949634	949683	949731	949780	949829	49
891	949878	949926	949975	950024	950073	950121	950170	950219	950267	950316	49
892	950365	950414	950462	950511	950560	950608	950657	950706	950754	950803	49
893	950851	950900	950949	950997	951046	951095	951143	951192	951240	951289	49
894	951338	951386	951435	951483	951532	951580	951629	951677	951726	951775	49
895	951823	951872	951920	951969	952017	952066	952114	952163	952211	952260	48
896	952308	952356	952405	952453	952502	952550	952599	952647	952696	952744	48
897	952792	952841	952889	952938	952986	953034	953083	953131	953180	953228	48
898	953276	953325	953373	953421	953470	953518	953566	953615	953663	953711	48
899	953760	953808	953856	953905	953953	954001	954049	954098	954146	954194	48
900	954243	954291	954339	954387	954435	954484	954532	954580	954628	954677	48
901	954725	954773	954821	954869	954918	954966	955014	955062	955110	955158	48
902	955207	955255	955303	955351	955399	955447	955495	955543	955592	955640	48
903	955688	955736	955784	955832	955880	955928	955976	956024	956072	956120	48
904	956168	956216	956265	956313	956361	956409	956457	956505	956553	956601	48
905	956649	956697	956745	956793	956840	956888	956936	956984	957032	957080	48
906	957128	957176	957224	957272	957320	957368	957416	957464	957512	957559	48
907	957607	957655	957703	957751	957799	957847	957894	957942	957990	958038	48
908	958086	958134	958181	958229	958277	958325	958373	958421	958468	958516	48
909	958564	958612	958659	958707	958755	958803	958850	958898	958946	958994	48
910	959041	959089	959137	959185	959232	959280	959328	959375	959423	959471	48
911	959518	959566	959614	959661	959709	959757	959804	959852	959900	959947	48
912	959995	960042	960090	960138	960185	960233	960281	960328	960376	960423	48
913	960471	960518	960566	960613	960661	930709	960756	960804	960851	960894	48
914	960946	960994	961041	961089	961136	961184	961231	961279	961326	961374	47
915	961421	961469	961516	961563	961611	961658	961706	961753	961801	961848	47
916	961895	961943	961990	962038	962085	962132	962180	962227	962275	962325	47
917	962369	962417	962464	962511	962559	962606	962653	962701	962748	962795	47
918	962843	962890	962937	962985	963032	963079	963126	963174	963221	963268	47
919	963316	963363	963410	963457	963504	963552	963599	963646	963693	963741	47
920	963788	963835	963882	963929	963977	964024	964071	964118	964165	964212	47
921	964260	964307	964354	964401	964448	964495	964542	964590	964637	964684	47
922	964731	964778	964825	964872	964919	964966	965013	965061	965108	965155	47
923	965202	965249	965296	965343	965390	965437	965484	965531	965578	965625	47
924	965672	965719	965766	965813	965860	965907	965954	966001	966048	966095	47
925	966142	966189	966236	966283	966329	966376	966423	966470	966517	966564	47
926	966611	966658	966705	966752	966799	966845	966892	966939	966986	967033	47
927	967080	967127	967173	967220	967267	967314	967361	967408	967454	967501	47
928	967548	967595	967642	967688	967735	967782	967829	967875	967922	967969	47
929	968016	968062	968109	968156	968203	968249	968296	968343	968390	968436	47
930	968483	968530	968576	968623	968670	968716	968763	968810	968856	968903	47
931	968950	968996	969043	969090	969136	969183	969229	969276	969323	969369	47
932	969416	969463	969509	969556	969602	969649	969695	969742	969789	969835	47
933	969882	969928	969975	970021	970068	970114	970161	970207	970254	970300	47
934	970347	970393	970440	970486	970533	970579	970626	970672	970719	970765	46
935	970812	970858	970904	970951	970997	971044	971090	971137	971183	971229	46
936	971276	971322	971369	971415	971461	971508	971554	971601	971647	971693	46
937	971740	971786	971832	971879	971925	971971	972018	972064	972110	972157	46
938	972203	972249	972295	972342	972388	972434	972481	972527	972573	972619	46
939	972666	972712	972758	972804	972851	972897	972943	972989	973035	793082	46
940	973128	973174	973220	973266	973313	973359	973405	973451	973497	973543	46
941	973590	973636	973682	973728	973774	973820	973866	973913	973959	974005	46
942	974051	974097	974143	974189	974235	974281	974327	974374	974420	974466	46
943	974512	974558	974604	974650	974696	974742	974798	974834	974880	974926	46
944	974972	975018	975064	975110	975156	975202	975248	975294	975340	975386	46
945	975432	975478	975524	975570	975616	975662	975707	975753	975799	975845	46
946	975891	975937	975983	976029	976075	976121	976167	976212	976258	976304	46
947	976350	976396	976442	976488	976533	976579	976625	976671	976717	976763	46

Common Logarithms of Numbers (*continued*).

No.	0	1	2	3	4	5	6	7	8	9	D.
948	976808	976854	976900	976946	976992	977037	977083	977129	977175	977220	46
949	977266	977312	977358	977403	977449	977495	977541	977586	977632	977678	46
950	977724	977769	977815	977861	977906	977952	977998	978043	978089	978135	46
951	978181	978226	978272	978317	978363	978409	978454	978500	978546	978591	46
952	978637	978683	978728	978774	978819	978865	978911	978956	979002	979047	46
953	979093	979138	979184	979230	979275	979321	979366	979412	979457	979503	46
954	979548	979594	979639	979685	979730	979776	979821	979867	979912	979958	46
955	980003	980049	980094	980140	980185	980231	980276	980322	980367	980412	45
956	980458	980503	980549	980594	980640	980685	980730	980776	980821	980867	45
957	980912	980957	981003	981048	981093	981139	981184	981229	981275	981320	45
958	981366	981411	981456	981501	981547	981592	981637	981683	981728	981773	45
959	981819	981864	981909	981954	982000	982045	982090	982135	982181	982226	45
960	982271	982316	982362	982407	982452	982497	982543	982588	982633	982678	45
961	982723	982769	982814	982859	982904	982949	982994	983040	983085	983130	45
962	983175	983220	983265	983310	983356	983401	983446	983491	983536	983581	45
963	983626	983671	983716	983762	983807	983852	983897	983942	983987	984032	45
964	984077	984122	984167	984212	984257	984302	984347	984392	984437	984482	45
965	984527	984572	984617	984662	984707	984752	984797	984842	984887	984932	45
966	984977	985022	985067	985112	985157	985202	985247	985292	985337	985382	45
967	985426	985471	985516	985561	985606	985651	985696	985741	985786	985830	45
968	985875	985920	985965	986010	986055	986100	986144	986189	986234	986279	45
969	986324	986369	986413	986458	986503	986548	986593	986637	986682	986727	45
970	986772	986817	986861	986906	986951	986996	987040	987085	987130	987175	45
971	987219	987264	987309	987353	987398	987443	987488	987532	987577	987622	45
972	987666	987711	987756	987800	987845	987890	987934	987979	988024	988068	45
973	988113	988157	988202	988247	988291	988336	988381	988425	988470	988514	45
974	988559	988604	988648	988693	988737	988782	988826	988871	988916	988960	45
975	989005	989049	989094	989138	989183	989227	989272	989316	989361	989405	45
976	989450	989494	989539	989583	989628	989672	989717	989761	989806	989850	44
977	989895	989939	989983	990028	990072	990117	990161	990206	990250	990294	44
978	990339	990383	990428	990472	990516	990561	990605	990650	990694	990738	44
979	990783	990827	990871	990916	990960	991004	991049	991093	991137	991182	44
980	991226	991270	991315	991359	991403	991448	991492	991536	991580	991625	44
981	991669	991713	991758	991802	991846	991890	991935	991979	992023	992067	44
982	992111	992156	992200	992244	992288	992333	992377	992421	992465	992509	44
983	992554	992598	992642	992686	992730	992774	992819	992863	992907	992951	44
984	992995	993039	993083	993127	993172	993216	993260	993304	993348	993392	44
985	993436	993480	993524	993568	993613	993657	993701	993745	993789	993833	44
986	993877	993921	993965	994009	994053	994097	994141	994185	994229	994273	44
987	994317	994361	994405	994449	994493	994537	994581	994625	994669	994713	44
988	994757	994801	994845	994889	994933	994977	995021	995065	995108	995152	44
989	995196	995240	995284	995328	995372	995416	995460	995504	995547	995591	44
990	995635	995679	995723	995767	995811	995854	995898	995942	995986	996030	44
991	996074	996117	996161	996205	996249	996293	996337	996380	996424	996468	44
992	996512	996555	996599	996643	996687	996731	996774	996818	996862	996906	44
993	996949	996993	997037	997080	997124	997168	997212	997255	997299	997343	44
994	997386	997430	997474	997517	997561	997605	997648	997692	997736	997779	44
995	997823	997867	997910	997954	997998	998041	998085	998129	998172	998216	44
996	998259	998303	998347	998390	998434	998477	998521	998564	998608	998652	44
997	998695	998739	998782	998826	998869	998913	998956	999000	999043	999087	44
998	999131	999174	999218	999261	999305	999348	999392	999435	999479	999522	44
999	999565	999609	999652	999696	999739	999783	999826	999870	999913	999957	43

Characteristic of log. of number containing one or more integral figs. is one *less* than the number of integral figs., and is positive.

Characteristic of log. of a number wholly decimal is one *more* than number of noughts at commencement of decimal, and is negative.

To multiply by logs., add logs., and find corresponding numbers.

To divide by logs.. subtract logs., and find corresponding numbers.

To extract the root, divide log by index, and find corresponding numbers.

To obtain any power, multiply log. by index, and find corresponding numbers.

Use of Differences.

Ex. 1.—Find the log. of 32685.
Difference (D) for 326=133 log. 32680=4·514282†
133×5= 665*
4·5143485

In all cases last fig. of * must be set one beyond last fig. of †.

Ex. 2.—Find number corresponding to log. 3·774743
Given log. . . . . 3·774743
Nearest log. in table . . 3·774736=log. 5953
Divide by Difference (D) 73)7 01
Required number =5953·1

Hyperbolic or Natural Logarithms.

Hyp. log. = Common log. × 2·30258509. Common log. = Hyp. log. × ·43429448.

No.	0·0	0·1	0·2	0·3	0·4	0·5	0·6	0·7	0·8	0·9
0	—∞	$\bar{3}$·69742	$\bar{2}$·39056	$\bar{2}$·79602	$\bar{1}$·08371	$\bar{1}$·30685	$\bar{1}$·48918	$\bar{1}$·64332	$\bar{1}$·77685	$\bar{1}$·89463
1	0·00000	0·09530	0·18213	0·26234	0·33646	0·40546	0·46998	0·53063	0·58776	0·64181
2	0·69315	0·74190	0·78843	0·83287	0·87544	0·91629	0·95548	0·99323	1·02962	1·06473
3	1·09861	1·13140	1·16314	1·19594	1·22373	1·25276	1·28090	1·30834	1·33406	1·36099
4	1·38629	1·41096	1·43505	1·45859	1·48161	1·50408	1·52603	1·54753	1·56859	1·58922
5	1·60944	1·62922	1·64865	1·66770	1·68633	1·70475	1·72276	1·74046	1·75785	1·77495
6	1·79175	1·80827	1·82545	1·84055	1·85629	1·87180	1·88658	1·90218	1·91689	1·93149
7	1·94591	1·96006	1·97406	1·98787	2·00149	2·01490	2·02816	2·04115	2·05415	2·06690
8	2·07944	2·09190	2·10418	2·11632	2·12830	2·14007	2·15082	2·16338	2·17482	2·18615
9	2·19722	2·20837	2·21932	2·23014	2·24085	2·25129	2·26191	2·27228	2·28255	2·29171

No.	0	1	2	3	4	5	6	7	8	9
10	2·30258	2·39789	2·48491	2·56494	2·63906	2·70805	2·77259	2·83321	2·89037	2·94444
20	·99573	3·04452	3·09104	3·13549	3·17805	3·21888	3·25810	3·29584	3·33220	3·36730
30	3·40120	·43399	·46574	·49651	·52636	·55535	·58352	·61092	·63759	·66356
40	·68888	·71357	·73767	·76120	·78419	·80666	·82864	·85015	·87120	·89182
50	·91202	·93183	·95124	·97029	·98898	4·00733	4·02535	4·04305	4·06044	4·07754
60	4·09434	4·11087	4·12713	4·14313	4·15888	·17439	·18965	·20469	·21951	·23411
70	·24849	·26268	·27667	·29046	·30406	·31749	·33073	·34380	·35671	·36945
80	·38203	·39445	·40672	·41884	·43082	·44265	·45435	·46591	·47734	·48864
90	·49981	·51086	·52179	·53260	·54329	·55388	·56435	·57471	·58497	·59512
100	4·60517	4·61512	4·62497	4 63473	4·64439	4·65396	4·66343	4·67283	4·68213	4·69135
110	·70048	·70953	·71849	·72739	·73619	·74493	·75359	·76217	·77068	·77912
120	·78749	·79579	·80402	·81218	·82028	·82831	·83628	·84418	·85203	·85981
130	·86753	·87519	·88280	·89035	·89784	·90527	·91265	·91998	·92725	·93447
140	·94164	·94876	·95583	·96284	·96981	·97673	·98360	·99043	·99721	5·00394
150	5·01063	5·01728	5·02388	5·03044	5·03695	5·04342	5·04985	5·05624	5·06259	5·06890
160	·07517	·08140	·08760	·09375	·09986	·10594	·11199	·11799	·12396	·12990
170	·13580	·14166	·14749	·15329	·15905	·16478	·17048	·17615	·18178	·18738
180	·19295	·19850	·20400	·20948	·21493	·22035	·22574	·23111	·23644	·24175
190	·24702	·25227	·25750	·26269	·26786	·27300	·27811	·28320	·28826	·29330
200	5·29832	5·30330	5·30826	5·31320	5·31812	5·32301	5·32787	5·33272	5·33754	5·34233
210	·34711	·35186	·35658	·36129	·36597	·37064	·37528	·37989	·38450	·38907
220	·39363	·39816	·40268	·40717	·41164	·41610	·42053	·42495	·42934	·43372
230	·43808	·44242	·44674	·45104	·45532	·45958	·46383	·46806	·47227	·47646
240	·48064	·48479	·48893	·49306	·49716	·50126	·50533	·50939	·51343	·51745
250	5·52146	5·52545	5·52943	5·53339	5·53733	5·54126	5·54517	5·54907	5·55296	5·55683
260	·56068	·56452	·56834	·57215	·57595	·57973	·58349	·58725	·59098	·59471
270	·59842	·60212	·60580	·60947	·61313	·61677	·62040	·62402	·62762	·63121
280	·63479	·63835	·64191	·64544	·64897	·65249	·65599	·65948	·66296	·66642
290	·66988	·67332	·67675	·68017	·68358	·68697	·69036	·69373	·69709	·70044
300	5·70378	5·70711	5·71043	5·71373	5·71703	5·72031	5·72358	5·72685	5·73010	5·73334
310	·73657	·73979	·74300	·74620	·74939	·75257	·75574	·75890	·76205	·76519
320	·76832	·77144	·77455	·77765	·78074	·78382	·78690	·78996	·79301	·79606
330	·79909	·80212	·80513	·80814	·81114	·81413	·81711	·82008	·82304	·82600
340	·82894	·83188	·83481	·83773	·84064	·84354	·84644	·84932	·85220	·85507
350	5·85793	5·86078	5·86363	5·86647	5·86929	5·87212	5·87493	5·87773	5·88053	5·88332

DESCRIPTIVE SECTIONS

SECTION II

PART II

Powers.

FROUDE HYDRAULIC DYNAMOMETER.

(Heenan and Froude Limited, Worcester, England.)

This dynamometer is standardised in the following chief types :—

1. *Types D.P.X., D.P.Y. and D.P.Y.D.*, in small and medium sizes up to 4,000 B.H.P. for testing the B.H.P. of engines such as those of motor cycles, motor cars and commercial road vehicles, high speed Diesel engines, etc.

2. *Types S.A. and F.A.*, in medium and large sizes for dealing with comparatively slow speed prime movers such as land and marine Diesels, oil, gas, and steam engines, slow speed or geared turbines, etc. Machines have already been constructed up to 60,000 B.H.P. in a single unit but a wide range of smaller sizes is available.

3. *Type F.V.*, which operates on somewhat similar principles to those of the F.A. and S.A. types, and which is employed for high-speed high-powered engines such as aircraft engines and turbines up to 5,000 B.H.P.

In operation the engine upon test is directly coupled to the dynamometer shaft and transmits power to a rotor revolving inside a casing, through which water is circulated to provide the

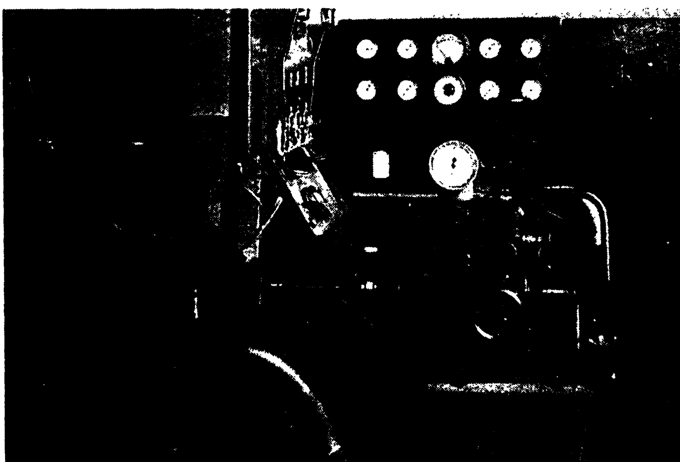


FIG. 1.

hydraulic resistance and simultaneously to carry away the heat generated by destruction of power. In the standard D.P.X. type dynamometer the load imposed is controlled by varying the position of metal sluice plates which mask the rotor to a greater or lesser extent; this operation can be carried out while running by the operation of a single handwheel. The dynamometer casing is supported upon anti-friction trunnion bearings so that it is free to oscillate under the influence of the torque absorbed, and is only restrained from doing so by the torque weighing gear connected to a balance arm bolted to the casing. By this means all forces resisting rotation of the dynamometer shaft are caused to react upon the weighing gear, which thus gives scientifically accurate readings in all conditions of load and speed.

The D.P.Y. and D.P.Y.D. are basically similar to the D.P.X. type, but a different form of weighing gear is used, adapted for heavier torques, and the D.P.Y.D. in particular is of higher capacity. Special high-speed models are available for testing such prime movers as gas turbines, steam turbines, etc.

In the S.A. and F.A. Types, similar power absorbing elements are employed, but the method of road control is different. Instead of the rotor cups always running full of water as in the D.P.X. type, the volume of water contained therein is varied by the operation of a valve and the D.P.X. type sluice-plates are not employed. This valve is of a special design which gives the dynamometer automatic self-governing properties such that should the speed of the engine increase, the

dynamometer load also increases. The contrary effect takes place upon a decrease in speed, and these features are of great assistance in keeping steady the speed of the engine upon test and minimising 'hunting,' often caused previously by the usual slight variations of engine output.

These large machines are also provided with a Torque Meter which eliminates the long balance arm and heavy weights previously necessary; it consists of a system which allows the torque to be accurately measured by small weights and greatly reduces the test shop floor space required by the dynamometer.

All 'Froude' Dynamometers have an extremely wide range of power and speed and one size can thus cater for a large variety of engines. Except in the smaller sizes, they incorporate means for adjusting the centres of the dynamometer shaft for ease in alignment of the engine, and in all cases they form self-contained units complete with their own bedplates and torque measurement gear, only requiring to be coupled to the engine and supplied with water to be in readiness for tests.

HEENAN-DYNAMATIC DYNAMOMETERS.

The Heenan-Dynamatic Dynamometer originated in America, where it has been installed on a wide scale, and has now come into extensive use in this country. It comprises a steel rotor, having no electrical windings or connections, running inside a plain steel housing, the latter being surrounded by a fixed coil. Rotation of the rotor inside the field set up by the coil causes the generation of eddy currents and thus absorbs power, the load being regulated by varying the excitation of the coil. The machine can be supplied in a range of capacities up to considerable powers. In all models the control gear is of electronic type, made in various standard forms, which provide definite technical advantages and make the Dynamometer particularly easy to operate.

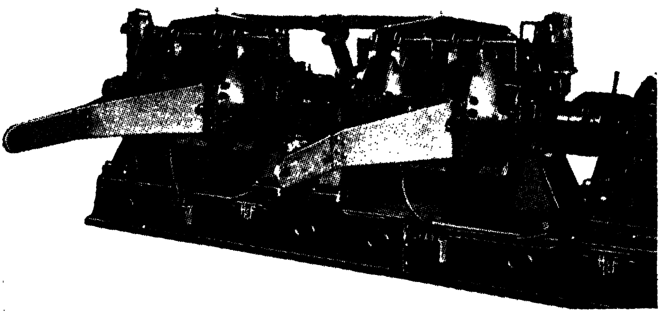


FIG. 2.

Being inherently reversible, it is specially useful to makers of marine engines and is now standard practice for testing aircraft engines. Special models are available for testing contra-rotation engines, an example of such a machine being illustrated in fig. 2. Very high speeds are catered for by special designs; machines have already been made for 2,400 B.H.P. at 24,000 r.p.m.

HEENAN ELECTRIC DYNAMOMETERS.

The Heenan D.C. Electric Dynamometers incorporate special control gear and are normally provided with torque weighing mechanism constructed on Froude principles and giving the same strict accuracy of reading. Models are available for production testing, research testing, or a combination of both; they can be made automatic or semi-automatic, and recover the engine power in the form of electric current, in addition to the benefit of motoring the engine for running-in or research.

A.C. Electric Dynamometers are also available for certain duties, particularly as Motoring Dynamometers for ascertaining the B.H.P. absorbed at various speeds by the mechanisms on test.

FROUDE AIRCRAFT ENGINE TESTING PLANT.

The Froude Wind Tunnel Testing Plant has been installed by most of the chief aircraft engine builders and by many foreign governments, and consists of a Dynamometer for testing the B.H.P. of the engine, acting in conjunction with a centrifugal or axial-flow fan which is employed when testing air-cooled engines for directing the necessary cooling air-blast over the engine. This plant is usually supplied complete in all details including wind tunnel itself with fan and driving motor, dynamometer of suitable type and capacity, cardan shaft between dynamometer and engine, universal engine cradle, and the necessary instruments and accessories such as petrol and

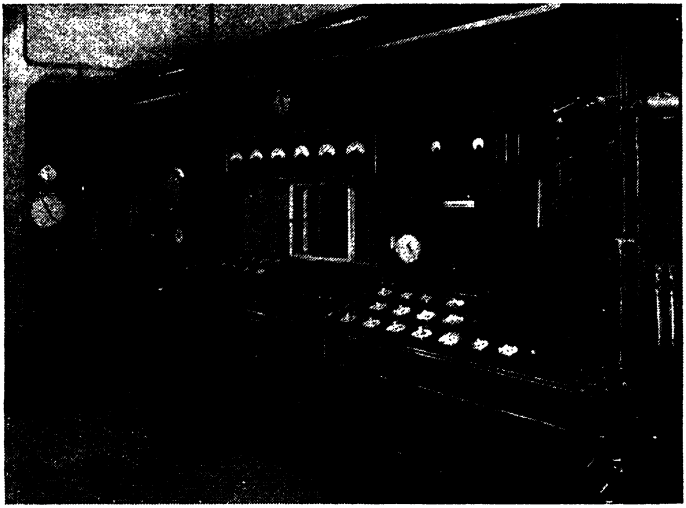


FIG. 3.

oil systems, flowmeters, gauges, etc. One plant of this type can deal with a large range of engines, whether air-cooled or liquid-cooled, and is equally suitable for production or research testing.

Modern wind tunnel plants are usually arranged for remote control. All measuring devices, controls, instruments, gauges, etc. are mounted conveniently to the testers; the control room is silenced, so that most of the testers' time is spent in a fresh, cool atmosphere free from exhaust fumes, heat and noise. Special precautions are taken to keep the control room free from petrol fumes, thus minimising fire risks. A view of the control panel of a typical research plant is given in fig. 3.

HEENAN HANGAR STAND.

The Heenan Hangar Stand is a simpler form of aircraft engine test plant than the Froude Wind Tunnel Plant described above, and is in extensive use for testing overhauled and repaired engines. In most cases the Cable-Suspended type is installed, as it provides a more flexible method of mounting the engine; in some installations, however, a rigid structure is employed, often with rubber mountings interposed between engine and structure. These Hangar Stands do not incorporate a dynamometer, reliance being placed upon a special 'test fan' or airscrew fitted on the engine shaft, to absorb a certain power at any given speed, and at the same time to provide an air blast for cooling air-cooled engines.

The accuracy of these Hangar Stands is not to be compared with the Froude Wind Tunnel Testing Plant, but it is often considered sufficient for dealing with engines which have been overhauled, as opposed to new engines tested in an engine factory.

THE CROSBY INDICATOR.

(Crosby Valve & Engineering Co., Ltd., Crosby Works, Ealing Road, Wembley.)

This indicator has been designed for use with either steam, gas or oil engines. Fig. 1 shows the Crosby Indicator as ordinarily made, having a drum 1¼ ins. in diameter, this being the correct size for high-speed work and answering equally well for low speeds.

The special feature in the indicator is the use of a double-coil spiral spring, very carefully calibrated; this form of spring enables the strains to which it is subjected to be transmitted from the centre of the piston through a ball-and-socket joint. Each spring is made of a single piece of wire wound from the middle into a double coil, the ends of which are screwed into a head with four wings having spirally drilled holes to receive and hold them securely in place. Adjustment is made by screwing the spring in or out of this head until it is exactly of the right strength, when it is firmly brazed.

Fig. 2 shows the Crosby New No. 2 Indicator, whilst fig. 3 shows a sectional view of the same indicator. This has the spring mounted outside the cylinder, where it is easily detached and replaced, and where it is removed from any effect of heat—a very necessary precaution when the indicator is used with superheated steam, or on gas engines. Another novel feature in this indicator is the shape of the piston. It is in form the central zone of a sphere, an arrangement which

practically eliminates cylinder friction. The piston is connected to the pencil mechanism by a rod terminating in a ball-and-socket joint in the centre of the piston. The position of the pencil lever on the drum can be altered by screwing the spring up or down and locking it in place.

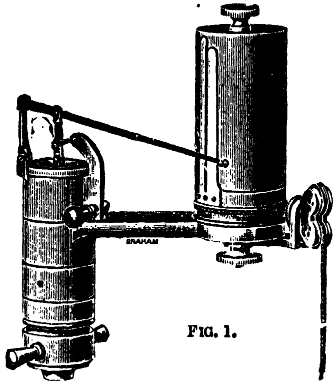


FIG. 1.

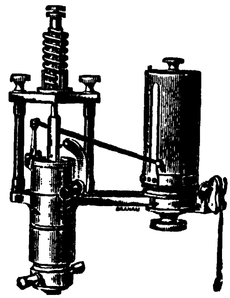


FIG. 2.

For the purpose of making the indicator of the utmost value to engineers, the Crosby New 'Continuous Drum' has been designed, and this apparatus makes it possible to obtain a series of diagrams on a roll of paper. The latest form has a novel arrangement, allowing the period

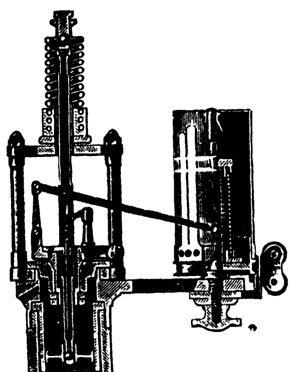


FIG. 3.

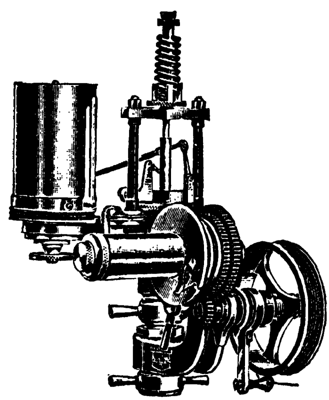


FIG. 4.

between each diagram to be altered at will; the diagrams can be taken close together or farther apart as desired, and the roll supplied is 12 ft. long. Such an apparatus is of great value in ascertaining variations of load in electrical tests, and in connection with rolling and winding engines.

Another useful device, which saves having a reducing apparatus added to an engine, is the Crosby 'Reducing Wheel.' This gives a perfectly correct reduction of the stroke of any engine to the stroke of the indicator drum, and it is connected directly to the indicator and mounted on the indicator cock. The arrangement is portable and can be used on any engine.

The latest pattern includes a detent device which enables the indicator to be stopped to put on a fresh card without disconnecting the reducing wheel from the engine crosshead, see fig. 4.

SECTION IV

Acoustics, Vibration and Noise.

Sound Abatement and Acoustical Treatment.

(Fibreglass Limited, Ravenhead, St. Helens, Lancs.)

The aim in modern buildings is, generally speaking, to omit almost everything that has been found unnecessary to hold the structure together and enable it to withstand the ravages of the weather, etc. Coinciding with this demand for minimum structural requirements there has been a great increase in the use of mechanical appliances, and this extended use of mechanism has forced us, more than ever before, to confront the problem of noise prevention.

The buildings being erected to-day offer little resistance to the transmission of sound from without, and the plasters and similar materials in use provide little absorption within.

The manufacturers of 'Fibreglass' offer a remedy in the form of a quilt (slabs and loose wool) and state that they have successfully developed their material to fill to some considerable extent the rôle previously occupied by solid structures.

They do not claim that the modern light structure, even when provided with insulation, offers the same resistance to sound transmission as that provided by the solid, massive structures of the older types of building. They do state, however, and point to the rapidly increasing use of their material in support of their claim, that 'Fibreglass' materials contribute largely to the 'comfortable' conditions experienced in the modern type of building where it is applied.

It is extensively used for floor insulation, and for this purpose is laid along the joists under the floor with battens between. Its function is to act as a cushion to take up sound vibrations, or, in other words, to break the continuity of hard rigid materials.

The quilt consists of long, flexible glass fibres encased in treated kraft paper or scrim cloth and stitched with thread to form a strong and durable quilt. 'Fibreglass' is chemically inactive, incombustible and odourless. It does not attract moisture and will not rot. Of great importance in building work is the fact that it offers no sustenance to vermin. Light and Medium Quilt is made up in rolls 1 yard wide by 27 yards long, and Thick Quilt in rolls 1 yard wide by 13½ yards long. Light Quilt is ¾ in. thick when uncompressed, Medium Quilt 1 in. thick when uncompressed, Thick Quilt 1½ in. thick when uncompressed.

'Fibreglass' has also been developed by the manufacturers for the acoustical treatment of speech halls, cinemas, studios, etc. For this purpose the material is made up in the form of panels consisting of a wooden framework enclosing 'Fibreglass' quilt faced with materials suitable to meet the requirements of interior decorations. For studio work it is supplied in the form of mattresses with 'Fibreglass' encased in wire mesh.

'Fibreglass' possesses very high sound absorption values and the following figures have been obtained from tests recently conducted by the National Physical Laboratory:

Reverberation absorption coefficients at various frequencies.

'Fibreglass' 1 in. thick in mats nailed on	250	500	1,000	2,000
battens 3 ins. by ¾ in. fixed to wall surface	0·75	0·90	0·90	0·95

SECTION IX.

PART II.

Roofing Materials—Painting.

'ROK' ROOFING.

(*D. Anderson & Son, Ltd., Stretford, Manchester.*)

'Rok' roofing forms a permanent and easily applied roof covering. Only the highest grade of felt fibre is used in its manufacture. This is saturated with an elastic water-proofing compound which does not dry out or evaporate in any climate, as it contains no oils, or volatile matter. The coating on the surface is a permanent one, composed of a natural bitumen of very high melting point. There is nothing of an organic nature in either the saturating or coating compound. It is acid and alkali proof and not affected by white ants. It is suitable for pitched, flat, or circular ('Belfast') roofs, and for reinforced concrete roofs it will be found to make a thoroughly water-tight job.

Rain water from roofs covered with it can be used for domestic purposes.

Being an excellent non-conductor, it ensures an even temperature, and is therefore an ideal roofing for hot climates.

It is laid on close jointed boarding, which should not be less than ¾ in. thick, either plain edged or tongued and grooved. The purlins or rafters should be so spaced that there is no appreciable spring in the boarding.

It may be fixed either longitudinally from gable to gable, or vertically from eaves to eaves. It is, however, considered advisable to lay it crosswise to the boarding. The joints should be formed with at least 2 ins. overlap, being stuck together with 'Rok' cement and carefully nailed with tacks not more than 2 ins. apart placed in the centre of the seams. The heads of the tacks where exposed and the outside of the seams should be dressed with 'Rok' cement.

'Rok' roofing is supplied in rolls of 72 ft. by 3 ft. wide, in the following grades:—

½-ply average weight,	50 lb.	per roll	2-ply average weight,	80 lb.	per roll
1-ply „	60	„	3-ply „	100	„

(The above weights do not include the nails and cement.)

The prices including the supply of special galvanised broad-headed nails and liquid cement or fixing will be quoted by manufacturers on application.

Each roll, if carefully laid, should cover 200 sq. ft.

The manufacturers will undertake to carry out the fixing if desired.

'Rok' roofing may also be used on concrete, fixed with special 'Rok' mastic, and it is advisable to write the manufacturers for full directions as to the method of applying.

It may also be used for covering flats and gutters. In this case it is recommended to use two layers, the first layer being tacked to the boarding, and the top layer stuck to the under one with 'Rok' mastic. In this way no nail heads are exposed. Flashings to walls, round lights, etc., may be made in the material.

'FLEXTILE' ROOFING.

'Flextile' Roofing is a new variety of roofing specially produced for use on dwelling-houses and other buildings where a more attractive appearance is desired than that of the usual bitumen roofings. It is made of similar materials to 'Rok,' and is of the same high quality in every respect, but the surface is covered with crushed minerals, in either red or green natural unfading colours. These minerals are firmly embedded in the surface of this roofing, which is quite as attractive in appearance as are slates or tiles, with the advantage that there is no possibility of breaking or cracking, and that no wind or water can ever penetrate to the interior. 'Flextile' Roofing is, in fact, storm-proof.

Supplied in rolls 12 yds. by 1 yd. sufficient to cover 100 sq. ft. Weight 80 lb. per 100 sq. ft., *i.e.* about ⅘ lb. per sq. ft. Price including nails and cement, red or green, quoted on application.

RUBEROID ROOFING.

(*The Ruberoid Co. Ltd.*, 28 *Commonwealth House*, 1-19 *New Oxford St.*, *W.C.* 1.)

Ruberoid is a bituminous roofing manufactured in sheet form and suitable for use on flat-pitched or curved roofs of wood or concrete, and has been so employed extensively for fifty years.

Complete roofing contracts are carried out by the firm's Contract Departments in any part of the British Isles and, through their agents, in any part of the world.

Ruberoid is light, flexible, waterproof, resistant to the effects of acid and alkaline fumes and sea air, and unaffected by extremes of heat and cold. It has excellent heat insulating properties and is consequently very suitable for use in tropical countries.

Standard Ruberoid roofing is made in three weights, and is grey. Where colour is required Ruberoid Red Roofing can be employed; or, alternatively, Ruberoid Mineral-Surfaced Roofing, which has a surface of natural crushed slate. It is supplied in three colours: Thatched Brown, Westmorland Slate Green and Natural Bangor Steel Blue.

Where an extra high degree of fire resistance is required, Ruberoid Astos Roofing, with an asbestos base, is available.

Ruberoid roofs are of two main types:—

(1) Roofs consisting of a single layer of Ruberoid.

(2) Built-up roofs consisting of two or more layers of Ruberoid underlay bedded together on the site with a finishing surface appropriate to the service required on the roof.

For small roofs consisting of a single layer of Ruberoid which can be fixed easily by unskilled labour, Ruberoid is supplied in rolls 36 in. wide, complete with accessories.

Ruberoid Built-up Roofs can be laid with several alternative finishes:—

(*a*) Standard Ruberoid.
(*b*) Ruberoid Astos Asbestos Roofing
(*c*) Mineral-surfaced Ruberoid.
(*d*) Gravel.
(*e*) Ruberdal tiles or Ruco Ruberoid.

Ruberdal is a Ruberoid Built-up roof finished with tiles. It is particularly suitable for verandahs and terraces subjected to foot or wheeled traffic.

Ruco Ruberoid is a Ruberoid Built-up roof with a surface of Ruco Mastic Asphalt and is suitable for foot and wheeled traffic, chairs, etc.

The Ruberoid steel roof deck is a rigid covering equally suitable for flat or sloping roofs and can be employed over purlins or joists spaced at maximum centres of 8 ft. The steel units are light and the complete roof only weighs 4½ lb. per square foot.

Insulation is included in this system of roofing to reduce transference of heat. The insulating material is usually provided in a half-inch layer of building board between the steel deck units and the weatherproofing. In this position it is isolated from moisture and can, therefore, function at its full efficiency. The weatherproofing usually consists of a single layer of Ruberoid bonded to the insulation on pitched roofs, while on flats the Ruberoid built-up system of two layers is employed.

Ruberoid steel roof deck can be accommodated to meet any spacing of joists or purlins up to 8 ft. No definite or exact spacing of supporting members is necessary.

SECTION X

Mortars—Concretes—Cements—Plasters—Placing Concrete.

THE CROCKATT SIMPLEX POLISHER FOR TERRAZZO FLOORS.

(*W. Crockatt & Sons, Ltd.*, 62 *Darnley Street, Glasgow, S.* 1.)

To meet the demand for increased speed of polishing Terrazzo, Mosaic, 'Cullamix' and similar floors, the Crockatt Major Double Disc polisher has been produced.

The latest electric model is illustrated and, as will be seen, has a direct worm drive from the motor to the two polishing spindles.

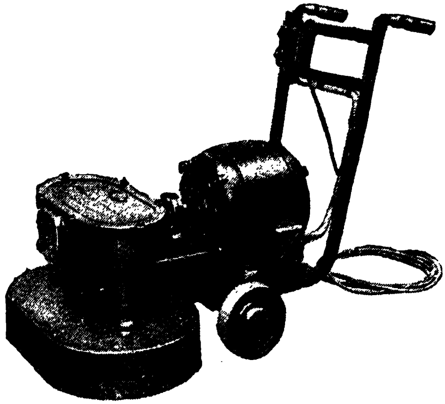
Each spindle carries three polishing blocks, flexibly mounted and arranged on a clover leaf form so that the two polishing heads intermesh while revolving. The result is that the machine is particularly easy to operate, as the action of the two heads make it practically self-propelling along the floor.

The machine polishes a strip 22 ins. wide, and will give any desired finish, depending on the grit and grade of blocks used.

It is, we are informed, particularly successful on 'Cullamix,' which is a very hard material and now largely used for swimming pools, as well as ordinary flooring work.

The machine is fitted with ball bearings throughout; the gearing runs in an oil bath and lubrication of the spindles, etc., is by grease gun.

An efficient splash guard is fitted and is easily removable so as to give access to the polishing heads. The polishing blocks are standard size and can be supplied in a variety of grits and grades to suit various materials.



The motor fitted as standard is single-phase, direct starting, but three-phase or direct current can be fitted if required.

If preferred, a petrol engine can be fitted so that the polisher can be used in buildings before the electric supply is laid on or is otherwise not available.

The machine, as well as the other models made, can be fitted with a flexible shaft and wall polishing attachment which is very simply brought into use when required.

Where space is limited—as for example on landings, vestibules, bathrooms, etc., the appropriate machine is the Minor Double Disc Type, which is very compact, light and easily handled, and can be operated from any electric light plug or socket.

Should it be preferred to use pre-cast tiles, the same firm makes polishing machines for these, the polishing being done on the same principle as already described.

These machines, known as the Jenny Lind pattern, are also used for polishing marble slabs, which can be cut into suitable pieces on the carborundum marble sawing machines made by Messrs. Crockatt. The polishing machines are arranged for wall mounting and for direct electric or belt drive.

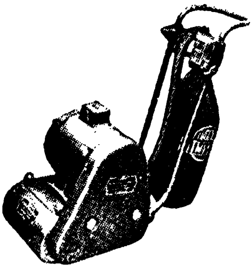
The Sawing Machines are also in use for sawing fireclay goods, such as sinks, etc., where a modification of standard sizes is required. They are also employed in the cutting of wall or floor tiles to special shapes and sizes.

THE CROCKATT-SIMPLEX FLOOR SANDING MACHINE.

(W. Crockatt & Sons, Ltd., 64 Darnley Street, Glasgow, S.1.)

The base of this sander is made of aluminium as are all the components except where they must be of iron or steel because of wear.

The sander has an aluminium drum which is covered with sponge rubber and has a simple method of securing the sand-paper. The length of the drum is 10 in. and the diameter 6¾ in. The drum pressure can be adjusted gradually from a very heavy cut to a light skim, this adjustment being by means of a lever close to the tubular handle. The switch is conveniently placed on the



handle and is so mounted that it cannot be accidentally switched on, that is to say, the switch must be pulled up to put it on, and not pushed down, the latter arrangement being the cause of occasional accidents.

The motor is 1½ h.p. single-phase 230 volts 50 cycles, as standard, and gives a drum speed of 1,500 r.p.m., but three phase or direct current motors can be supplied. Full particulars of current supply are necessary. As electric power is not always available, an air-cooled petrol engine can be fitted, this being easily replaced by an electric motor if desired.

The drive is by means of Vee belts. The reason for adopting Vee belts rather than chain, is that it is found by experience that the chain is apt to cause markings on the floor, particularly when this has to be highly polished.

The fan is mounted in the base as a complete unit and gives a very effective suction, the air and dust being discharged into an easily removable bag mounted at the end of a copper pipe.

Ball bearings are fitted wherever practicable.

The machine is a first-class job as to efficiency, workmanship and appearance.

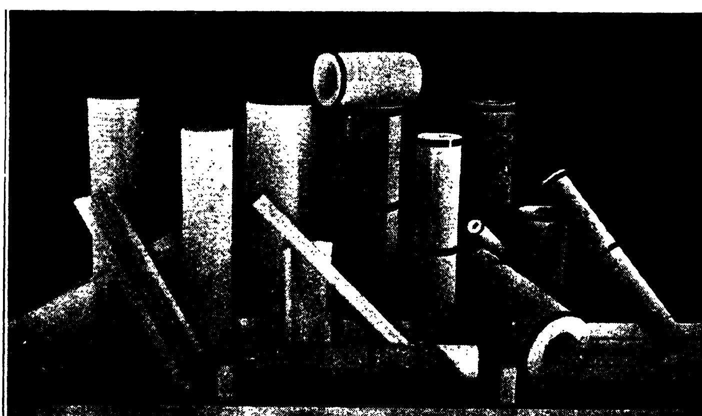
SECTION XI

Clays and Allied Products used by Engineers.

Heat Insulating Materials.

(The Chemical and Insulating Co., Ltd., Darlington.)

Darlington 85 per cent. Magnesia Coverings.—Among modern high grade boiler coverings those composed of magnesia take a leading place. The coverings, sheets, slabs and bricks manufactured by the Chemical and Insulating Co., Ltd., of Darlington, are composed of 85 per cent. of pure carbonate of magnesia which is manufactured by the firm by the Pattinson process. While the micro-cellular structure of magnesia forms an ideal insulating substance, further strength is obtained by the intermixture with it of 15 per cent. of best quality asbestos fibre, which binds the magnesia and gives a resultant compound possessing unusual strength and tenacity. The Darlington works of the firm are a modern factory which have been laid out for the production of magnesia coverings in all forms, and are equipped with the latest specially designed machinery for such work. Where large heated surfaces are to be insulated, skilled application of coverings is required for best results and the company undertakes the application of its materials in most countries of the world. The insulation of steam piping, etc., may readily be undertaken however with unskilled labour by the use of Darlington 85 per cent. Magnesia Moulded Sectional Covering, which is supplied in standard thicknesses of 1 in., 1½ in., and 2 ins., and for pipes from ½-in. to 8-in. bore. This covering can be supplied with an outer wrapping of canvas, asbestos paper or roofing felt, together with neat steel strips, one per foot length of covering, for securing the insulation in place.



For insulation of boilers, engine-room casings, drying cupboards, turbines and sundry other applications, Darlington 85 per cent. Magnesia Slabs are supplied in standard sizes of 36 ins. by 6 ins., by any thickness from 1 in. to 4 ins. Other sizes can also be supplied. Where semi-portable insulation is required these slabs are wrapped in asbestos paper.

For insulation of irregular surfaces and for general use, Darlington 85 per cent. Magnesia Plastic is available. This is a powder, delivered in half-hundredweight bags, which is supplied ready for mixing with clear water to the consistency of mortar, for application to the surface to be insulated, which must, however, be heated. One ton of Darlington 85 per cent. Magnesia Plastic will cover about 2,400 sq. ft. surface when applied to a thickness of 1 in.

Dextramite' High Temperature Coverings.—For temperatures above 600° F. the firm specially recommends 'Dextramite' high temperature coverings, which it has been shown by National Physical Laboratory tests will not char or disintegrate when subjected to temperatures up to about 1,600° C. They are manufactured in forms generally similar to the Darlington 85 per cent. magnesia coverings, and also in brick form for insulating blast furnaces, gas mains, retorts and boiler casings, etc. The bricks have a crushing stress of 3½ tons per sq. ft.

The illustration on p. 1102 shows a variety of the products of the firm, which is in a position to advise and to supply expert information and materials for all classes of insulation work.

'FIBREGLASS' FOR HEAT INSULATION.

(Fibreglass Limited, Ravenhead, St. Helens, Lancs.)

'Fibreglass,' although a comparatively new insulating material, is rapidly being adopted over the complete steam pressure and temperature ranges in modern engineering practice.

The material is stated to consist entirely of glass of a high melting point; it withstands high temperatures without exhibiting any defects.

'Fibreglass' for heat insulation is made up in a variety of forms and consists essentially of a mass of very fine glass fibres lying closely together and enclosing within their structure millions of minute air cells, which, aided by the characteristic sheen of the silken fibres, offer a determined resistance to the passage of heat.

Very high insulating efficiencies are claimed by the manufacturers, and the following are figures determined by the National Physical Laboratory:

	Insulating Efficiency.		
	At 50 lb. per sq. in. gauge.	At 100 lb. per sq. in. gauge.	At 200 lb. per sq. in. gauge.
'Fibreglass' 1 in. thick	87·9 per cent.	88·9 per cent.	88·9 per cent.

The following are the heat losses based on the N.P.L. figures from a 4½-in. O.D. steam pipe with steam at 370° F. and an atmospheric temperature of 70° F.:

	Loss. B.Th.U.'s per sq. ft. of pipe surface. Per hour.
'Fibreglass' 0·5 in. thick	174
" 1·0 " "	110
" 1·5 ins. "	84
" 2·0 " "	69
" 2·5 " "	60
" 3·0 " "	53

'Fibreglass' for heat insulation is made up in the form of strips for convenient winding round steam pipes and in the form of rigid sections. It can also be provided in sheets or mattresses for placing on boilers or other large surfaces. The outer finish may be sheet steel, hard-setting cement, asbestos cloth, canvas or waterproof felt, depending on requirements.

It will be readily understood that the material, being composed entirely of the chemically inert substance—glass—is extremely durable. Durability tests over a period of eight years are reported by the manufacturers. The tests have been conducted on small coasting vessels where the insulation has been more often awash than otherwise. Notwithstanding the excessive treatment, it is stated that the material after ten years continues to give high insulating values and shows no signs of deterioration. The long fibres of which it is made up cling together with remarkable tenacity. It would appear that this is due partly to the semi-woven nature of the threads and partly to the high coefficient of the friction between glass surfaces. It is this property which provides 'Fibreglass' with its remarkable resistance to vibration and other influences tending to cause disintegration.

The material is, of course, quite incombustible, and of particular interest is the fact that it offers absolutely no sustenance to vermin.

The manufacturers guarantee the material up to 900° F., but state that a series of tests have shown that it will withstand temperatures considerably higher, and is, in fact, in use on installations at 1150° F. without showing the slightest sign of disintegration or fusion.

SECTION XIII

Earthwork — Excavation — Foundations — Piling — Dams—Scaffolding — Stagings — Shoring - Retaining Walls — Chimneys — Stone and Brick Arches—Piers.

STEEL TUBULAR SCAFFOLDING.

(Burton's Patent.)

(The London & Midland Steel Scaffolding Co., Ltd., Burwood House, Caxton Street, London, S.W. 1.)

Tubular steel scaffolding has now almost entirely superseded the old wooden poles. The advantages of steel are outstanding as compared with timber. From a structural point of view it is safer; the depreciation is negligible, risk of fire is eliminated, and the cost of erection in the saving of time and men is very considerable.

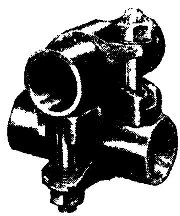


FIG. 1.

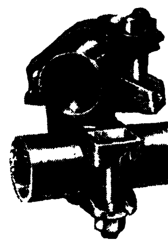


FIG. 2.

BURTON'S PATENT DOUBLE COUPLER (Fig. 1) is used for coupling upright tubes to horizontal tubes. Its features are great strength and load carrying capacity, and the speed with which it can be securely fixed or released. It is a one-piece coupler, no loose parts to get lost. The erection of scaffolding with this 'Quickgrip' coupler is simplicity itself.

BURTON'S PATENT SWIVEL COUPLER (Fig. 2) for coupling bracing tubes to strengthen scaffold.

BURTON'S PATENT PUTLOG COUPLER (Fig. 3). This coupler for fixing putlogs or transoms to horizontal tubes is very simple to secure. There are no projections interfering with the wood scaffolding boards and all wobbling or rocking is eliminated.

BURTON'S PATENT SPLIT JOINT PIN (Fig. 4). Half a turn of this bolt firmly secures two tubes together end to end.

BURTON'S BALL BEARING STEEL CASTORS are specially made for all types of mobile scaffolds. They are fitted with solid steel wheels or with solid rubber tyres and with or without a screw jack.

All the foregoing fittings are specially made for use with standard 1½ in. internal, 1$\frac{29}{32}$ in. external diameter tube. They are also made in 1 in. and 3 in.

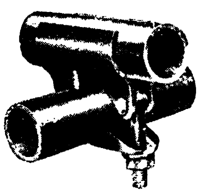


FIG. 3.

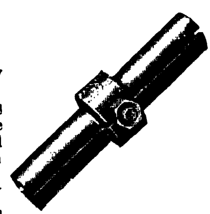


FIG. 4.

In addition to scaffolding the Burton's patent system lends itself to manifold uses, such as making heavy and light racks for storage purposes, temporary bridges, framework for huts, stands and launching platforms, diving stages and playing field equipment; in fact, it has supplanted the use of timber in many notable ways.

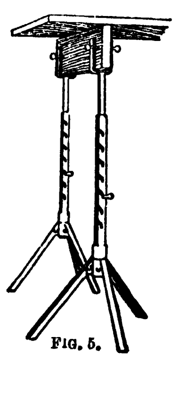


FIG. 5.

STEEL PROPS OR SHORES.

(Burton's Patents.)

These are used for centring floors, shuttering, etc., and have a positive adjustment by a very strong square thread pressure screw. No spanners or jacks are needed.

BURTON'S PATENT SPLIT HEADS

(Fig. 5). 'The Little Gem' Revolving Fork Head for interior scaffolds, for plasterers and decorators. To take one or two boards, with folding legs, or can be supplied with shoes for outside work on soft ground. No rachets or pegs to engage or bolts tighten, and positive adjustment not dependent on friction. Supplied in four sizes: 1 ft. 3 in. to 1 ft. 9 in. to reach 7 ft. to 8 ft; 2 ft. to 3 ft. to reach 8 ft. to 10 ft; 3 ft. 6 in. to 5 ft. 6 in. to reach 9 ft. to 12 ft.; 5 ft. 9 in. to 9 ft. to reach 12 ft. to 15 ft.

SECTION XVIII

PART VI

Water Supply.

Automatic Control Equipment for Water Softening Plant and Filters.

(*Filtrators, Ltd.*, 92 *Seymour Place, London, W.* 1.)

A typical example of the effectiveness of the automatic control of water softening plant is the complete control of the regeneration by brine of the zeolite bed of a base exchange water softener. In a plant of this type the only manual operation which has to be carried out in connection with its normal running is the replenishment of salt in the saturator at approximately 10-day intervals. In addition to the incorporation of this gear in new installations, existing manually controlled plants can be readily modernised by the fitting of automatic control equipment. The control system, which is extremely simple in respect of both lay-out and individual components, consists of an automatic control unit mounted on or near the softening or filtration plant. Each element of this unit is connected to its respective hydro-valve or operating ram by a $\frac{3}{16}$-in. copper pipe, generally as shown in figs. 1 and 2.

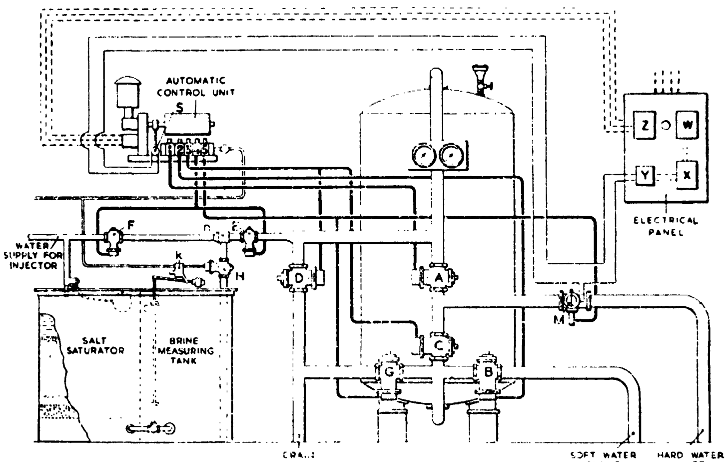


FIG. 1.—Base Exchange Softener Control System.

Hydro-Valve Key.

A. Hard water inlet; B. Soft water outlet; C. Backflush inlet; D. Washout; E. Brine inlet F. Water inlet to injector; G. Drain; H. Brine suction; M. Water meter and resetting mechanism; k. Brine float control valve; n. Injector. *Electrical panel components:* W. Fuse switch for rectifier; X. Rectifier; Y. D.C. fuse switch; Z. Fuse switch for motor.

▬▬ Controlled power water.	——— 80 volt D.C. circuit.
- - - Uncontrolled power water.	- - - - 230/40 volt mains supply.

Automatic Control Units.—Two main types of controller are in service, the smaller being used mainly for the automatic control of filters and single-unit softeners, whilst the larger is used for large softeners or multiple unit filters and softeners. Basically, the controller consists of a revolving drum directly driven by means of a fractional h.p. electric motor through appropriate gearing. The drum, which is fitted with a series of adjustable cams makes one revolution in the time required or regenerating a softener, for performing any other complete process. The regeneration period for a typical softening operation occupies approximately one hour after each

complete softening cycle. Immediately below the cam drum is a relay block containing a number of relay valves, each being operated by the appropriate drum cam through the medium of a valve plunger. Additionally, any relay valve can be operated independently by hand control, if required, through the medium of a hand lever and the valve plunger.

Hydraulically Operated Valves.—The hydro-valves which are an essential feature of the automatic control system are well known for their smooth working and the long life which is obtainable from their seatings. As a result of the smooth working of these non-shock valves, water hammer in the pipes is entirely non-existent. For the automatic operation of gate valves, when these are incorporated in a system, servo action is provided by means of hydraulically operated double-acting rams. As a general rule pipes up to 3-in. diameter are fitted with hydro-valves, whilst pipes of 4-in. diameter and over have ram operated gate valves. If required for any special reason,

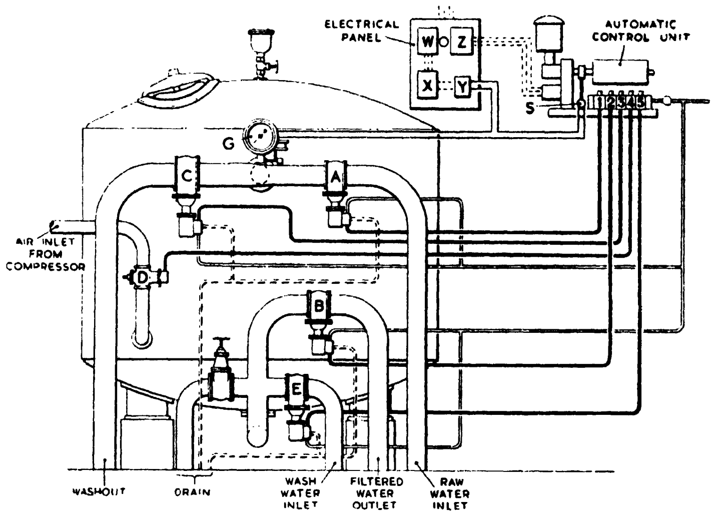


FIG. 2.—Filter Control System.

Key.

A. Raw water inlet gate valve; B. Filtered water outlet gate valve; C. Washout gate valve; D. Air inlet hydro-valve; E. Wash water inlet gate valve; G. Differential pressure gauge; S. Solenoid switch; W. Fuse switch for rectifier; X. Rectifier; Y. D.C. fuse switch; Z. Fuse switch for motor.

▬▬ Controlled power water. ——— 80 volt D.C. circuit.

······ Uncontrolled power water. - - - - 320/40 volt mains supply.

however, ram operated valves can be fitted to pipes down to 2-in. diameter. Apart from the electricity supply for operation of the motor the only other power required for automatic control is water pressure at 25 to 30 lb. per sq. in. Where, however, an air supply is more readily available than water pressure, this is equally suitable for use as the operating medium.

Typical Systems.—Two typical automatic control systems are shown diagrammatically in figs. 1 and 2, the former showing a base exchange softener system and the latter a pressure filter system. In the case of the softener system the controller is started by an electrical contact in the water meter M, making the circuit from the rectifier X in the electrical panel to the solenoid S on the automatic control unit. The energising of this solenoid causes it to operate a mercury switch which closes the main circuit to the control unit motor. Rotation of the control unit drum, which throughout the softening cycle had been stationary holding inlet and outlet hydro-valves A and B open, then commences and the regeneration process is in progress. The complete and correct sequence of shutting down the softener, backflushing, brine injection, rinsing and recommencing the softening operation is then carried out accurately and without any attention

whatsoever. The backwashing operations for the filter system, fig. 2, are carried out in a similar manner. In this case, however, the only hydro-valve incorporated is used to control the air inlet, all the other valves being controlled by double-acting rams. The making of the 80-volt circuit to the automatic controller is effected by a contact in the differential pressure gauge, G, which measures the loss of head through the bed of filter media. The development and supply of this simple automatic control system by Filtrators, Ltd., has the effect of providing an efficient method of control for water treatment plant. The firm is able to supply technical booklets which describe the equipment.

(*Reprinted from ' Water and Water Engineering '*).

In addition to the above plant, Filtrators Ltd. manufacture Filtrol base exchange softening plant, lime soda softening plant, using their special stabilisation process and acceleration methods, pressure and rapid gravity filtration plant, iron and manganese removal plant, water de-ionisation plant, chlorinating equipment, effluent treatment plant, condensate de-oiling plant (electrical process), continuous blowdown and heat recovery plant, and all types of dosing gear for dealing with every water treatment problem.

Waste Water Detection.

(*The Palatine Engineering Co., Ltd., Hawthorne Road, Bootle, nr. Liverpool.*)

The detection of invisible leakages between the reservoir and the consumer is usually done by meters of the ' Deacon ' type, similar to fig. 1. A large recording meter is fixed on the main

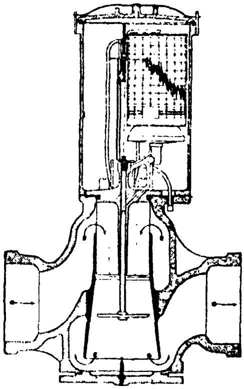


Fig. 1.

at the reservoir. The supplied area is divided into districts of about 2,000 inhabitants and arranged so that the whole supply comes from one direction, and passes through a Deacon meter. Fig. 2 is a plan of the area showing valves.

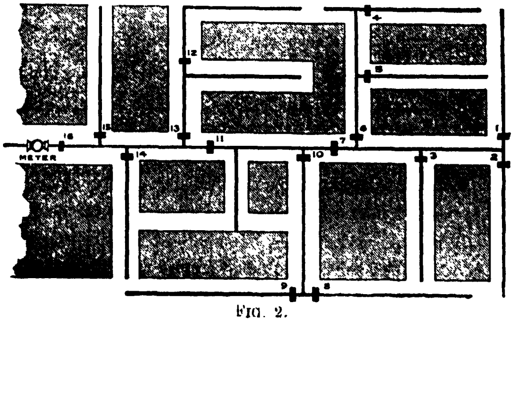


Fig. 2.

A typical chart for twenty-four hours in an uncontrolled district is shown in fig. 3. The minimum flow during the night is 2,000 gallons per hour (represented by the shaded portion), or 48,000 gallons per day. In the absence of night supplies for trade purposes, this water is leaking through defects. The water used (represented by the cross-hatched portion) is 29,775 gallons making a total supply of 77,775 gallons per day.

To locate this waste a chart is fitted in which the time scale is multiplied by four. During the night the inspector proceeds to the end of the district and closes No. 1 valve, and the other valves in succession, at about 15-minute intervals, noting the time and the location of the valve

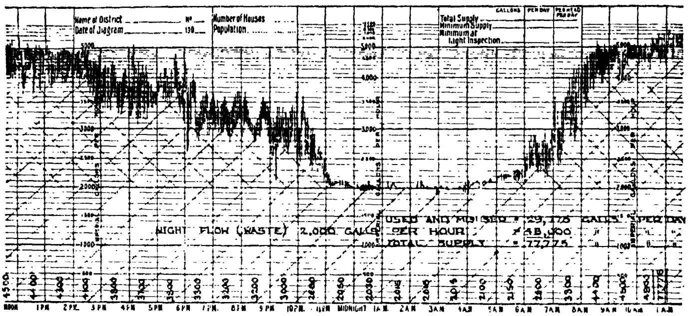


FIG. 3.

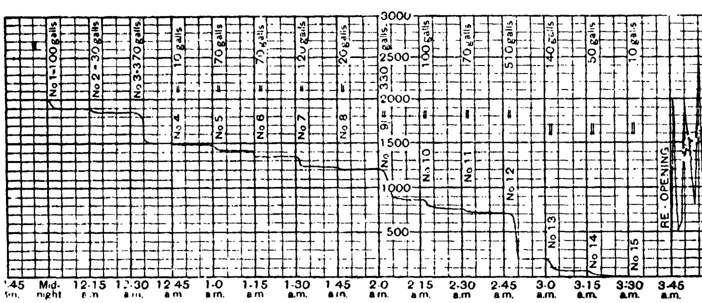


FIG. 4.

The result is shown in fig. 4, the rate of flow in each portion before isolating being clearly recorded. There is an indication of large defects in portions supplied through valves Nos. 3, 9, and 12. These are afterwards traced by a form of stethoscope and examination of fittings. In some of the areas where this system is carried out the consumption, in gallons, per head, per day, is:—Bath, 21; Birkenhead, 20·75; Bristol, 24·12; Cambridge, 23·2; Chesterfield, 16·0; Weston-Super-Mare, 22·1; Halifax, 17·74; South Staffs Waterworks Company, 15·13; Wakefield, 19·68. Trade requirements are measured through separate meters and are not included in these figures.

Filtration.

THE 'PULSOMETER' FILTER.

(The Pulsometer Engineering Co., Ltd., Nine Elms Iron Works, Reading.)

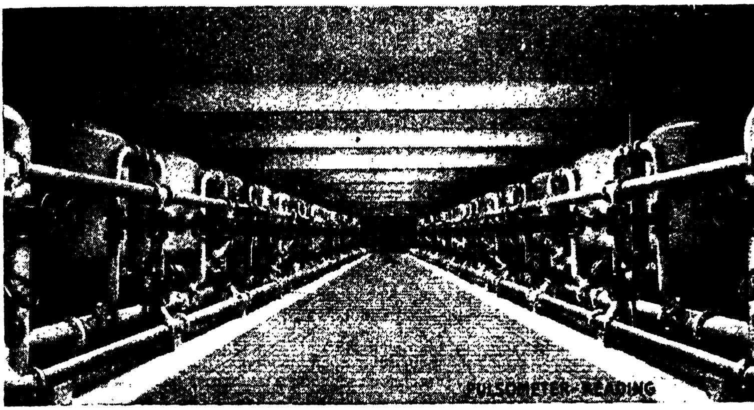
Filtration is effected by passing water under pressure through special carefully graded filtering media.

Arrangements are made so that cleansing can be carried out thoroughly and systematically.

Dual purpose nozzles fitted on the collecting system in the filter offer very little frictional resistance during filtration; a factor which contributes largely to low running costs.

The cleansing process is effected by compressed air forced back through the nozzles and agitating the bed. A reverse flow of water then carries all dirt in the filter over a skimming tundish and then to waste. The design of the nozzles is such that it is impossible to carry on the process of agitating and washing simultaneously, without mutual interference ; a low wash water consumption thereby being obtained.

An alternative method of agitating the filter bed is by use of steam.



Where steam at 40 lb. per sq. in. is not available for entraining air for filter bed cleansing, compressed air is usually provided by means of one of their ' Hydrair ' Motorless Air Compressors. This operates hydraulically and the risk of mechanical breakdown of moving parts or contamination of the water by leaking oil, as from a mechanically operated compressor, is therefore eliminated.

Automatic chemical dosing gear is installed for coagulation *p*H recording and control.

The illustration shows *one* bay of a recent pressure purification plant arranged to treat between six and eight million gallons of peaty water daily, producing a filtrate which fulfills all the requirements of a high grade drinking water for a town's supply.

The filters can be built to withstand any pressure and are made vertical or horizontal to suit accommodation and other conditions on site.

SECTION XIX

PART II

Hydraulic Transmission of Power.

Variable Speed Drive.

(Keelavite Rotary Pumps and Motors Ltd., Allesley, Coventry.)

This is a positive hydraulic drive in which an oil pump is operated by the prime mover and an oil motor transmits the power to the driven mechanism. By varying the respective capacities per revolution of the pump and/or hydraulic motor, complete control is obtained of both the speed and output torque of the hydraulic motor.

The pump and motor units are similar in construction and are in fact frequently identical. Where the oil motor is required to operate at approximately the same maximum torque throughout its speed range, it can be of fixed capacity; speed control from zero to maximum is then obtained by varying the capacity of the pump. If, on the other hand, it be required to exert a maximum horsepower which is constant over a substantial portion of the total speed range, so that as the speed falls the output torque must rise, then it is necessary to vary the capacity of the hydraulic motor over the corresponding portion of the speed range while keeping the capacity of the pump fixed. By means of a combined control of the capacities of pump and motor, any required relationship may be obtained between the speed and the available torque.

The Keelavite unit illustrated (fig. 1, p. 1111) is a standard model suitable for working pressures up to 1,000 lbs. per sq. in. The flow of oil from the inlet to the outlet port is controlled by the volume swept in the working chamber (1), by the blades of the rotor (2). This is driven by or drives the rotor housing (3), according to whether the unit is functioning as a pump or as a motor; the blades are a sliding fit in the slots in the rotor housing, and the capacity of the unit per revolution is proportional to the length of blade projecting from the rotor housing. The rotor is located axially by means of the locating rotor bearing (4), so that when the non-rotating rotor centre (5) is moved axially by means of the rack and pinion capacity control (6) the blades are carried with it, and so are enabled to maintain a fine clearance sealing fit over their end faces with the face (7) of the rotor centre, which constitutes one end wall of the working chamber. The rotor and the rotor housing both receive radial location from the non-locating rotor housing bearing (8), through which the rotor slides when the capacity is varied, while the rotor housing is located by the locating rotor housing bearing (9) so that its face, which constitutes the other fixed end wall of the working chamber, is flush with the face of the abutment end plate (10). This completes the circumferential seal round the outside of the rotor housing, and also maintains a face seal with one end face of the abutment (11), the other end face of which seals with the abutment bearing housing (12). The circumference of the abutment seals with the abutment bore in the casing, and also with the arc AA of the rotor centre. Owing to the fact that this arc AA is longer than the arc BB limiting the abutment recesses, it will be seen that at all times the abutment maintains a seal between the inlet and outlet ports. The abutment recesses enable the blades to pass from the outlet to the inlet ports when not pumping, the rotor and abutment being timed together by the timing gears (13), (14), so as to permit the free passage of the blades through the recesses. Owing to the fact that the blades completely enter the abutment recesses before the abutment commences to seal again at the end of the arc AA, adjacent to the delivery port, the blades are able to displace exactly their own volume of oil out of the abutment recesses, so that there is no net change in displacement on the delivery side of the pump. As a result, the output is entirely free from pulsation, a feature the importance of which cannot be over-estimated.

Radial hydraulic balance is ensured in the case of the rotor by the fact that the leak paths on its inside and outside diameters are similar, while in the case of the abutment the patented Keelavite system of balancing recesses and passages ensures that there is a perfect symmetry of the radial pressure forces. Axial balance of the abutment is ensured because it is located with equal clearance on either face, while the rotor is kept in axial balance by maintaining the casing pressure by means of an automatic valve at the mean of the inlet and outlet pressures. As a result, there are no loads at all on the abutment bearings, and only the torque reaction load on the rotor bearings.

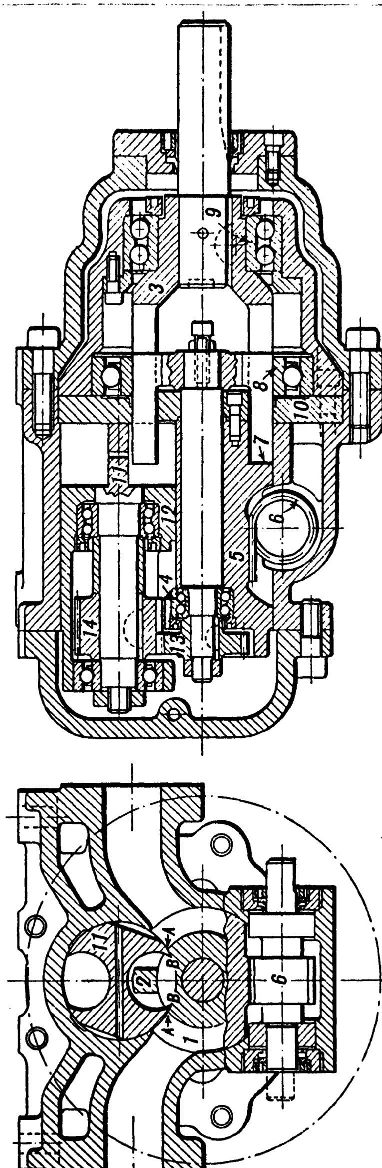


FIG. 1.—Arrangement of Keelavite Variable Capacity Pump.

In large units where even the torque reaction constitutes an undesirable bearing load, a twin abutment design is utilised with four ports, two inlet and two outlet, ensuring a complete symmetry of pressure conditions around the rotor; as a result, there are no bearing loads at all on the rotor bearings, the rotor itself being in pure torsion, as in the case of an electric motor.

Typical performance curves are shown in figs. 2 and 3 taken from standard units of 5 cub. ins. capacity. These indicate very clearly the very high mechanical efficiencies attained with Keelavite pumps and motors. In addition, the high starting torque and very flat characteristic over a wide speed range should be noted. As a general guide it can be stated that power efficiencies of the order of 85 to 90 % per unit can be relied upon under reasonable conditions at maximum power without need for the units to be of uneconomic size. In the case of rotary drives, the maximum pressures consistent with this order of efficiency range from 400 to 1,500 lb. per sq. in.,

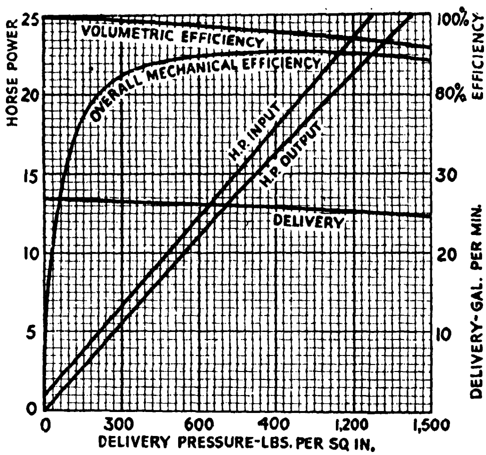


FIG. 2.—Performance Curves of Keelavite Pump.

Capacity: 5·0 cub. ins. per revolution.
Oil Viscosity: 150 secs. Redwood No. 1.

increasing with the horse power under consideration. When a hydraulic pump only is required to operate a receiver of high volumetric efficiency (such as a hydraulic press cylinder) even the smaller units can be operated at pressures up to 1,500 lb. per sq. in., and in certain cases up to 2,250 lb. per sq. in. without undue sacrifice in efficiency.

Owing to the fact that there is an absolute absence of frictional contact in the mechanism of the Keelavite units, their working life, provided that the oil is kept clean, is dependent only upon that of the ball bearings employed. These are kept well within their catalogue rating, and accordingly it can be stated that the life of the units is of indefinite duration.

In addition to pumps and rotary hydraulic motors, Keelavite standard equipment includes a comprehensive range of valve gear, which is intended for use in connection with rotary drives and also for use in conjunction with the pumps to control various types of reciprocating movements in which hydraulic rams are employed in place of rotary motors.

Thus, piston type directional valves are available, which for example may be interposed between a pump and rotary hydraulic motor in order to obtain reversal of the direction of rotation of the hydraulic motor. This type of valve is manufactured so that when required it can also be used to stop the motor, in which case it has three control positions corresponding to forward, neutral, and reverse; while the valve itself may be either directly, manually, or mechanically controlled, or may be controlled by a hydraulic pilot valve which is in turn suitable for solenoid operation where electrical control is desirable.

In order to avoid overloading on hydraulic circuits when sudden changes in speed or load take place, or when starting, stopping, or reversing, the Keelavite piston controlled balanced poppet type relief valve has been specially designed to secure instantaneous response together

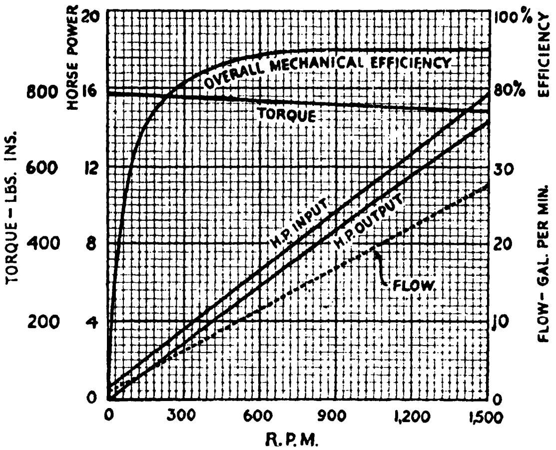


FIG. 3.—Performance Curves of Keelavite Motor.

Capacity: 5·0 cub. ins. per revolution.
Oil viscosity: 150 secs. Redwood No. 1.
Input pressure: 1,000 lbs. per sq. in.

with a quiet performance even when passing large flows. This valve has the further advantage that the pressure setting is independent of the rate of flow through the valve so that peak loads can be accurately controlled under all conditions; while in addition it is suitable for operating as a remote controlled by-passing valve when required.

In addition the Keelavite range of valve gear comprises various types of flow control valves arranged to give accurately controlled flows independent of the operating pressure in question from any suitable source of pressure fluid, such as a fixed capacity pump or hydraulic accumulator. Such flow control equipment is of the greatest value where it is required to obtain consistent speed control over a wide range, particularly at low operating speeds of hydraulic rams or motors.

THE 'VSG' MARK III. VARIABLE DELIVERY PRESSURE PUMP.

(*Vickers-Armstrongs Ltd.* (*Variable Speed Gear Dept.*), *Vickers House, Westminster, S.W. 1.*)

This pump is of the 'Williams-Janney' or horizontal piston type. Fig. 1 (p. 1114) shows a vertical section through the latest design (arranged for lever control).

The 'mainshaft' is driven in one direction, and carries a rotating block or 'cylinder barrel' mounted on it by a universal joint and which is free to move axially.

This 'cylinder barrel' contains cylinders bored parallel with the mainshaft, having ports of reduced section opposed to two kidney-shaped ports contained in the face of the 'valveplate,' which is stationary.

The 'cylinder barrel' rotates in contact with the 'valveplate' face in a state of hydraulic balance, initial contact being obtained by spring loading.

Each cylinder is fitted with a 'piston' connected by a ball-ended rod to the 'socket ring' which in turn is connected to the 'mainshaft' by a double universal joint.

The 'socket ring' rotates in a non-rotatable but pivoted member called the 'tilting box' 'Michell' type bearings being interposed to take the thrust and radial load. When the 'socket

ring' is vertical the pump is in the neutral or no-stroke position, but any deviation from the vertical imparts stroke to the 'pistons' and causes delivery through one of the 'valveplate' ports and return through the other.

The greater the tilt the more delivery given, owing to the increased travel of the pistons. By reversing the angle of tilt the flow will also be reversed, and what was previously the delivery port will become the return port.

Lever or wheel controlled pumps can be arranged to give delivery in both directions from zero to maximum, but in the case of pumps fitted with auto-pump control, as applied to operate rams, etc., tilt is usually applied in one direction only, and the flow from these pumps is therefore not reversible.

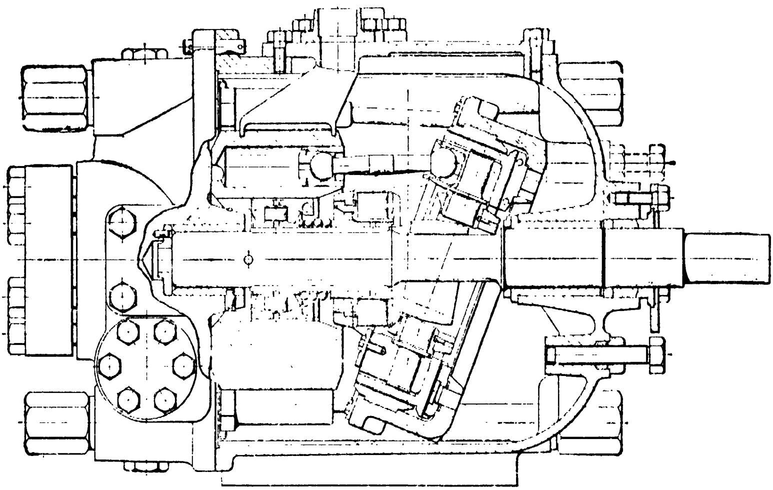


FIG. 1.

Two distinct types of pumps can be supplied, viz.: —

Lever or Wheel controlled Pumps.	To give variable delivery at fluid pressures up to the maximum pressure at which the relief valves are set, controlled by hand or mechanical means.
Automatic Pressure controlled Pumps.	To give automatically regulated delivery over a predetermined pressure range, or alternatively at a practically constant pressure. Delivery is controlled by the pressure of the fluid in circulation acting on rams connected to the tilting box in opposition to suitably rated springs.

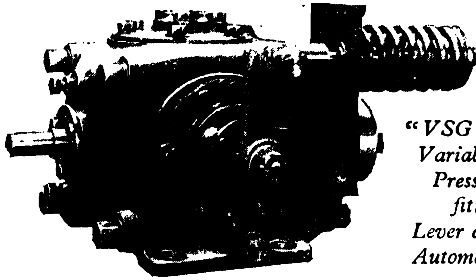
This pump also forms the 'A' end unit of the hydraulic transmission gear manufactured by this Company described below.

VARIABLE-SPEED GEAR.

THE 'VSG' MARK III. HYDRAULIC VARIABLE-SPEED GEAR.

(Vickers-Armstrongs Ltd. (Variable Speed Gear Dept.), Vickers House, Westminster, S.W. 1.)

This gear consists of two units, viz. an 'A' end or pump unit (fig. 2, p. 1116), and a 'B' end or hydraulic motor (fig. 3). These 'A' and 'B' ends can be opposed to one another directly with a common connecting plate termed a 'valveplate,' forming a type 'C' gear, or they can be separated to any reasonable distance and connected by piping, each unit having a separate valveplate, forming a type 'K' gear.



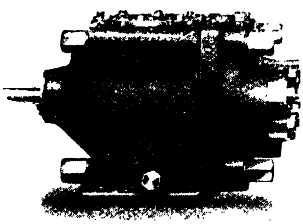
"VSG" Mark III Variable Delivery Pressure Pump fitted with Lever and External Automatic Control.

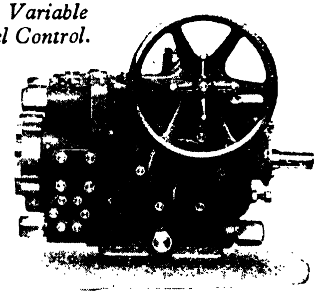
"VSG" VARIABLE DELIVERY PRESSURE PUMPS

(Regd. Trade Mk.)

SUITABLE FOR ALL SYSTEMS EMPLOYING OIL AS THE HYDRAULIC MEDIUM

Type "K" "VSG" Mark III Variable Speed Gear, fitted with Handwheel Control.





"VSG" HYDRAULIC VARIABLE SPEED TRANSMISSION GEARS

(Regd. Trade Mk.)

Infinite variation of speed without steps, forward and reverse. Positive control. High starting torque. Eminently suitable for controlling machinery for industrial purposes, viz.:—

Annealing Furnaces	Floorcloth Machinery	Photographic Paper and Film Treating Plant
Cable-Making Plant	Iron & Steel Works Machinery	Rubber Machinery
Colliery Plant	Linoleum Machinery	Textile Machines
Conveyors	Paper-Making Machinery	Wire-Making Machines
Coilers for Belts, etc.	Printing Machines	Woodworking Machines
Foundry Plant		

Also for Ship Machinery, including Steering Gears, Windlasses, Capstans, Boat Hoists, Winches, etc.

Extensively used for control of Ordnance, Director Firing Gear, Ammunition Hoists, and other special purposes where positive control is essential.

VICKERS-ARMSTRONGS LIMITED

(VARIABLE SPEED GEAR DEPARTMENT)

VICKERS HOUSE, WESTMINSTER, S.W.1.

Telephone: Abbey 7777. *Telegrams: "Varispeed, Sowest, London."*

F. 1114

The flexibility of arrangement possible with the ' VSG ' gears is a valuable feature, particularly so where considerations of space enter into account. Oil, which is practically incompressible, is the medium of transmission of power, and as this is supplied under pressure to the most important working surfaces and the whole of the machine casing is filled with oil, lubrication is perfect and wear negligible.

Construction.

The ' B ' end or hydraulic motor is generally similar in construction to the ' A ' end or pump unit, which is fully described above, a number of its components being identical with those of the ' A ' end. For the majority of purposes no controls or relief valves are necessary in the ' B ' end, and the tilting box is usually fixed, giving the ' B ' end a fixed capacity.

Where cooling is desirable, a water cooling coil is fitted in the ' B ' end around the cylinder barrel.

Whilst in some cases ' Michell ' type bearings are fitted to take the axial and radial thrust behind the socket ring in the ' B ' end, in certain equipments ' VSG ' ' B ' ends are fitted with roller bearings.

Operation of the Gear.

When a ' VSG ' gear is ready for running, the entire space within the cases and valveplate of the type ' C,' and valveplates and piping in the type ' K,' is filled with oil.

A definite portion of the oil is enclosed within the ' cylinders ' ahead of the pistons, valveplate port passages, and piping of type ' K.' This is the working oil used in transmitting energy, the case oil not being subject to pressure.

Assuming the tilting box with its socket ring is set at the neutral position (*i.e.* perpendicular to the mainshaft), the ' A ' end mainshaft, if rotated, will carry with it the socket ring and the cylinder barrel, together with the pistons and connecting rods. The pistons will not reciprocate or move to and fro in the cylinders, and as there will be no delivery from the pump, the ' B ' end will remain stationary.

If the control be moved a little, so as to move the top of the tilting box away from the ' valveplate,' and the ' A ' end mainshaft is rotated in a clockwise direction (looking on its shaft end) all the pistons as they move down on the right-hand side will be driven in towards the valveplate, expelling the oil contained in the cylinders through the cylinder ports and the large port in the same side of the valveplate. The ' B ' end, as is explained later, will then commence to rotate at a speed corresponding to the amount of delivery given by the ' A ' end.

The pistons on the left-hand side of the ' A ' end will, as they ascend, be drawn away from the valveplate, and their cylinders will fill with oil from the valveplate port facing them.

It should be noted that when a piston reaches the top or highest position in its revolution, and also the lowest position, it for an instant makes no end movement. This corresponds with the cylinder port crossing the top or bottom lands or spaces between the two valveplate ports.

The quantity of oil forced through the valveplate port depends upon the length of stroke imparted to the pistons and consequently upon the angle at which the tilting box stands.

The ' B ' end is an inversion of the 'A' end, and the ' B ' end socket ring is carried normally at a fixed angle of about 20°, the top of the angle box lying back from the valveplate, and when the ' B ' end mainshaft is rotated the ' B ' end pistons will make their full stroke as they pass between the bottom and top pistons.

The oil expelled by the 'A' end pistons through the pressure port of the valveplate enters the ' B ' end cylinders facing that port and exerts pressure on the pistons in those cylinders.

Due to the socket ring being housed in the angle box, the pistons cannot move except by rotating the socket ring in its bearings on the inclined plane, thereby converting the reciprocating motion of the pistons into rotary motion of the socket ring and the shaft to which it is connected.

The socket ring therefore revolves and carries with it the mainshaft to which it is connected by the universal joint. The ' B ' mainshaft in turn rotates its cylinder barrel, and the whole of the ' B ' end rotating group revolves in the opposite direction to the rotation of the 'A' end mainshaft.

The speed of rotation of the ' B ' end mainshaft depends upon the quantity of oil delivered by the 'A' end. For example, if each cylinder has a capacity of, say, 3 cu. ins., one complete revolution of the ' B ' end group of 11 cylinders would transfer 33 cu. ins. of oil from the right-hand side to the left-hand side. If the top of the 'A' end tilting box be tilted away from the valveplate sufficiently to cause the socket ring to impart a small stroke to the 'A' end pistons corresponding to an output from each cylinder of, say, 0·01 cu. in., all the 11 pistons of the 'A' end will together transfer 0·11 cu. in. of oil from the left-hand to the right-hand side at each revolution of the mainshaft.

On the basis of the capacity of the ' B ' end cylinders per revolution being 33 cu. ins., 300 revs. of the 'A' end mainshaft at this degree of tilt will be necessary to rotate the ' B ' end mainshaft once. If, however, the 'A' end socket ring be tilted still farther, the output of the 'A' end will be greater and the ' B ' end mainshaft will rotate proportionately faster, untilat full stroke both ends are at the same speed.

So far reference has been made to tilting the 'A' end socket ring in one direction only—namely, by moving the top of the tilting box away from the valve plate. If, however, it be moved in the

THE 'VSG' MARK III. HYDRAULIC VARIABLE-SPEED GEAR.

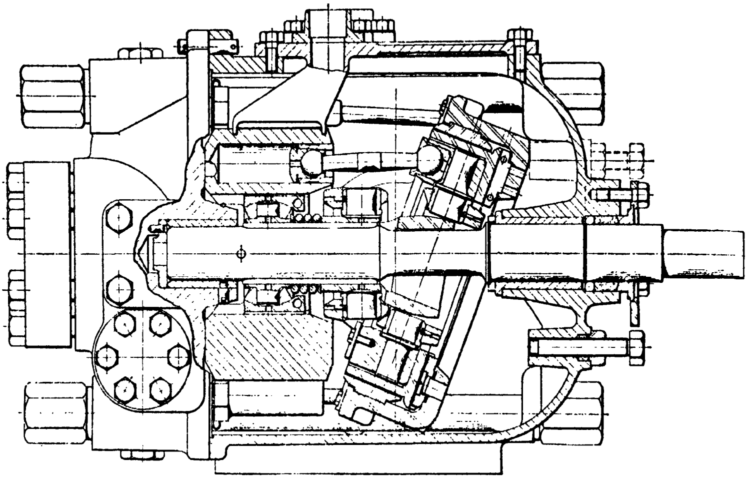


FIG. 2.—Vertical section through 'VSG' Mark III. 'A' end, fitted with lever control.

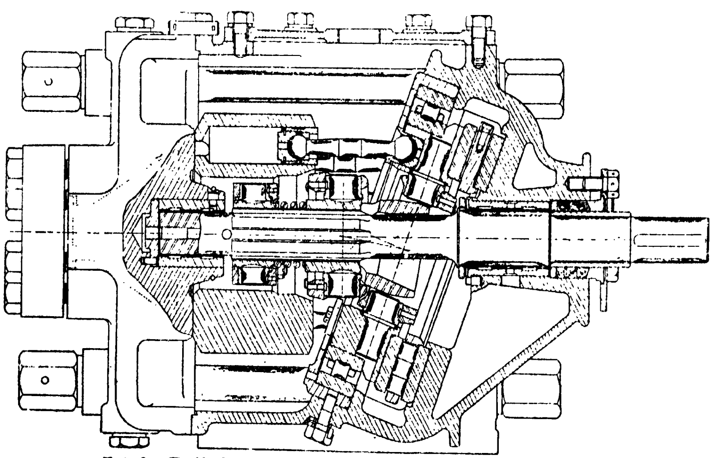


FIG. 3.—Vertical section through 'VSG' Mark III. 'B' end, fitted with roller type bearings.

opposite direction, the oil will be discharged through the left-hand port instead of through the right-hand port as previously; this will cause the 'B' end group to rotate in the opposite direction to that in which it has previously revolved, and it will now correspond in direction of rotation with the 'A' end mainshaft—viz. clockwise. The gear is therefore fully reversible.

When the 'A' and 'B' ends are of equal capacity, the maximum speed of the 'B' end mainshaft will be equal to that of the 'A' end, forward or reverse as required.

For some purposes 'A' and 'B' ends of different capacity are combined, usually a large 'B' end being employed, thereby giving a fixed reduction ratio as well as variable and reverse speeds.

Where a reduction or step-up gear is necessary on either the input or output shaft of the gear this can, in many cases, be arranged within the gear casing.

Controls.

Many forms of control have been developed to meet various requirements, these being designed basically to vary the stroke of the pistons of either 'A' end or 'B' end units, or both, as desired, giving perfectly smooth and stepless speed variation, although valve controls are also employed in some cases.

In its simple form the control may consist of a lever mounted directly on an extension of one of the tilting box trunnions or a rotary shaft or handwheel operating through gears directly on the top of the tilting box, for manual or mechanical operation.

It is also possible to synchronise and/or link the controls of a series of machines so that predetermined relative speeds can be maintained. Remote control is a frequent requirement and can be arranged hydraulically, electrically or mechanically to suit the conditions obtaining.

Various types of automatic control operated by pressure generated in the hydraulic system have also been devised. By this means automatic reduction of 'A' end delivery as oil pressure rises can be arranged, or, under certain conditions a constant oil pressure ensured irrespective of delivery.

The characteristics of such controls ensure that the pre-determined torque and horse-power are not exceeded whilst retaining the ability of the gear to work over a large speed and torque range. In the event of overload the control stalls the gear without risk of damage or stopping of the prime mover and obviates the necessity of using skilled operators to safeguard the equipment. It is a very valuable feature where dealing with loads which may vary considerably or where obstructions are likely to be encountered.

Automatic controls when fitted to 'VSG' pumps operating self-contained hydraulic systems such as on presses, render the fitting of accumulators unnecessary.

It should also be mentioned that pressure control of 'B' end tilt can also be provided so that in cases where constant horse-power is required such as in a coiling drive, where the linear speed and tension of the material to be dealt with must be maintained at pre-determined rates, this can be achieved automatically by the pressure in the system.

Advantages.

The special properties of the 'VSG' Hydraulic Variable Speed Gear can be summarised as follows:—

(1) Speed change without declutching from prime mover.
(2) Ability to 'inch' and rotate at very low speeds.
(3) High torque at low speed.
(4) Perfectly smooth acceleration and retardation at any constant or variable rate.
(5) Quick reversibility without shock.
(6) Complete positive and simple control in either direction.
(7) Steady speed under fluctuating loads.
(8) Enables constant speed non-reversing prime mover to be used.
(9) Protects complete installation from damage due to overloads.
(10) Reduces labour and operating costs.
(11) Repairs and maintenance negligible.

An important feature inherent in the 'VSG' gear is the braking effect which can be obtained when it is used with a regenerative prime mover, enabling the retardation of masses having large momentum and unbalanced loads to be carried out under absolute control, *e.g.* in the case of a hoist, loads may be lowered at any desired speed, decelerated and held in all positions without the use of mechanical brakes. In fact, the same degree of control is available when lowering as when hoisting.

With electrically driven machinery a distinct advantage is obtained by the use of the 'VSG' gear, inasmuch as the simplest and cheapest type of motor can be installed, *i.e.* squirrel cage induction or shunt wound DC., also controllers and resistances with their attendant heavy current losses are dispensed with, a light load starter only being necessary. Current peaks when starting heavy loads are eliminated.

THE EXACTOR 'SELF-SEALING' PIPE COUPLING.

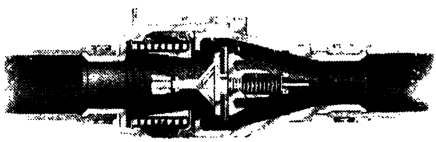
(World Patents.)

(*Exactor Control Company, Ltd.*, 14 *Berkeley Street*, *London*, *W.* 1.

The Exactor Self-sealing Pipe Coupling is a coupling with a double function. First, to bring two pipe lines into connection. Second, to seal off each pipe line when disconnection has to be made.

It is therefore by turn an orthodox coupling and an automatic sealing device. This double function is effected by means of a two-part construction. When the two parts are joined, the device resembles and functions like a normal pipe coupling. When separated, each half acts as a seal on the pipe line to which it is attached.

The automatic sealing function ensures that there is no escape of the liquid or gas—not even a 'dribble'—when disconnection is made, and eliminates the need for operating any supply cock or for draining down.



CONNECTED.
Full-bore flow through coupling.

DISCONNECTED.
Both halves completely sealed off.

Re-connection is an equally simple and swift operation, free from spillage or air trapping. The range of standard Exactor Self-sealing Couplings covers medium-pressure systems up to 300 lbs. per sq. in. or high-pressure systems up to more than 2,500 lbs. per sq. in., in sizes from ¼-in. bore to 4-in. bore and upwards. They are serviceable for a variety of fluids, from hot oil to refrigerants, under any conditions of vibration or severe treatment.

Description of Operation.

Each half of the device contains a resilient-faced sliding valve, spring-loaded on to its seat. As the two halves connect by screwing home the outer ring, the two springs are compressed and the valves slide back. Thus a full-bore flow path is created between the two pipe lines.

The process works in reverse as the outer ring is unscrewed to effect disconnection, viz., the two springs gradually extend, progressively close the valves and so close the flow path as the resilient valve-faces come to rest on their seatings.

At this stage, the two halves are still engaged but the pipe lines are now effectively sealed. A few more turns of the ring then disengages the two halves of the coupling and the pipe lines are entirely disconnected.

The patented features of the device provide against leakage of the smallest degree whilst two pipe lines are connected and whilst they are in process of connection or disconnection. All tests for leakage or air trapping give a 'Nil' result.

THE EXACTOR PIPE-LINE CHANGE-OVER SWITCH.

The Exactor Pipe-line Change-over Switch is a device based upon the use of the Exactor 'Self-sealing' pipe coupling described in the foregoing paragraphs.

It provides a method of switching hydraulic pipe lines of all kinds from one alternative supply circuit to another with a precision and speed analogous to the switching of electric conductors.

It has been designed to overcome all the shortcomings of a change-over effected by manual operation of individual separate cocks or valves, and makes incorrect combinations or 'crossed' circuits impossible. Each pipe in the system must be either positively engaged or positively sealed off.

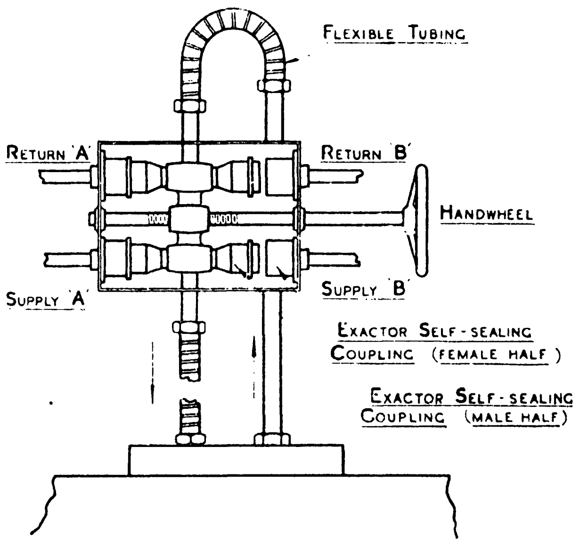


Diagram to illustrate working principle of Exactor Pipe-line Switch.

Apart from its fool-proof character, the Exactor Switch Unit has been proved in service to offer great savings in time required to effect a change-over.

Typical Uses.—To switch the circulation through an apparatus from hot water (or steam) back to cold water and *vice versa.* Many rubber calendering rolls and press platens have been equipped with Exactor Switch Units for this purpose.

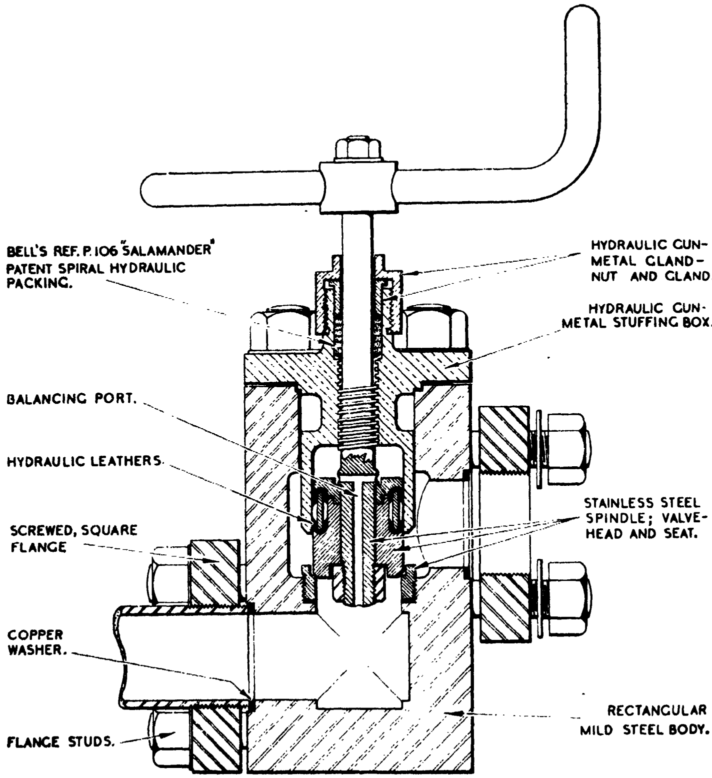
Description of Operation.—As the handwheel is rotated in either direction, the threaded portion of its spindle causes the central carriage to move to the left or to the right accordingly. Thus the carriage will engage alternatively the circuits marked 'A' or 'B' on the diagram, and so determine the service circulated through the jacket or other chamber represented at the bottom of the diagram.

The Exactor Self-sealing Couplings slide together or apart without any spilling or air inclusion and their progressive action permits throttling control by the handwheel.

High Pressure Balanced Hydraulic Stop Valve.

(Bell's Asbestos and Engineering Limited, Slough, Bucks.)

Modern hydraulic equipment frequently operates at pressures of several tons per sq. inch. The special balanced action of the 'Bestobell' high pressure hydraulic stop valve ensures easy opening and closing even at maximum line pressure, and avoids the use of a by-pass fitting.



The balanced action is produced by means of a small port drilled up the centre of the lower part of the valve spindle, causing pressure to act on both the upper and lower surfaces of the valve head. By this means finger-light manipulation can be obtained.

The valve body is constructed from a solid mild steel billet, and the spindle, valve and seat are of stainless steel.

The standard valve is designed for pressures up to 3,000 lbs. p.s.i. and a heavier type is available for pressures of 6,000 lbs. p.s.i. and over.

SECTION XIX

PART III

Pumps and Pumping.

THE CLARKE CHAPMAN DIRECT-ACTING FEED PUMPS.

(Woodeson's Patent.)

(*Clarke, Chapman & Co., Ltd., Gateshead-on-Tyne.*)

The single-cylinder direct-acting type of pump, on account of the small number of working parts and consequent small liability to break down, is very largely employed. Figs. 1 and 2, p. 1122, show Woodeson's 'Simple' Patent Direct-acting Pump. This pump, which is extensively used, works with all steam pressures up to 400 lbs. per sq. in.

The steam chest is manufactured of the best hard close-grained cast-iron, of a suitable thickness to withstand any tendency to distort through the temperature of high-pressure steam.

The working parts of the steam chest consist of one auxiliary valve and one main valve, both of piston valve type.

The auxiliary valve is operated by the valve gear, actuated by the piston-rod crosshead. The main valve is partly moved mechanically and partly operated by steam, controlled by the auxiliary valve.

The action of the valves is as follows :—

When the main piston is nearing the end of its stroke, and the auxiliary valve commences to be moved by the valve gear, the first part of the movement closes the port through which the exhaust from one end of the main valve has been passing. The next part of the movement of the auxiliary valve opens steam to this end of the valve, and as the steam to the opposite end has not been closed, the main valve is in equilibrium. By the time that this takes place the auxiliary valve has moved so as to be in contact with the main valve, and further movement of the valve gear causes the auxiliary valve to move the main valve mechanically and so close the ports to the main cylinder. When these ports are nearly closed the auxiliary valve has closed the steam port at the end of the main valve, towards which the valves are moving, and opened an exhaust port, with the result that the main valve is moved by steam to the end of its travel and the steam cylinder ports are opened for the return stroke.

Should the main valve by any chance be delayed in being moved by steam, it would be worked mechanically during the clearance in the steam cylinder to a sufficient extent to open the steam port at the end of the cylinder towards which the piston was moving, and so forming a steam cushion to prevent the possibility of the piston striking the cover.

Advantages of the gear are :—

The main and auxiliary valves can all be mechanically actuated from the outside; this is an advantage if the pump is standing for any length of time, as it keeps the faces from getting rusted up.

During the process of the mechanical movement, the piston to which this movement is imparted is in perfect equilibrium.

There is entire absence of valve setting.

These pumps are also made on the TANDEM COMPOUND principle. Great economy has been obtained with the pumps, and they are specially applicable for electric power stations and other high-class installations where economy of steam is an important consideration.

Fig. 3, p. 1122, shows a pair of these pumps fitted with automatic controlling gear, as used for MAIN INDEPENDENT FEED-PUMPS FOR MARINE PURPOSES; with this automatic controlling gear there must be no dead points, and the pump must be able to start from any position, this being a special recommendation for a feed-pump for marine purposes, where, in order to derive all the benefits to be gained by using independent main feed-pumps, the speed of the feed-pumps must be automatically regulated according to the quantity of water to be dealt with, which varies with the speed of the main engines. The 'Woodeson' pump contains all the necessary features for this work.

No flat surfaces exist in any part of the water end exposed to pressure. Both suction and delivery valves are arranged in groups, and are easily accessible for overhauling. The plunger junks are of gunmetal, and are fitted with composition plunger rings, specially manufactured to deal with high pressure and high temperature.

Test taken with Clarke, Chapman Simple Type of Slow Speed Feed-Pump (Woodeson's Patent) :—Diameter of steam cylinder, 10½ ins.; Diameter of water cylinder, 8 ins.; Length of stroke, 18 ins.; Water delivered per lb. of steam used, 81 lbs.; Steam used per water h.p. hour, 59·5 lbs.; Steam pressure per sq. in., 180 lbs.; Pressure in discharge pipes per sq. in., 195 lbs.; Volumetric Efficiency of pump, 97 per cent.

Test taken with Clarke, Chapman Tandem Compound Type of Slow Speed Feed-Pump (Woodeson's Patent) :—Diameter of h.p. cylinder, 8½ ins.; Diameter of l.p. cylinder, 13 ins.; Diameter of water cylinder, 8½ ins.; Length of stroke, 18 ins.; Water delivered per lb. of steam used, 110·5 lbs.; Steam used per water h.p. per hour, 36·5 lbs.; Steam pressure per sq. in., 195 lbs.; Pressure per square in. in discharge pipes, 215 lbs.; Volumetric Efficiency of pump, 97 per cent.

CLARKE CHAPMAN STEAM PUMPS.

FIG. 1.

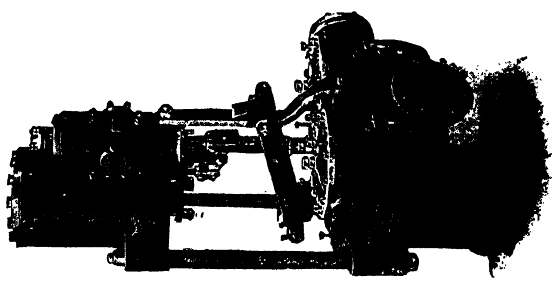


FIG. 2.

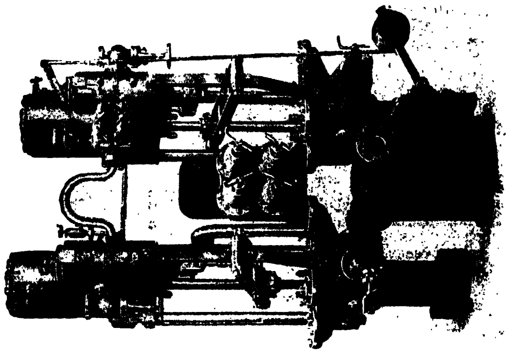


FIG. 3.

MADAN'S AIRHYDROPUMP.

(*Charles S. Madan & Co., Ltd., Vortex Works, Broadheath, Altrincham.*)

This machine is an air operated hydraulic ram pump providing a small supply of water, oil or kerosene at any desired pressure within limits, at the expense of a small quantity of compressed air taken from the usual air line found in most engineering shops.

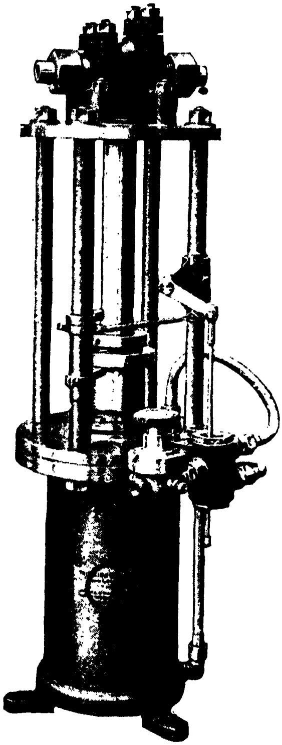


FIG. 1.—Single-Acting Pump.

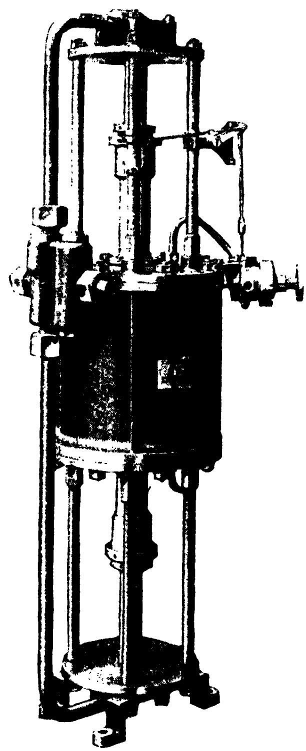


FIG. 2.—Double-Acting Pump.

The following description will serve to explain how the pump works:—

A double acting air pressure cylinder and piston are provided, and the movement of the piston is imparted to a water cylinder containing a fixed ram. The air piston being of larger diameter than the ram, an increased pressure is produced in the water cylinder.

In operation, so long as the total load in the water cylinder is less than that in the air cylinder, the pump will continue to function and will automatically reverse at the end of each stroke. When the pressure builds up so that the loads are equal the pump comes to rest, but restarts as soon as the balance is disturbed.

The hydraulic pressure is regulated by a simple control valve on the air supply, which may be left set at any desired pressure.

Models are made giving maximum pressures of 40 to 15,000 lb. per sq. in. hydraulic, with air at 100 lb. maximum pressure.

The pump weighs from 110 lb. to 260 lb. according to type.

THE PULSOMETER STEAM PUMP.

(*The Pulsometer Engineering Co., Ltd., Reading.*)

In this pump, figs. 1 and 2 (general view), the neck J contains a gun-metal ball I, above which is a steam pipe K. EE are suction valves placed at the base of two chambers AA. Immediately above the suction valves the two chambers communicate with a 'discharge box'

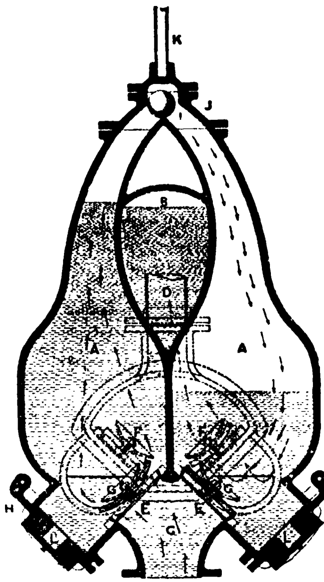


FIG. 1.

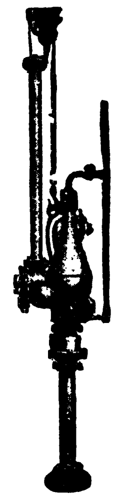


FIG. 2.

The Pulsometer Steam Pump.

through two discharge valves FF. The valves in the standard pattern of pump are of the grid type, consisting of a grid, rubber, and guard. In addition to the above valves there are three small gun-metal valves (not shown in the figure) for the admission of air, one of each being screwed into the upper part of the two water chambers and the air vessel.

Action.

Assuming one of the chambers to be open to steam and full of water, the steam entering by the steam pipe and past the ball passes into the chamber and presses upon the small surface of water exposed, depresses it without agitation (and therefore with little condensation), and drives it (as seen in the figure) through the discharge valves FF into the rising main, D. The moment the water in the chamber falls to the level of the opening in the branch leading to the discharge box, the steam blows through with a certain amount of violence, and as it is brought into intimate contact with the water in the discharge box an instantaneous condensation takes place, and the vacuum thus formed in the emptied chamber immediately pulls the control ball over on to the corresponding seat and cuts off further admission of steam, allowing the vacuum to be completed. Water immediately enters through the suction pipe, and lifting the inlet valve rapidly fills the chamber again. A similar operation has been taking place in the fellow chamber, the period occupied by filling one chamber corresponding to that of emptying the other, and these operations continue alternately in the two chambers so long as the pump is supplied with steam and water. The function of the air valves is to introduce a small quantity of air at each stroke or 'pulsation' for the purpose of cushioning the ball when it changes its position, and for separating the steam from the water by a non-conducting film so as to prevent loss by condensation during the expulsive portion of the cycle.

The 'Pulsometer' Steam Pump is largely used by contractors for pumping from excavations for the following reasons: it works equally well when fixed or suspended from a chain or rope; its small size proportionate to the quantity of water thrown makes its use possible where other pumps cannot be used; and it can pump water heavily charged with mud and sand.

The 'Pulsometer' Steam Pump will work day and night without attendance so long as steam and water are supplied and conditions remain constant.

TURBINE PUMPS.

(The Pulsometer Engineering Co., Ltd., Reading.)

Fig. 3 shows a typical example of a high lift turbine pump suitable for water supply, mine drainage, boiler feeding and similar duties.

The pump consists of a number of cells bolted together, each with its own impeller, the water leaving one impeller to pass through a diffuser to the next. It is, therefore, simple to arrange for a series of impellers which together are capable of giving total heads up to 3,500 ft. or more.

To keep the overall length of the pump within manageable dimensions, each impeller is of the single entry type. The areas of the two shrouds therefore differ by the area of the impeller eye, and an axial thrust is set up equivalent to the total pressure the pump is developing multiplied by this area. A hydraulic balancing device in the pump takes up the thrust more simply and reliably than any mechanical thrust bearing. Fixed to and revolving with the spindle is a balancing disc, its periphery moving just clear of a seat fixed to the delivery cover. Having no collars on the spindle, the whole rotating element is free to move axially when the pump is at rest. On starting up, however, the out-of-balance thrust moves the shaft axially towards the suction, reducing the clearance between the disc and the seat, thus building up a difference in pressure between the two sides of the balancing disc. This building up process continues until the difference in pressure balances the axial thrust towards the suction, the area of the disc being calculated to hold the rotating element in position against the axial thrust with a fine clearance between disc and seat which keeps the water passing the balance disc down to a negligible quantity.

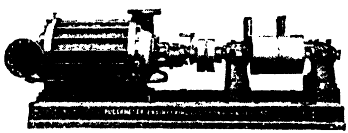


FIG. 3.

The balance chamber into which this water discharges is connected back to the pump suction branch. Therefore, no matter what head the pump is working against, the delivery gland is subject to suction pressure only.

Absence of all metallic contact in the pump means that there will be no wear except when gritty water is pumped. A gauge provided to indicate any wear taking place on the balancing device, makes possible the adjustment and restoration of the pump to its original efficiency.

CENTRIFUGAL PUMPS.

(The Pulsometer Engineering Co., Ltd., Reading.)

The pump shown in fig. 4 is a good example of the present-day trend in centrifugal pump design. The casing is arranged in two halves and is split along the horizontal centre line, the suction and delivery branches being cast in the bottom half of the casing, which is bolted to the pump bedplate. By this means it is possible by the simple expedient of removing the top half of the casing or cover to inspect the whole of the interior of the pump and, if necessary, even remove the rotating element, without in any way interfering with the alignment of the pump or breaking the pipe joints. The impeller, which is of the double entry type, is inherently self-balancing and so no axial thrust is set up while the pump is in operation, a light locating thrust bearing being provided to centre the shaft when the pump is being started up or shut down.

The shaft being supported at each end in a sleeve or ball bearing makes the pump particularly suitable for water which contains a little grit or other abrasive material. Renewable

rings are fitted in the casing where the impeller runs with a fine clearance, so that the clearance can be simply and cheaply restored and the pump brought back to its original efficiency. The glands normally are sealed by water bye-passed from the discharge side of the pump, but again, when abrasive materials are to be handled, it is a simple matter to convert the pump so that the glands are sealed by grease or by an external supply of clean water.

This form of split casing construction is also extended to two-stage pumps and multi-stage pumps where the head to be developed by the pump is higher than can be efficiently obtained with a single impeller, and the accessibility to the internal parts of the pump is of considerable advantage. In the case of the two-stage pump this is arranged with the impeller eyes facing outward, the suction branch communicating with the eye of the first-stage impeller, the delivery from which is carried through volute passages crossing over the second stage to the eye of the second impeller and then to the delivery branch. By this means it is possible to balance the hydraulic thrust on the impeller, and the shaft is definitely located by means of an external ball thrust bearing, generally as described above.

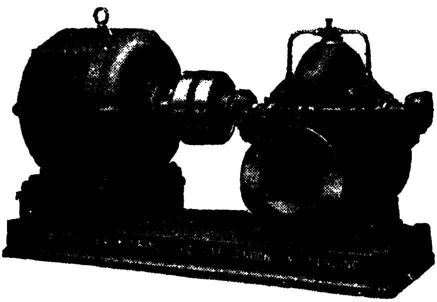


FIG. 4.

When this construction is extended to the multi-stage pump, complicated construction and high pressure delivery end stuffing box are avoided by mounting the impellers all the same way, passing the water direct from the first to the second impeller in series, and providing a balance disc to balance the hydraulic thrust as in the case of the turbine pumps previously described.

Single Entry Centrifugal Pumps.

For industrial and general service applications a very comprehensive range of end suction single-stage pumps can be used. These are arranged with vertical flanged joints and single entry overhung impeller, which is hydraulically balanced either by means of a ring at the back of the impeller inside which is a space connecting with the suction side of the pump, or by means of balancing vanes. Either of these methods have the object of balancing the hydraulic thrust and relieving the pressure on the stuffing box of the pump which is located as close as possible to the back of the impeller.

In some cases where the pump is handling perfectly clean water, and other liquids having comparatively good lubricating properties, it is possible to incorporate an internal bearing in the stuffing box immediately behind the impeller, and one external bearing arranged to share the journal load and locate the rotor in its correct position.

A more reliable arrangement, particularly for those pumps handling dirty water and liquids having no lubricating properties, can be embodied in the Pulsometer pumps of this type. In this design the rotating element is carried on two well spaced external bearings, one being placed as close as possible to the overhanging impeller, and the other one arranged to share the rotating weight and prevent any mechanical contact in any part of the pump, except through the gland packing. The inner bearing locates the pump rotor axially, and the outer bearing is normally a roller bearing which would be suitable for carrying a comparatively heavy belt pull for overhanging pulley drive. Adaptations of this type of pump have been provided by the Pulsometer Engineering Co. for handling a very diverse range of liquids, and the working parts of the pump can be provided in materials which include special irons and steels, bronzes, stainless steel, ebonite, rubber and stoneware.

Direct-Acting Feed Pump.

(G. & J. Weir, Ltd., Cathcart, Glasgow, S. 4.)

The Weir Patent Direct-Acting Feed Pump, shown in fig. 1, p. 1128, is the original of a type of pump which is largely used. Introduced at first for marine boiler feeding, its adoption by the mercantile marine, the British Navy, and numerous foreign navies, caused it afterwards to be placed on the market in a form suitable for land installations, power plants, etc.

The valve gear is positive—*i.e.* the steam valve can never be in such a position that the pump will not start immediately when steam is turned on. The only possible way in which the main valve can rest is at full travel, either for an up or down stroke of the piston. The valve arrangement also ensures constant length of stroke and certainty of action.

The steam valve is a D slide-valve, with a small auxiliary slide valve working on the back, and hence has not the same liability to leak as a piston valve. These are the only two moving parts proper in the steam chest, so that there is little opportunity for wear, and no delicate adjustments to get out of order. The steam is used expansively, and the cut-off can be regulated from the outside while the pump is working.

The water valves are of the Weir group type, providing a large area with only a small lift, thus ensuring easy working and little wear and tear. The pumps are designed to work at a moderate speed and the piston is slowed down towards the end of the stroke, enabling the valves to settle quietly on their seats. The working and maintenance of the pumps is therefore much more satisfactory than in pumps running at a high speed. In the matter of steam consumption, numerous tests and comparisons show that the Weir pump is highly economical; this is due to the fact that it has fewer ports and passages than a duplex pump, and the simple expansion arrangement enables it to be built so that the clearance is small.

Where it is desired to regulate the boiler feed automatically, a special design of float tank and regulating gear is furnished.

The Weir Feed Pump is more extensively used than any other type for feeding water tube boilers; these contain a relatively small quantity of feed water and to maintain a constant water level in the boiler for continuous steady steaming the feed pump must be capable of providing a regular and uninterrupted supply of feed water at any pressure necessary. For this service the Weir Pump is unequalled on account of its slow steady movement and complete reliability under any pressure. The pump can work in conjunction with any of the recognised types of boiler feed regulators.

For steam wagons, rail coaches, cranes, and all small stationary or portable boilers, a horizontal pump known as the Weir Junior Feed Pump is built, for capacities up to 130 galls. per hr.

Standard Sizes and Capacities of Weir Feed Pumps.

Size.	Normal Duty.		Maximum Duty.	
	Gallons per Hour.	D.S. per Min.	Gallons per Hour.	D.S. per Min.
Pump. Cyl. Stroke. Ins. Ins. Ins.				
2½ × 3½ × 5	270	32	340	40
3 × 4½ × 6	410	28	535	36
4 × 6 × 7	660	21	810	26
4 × 6 × 12	960	18	1,205	22½
5 × 7 × 12	1,450	17	1,725	20
6 × 8½ × 13	2,000	14	2,350	16½
6 × 8½ × 18	2,700	13½	3,350	16½
7 × 9½ × 21	4,000	13	4,750	15
8 × 10½ × 22	5,500	12½	6,350	14½
9 × 12 × 24	7,200	12	8,100	13½
10 × 13½ × 24	9,000	12	10,000	13½
11½ × 15½ × 24	11,250	11½	12,000	12½
12½ × 17 × 24	13,200	11½	13,800	12
12½ × 17 × 28	14,700	11	15,400	11½
14 × 20 × 27	16,500	10	18,100	11
14 × 20 × 32	19,600	10	21,500	11

Standard pumps suitable for discharge pressures of 300 lb. per sq. in.

THE WEIR DIRECT-ACTING FEED PUMP.

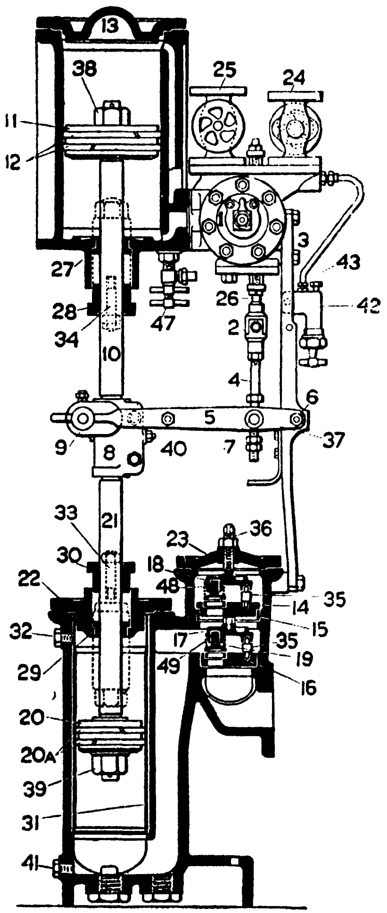


FIG. 1.

List of Parts.

1. Steam slide valve chest.
2. Double joint.
3. Front stay.
4. Bottom spindle.
5. Valve gear levers.
6. Front stay bush.
7. Ball crosshead.
8. Main crosshead.
9. Crosshead pin.
10. Piston rod.
11. Piston body.
12. Piston rings.
13. Cylinder cover.
14. Discharge valve seat.
15. Discharge valve seat ring.
16. Suction valve seat.
17. Suction valve guard.
18. Discharge valve guard.
19. Water valves.
20. Bucket.
20A. Bucket rings.
21. Pump rod.
22. Pump cover.
23. Valve chest cover.
24. Steam stop valve.
25. Exhaust stop valve (when fitted).
26. Auxiliary valve spindle.
27. Cylinder neck-ring.
28. Cylinder gland.
29. Pump neck ring.
30. Pump gland.
31. Pump liner.
32. Liner fixing pin.
33. Pump gland studs.
34. Cylinder gland studs.
35. Suction and discharge columns.
36. Top pins.
37. Ball crosshead bushes.
38. Piston rod nut.
39. Pump rod nut.
40. Crosshead taper pin.
41. Pump drain plug.
42. Plunger lubricator.
43. Adapter for lubricator.
44. Ball valves and spring for lubricator (not shown).
45. Gland packings complete.
46. Pump and cylinder joints.
47. Cylinder drain valve.
48. Discharge valve spring.
49. Suction valve spring.

Turbo-Feed Pump.

(G. & J. Weir, Ltd., Cathcart, Glasgow, S. 4.)

Weir Turbo-Feed Pumps are built in both single and multi-stage types and the range of sizes covers all boiler feeding requirements from 4,000 galls. per hr. upwards.

The driving unit consists of a steam turbine of the impulse type, having one pressure and several velocity stages. The turbine casing and bearing housings are divided horizontally, the lower portions being embodied in the general framework or base. This framework is extended at the

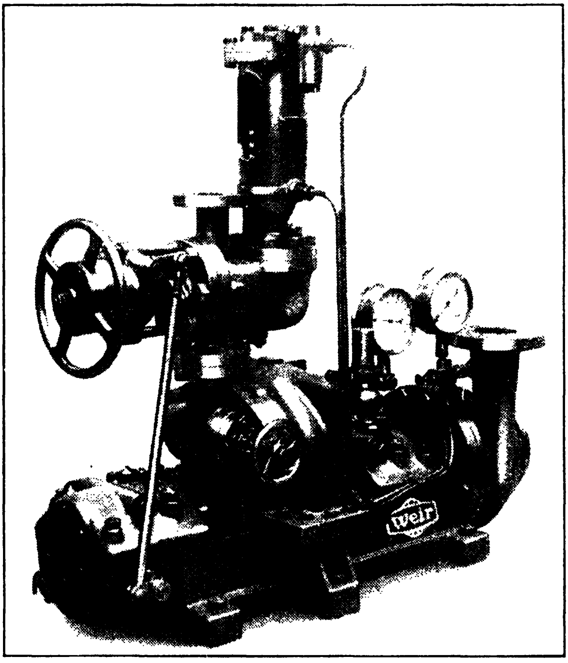


Fig. 2.—The Weir Single-Stage, Two-Bearing Turbo-Feed Pump.

pump end to form a flange to which the pump is bolted. In the single-stage, medium pressure type, the impeller is carried on the over-hung shaft, while in the high pressure type an outer pump bearing is added to prevent shaft deflection under load. The multi-stage pumps are all of the three-bearing type. The supports of the pump and turbine are arranged so as to ensure alignment under all operating conditions, centre-line suspension being employed in many cases where temperatures are very high.

The pump is self-regulating for all ordinary changes in water and steam conditions, a pressure operated governor, working a balanced double-beat throttle valve, being fitted. Auxiliary nozzles may be brought into use by a hand-operated valve when required. An emergency governor of the eccentric ring type is provided, which, in the event of the speed of rotation exceeding a predetermined limit, actuates a trip gear which releases a spring and instantaneously closes the stop valve; there is thus no possibility of the speed of the set reaching unsafe limits in the event of an accidental failure of the water supply.

The turbine glands are provided with carbon packing, and all parts are readily accessible for examination and overhaul. The pump is fitted with a hydraulic device which automatically balances the end thrust on the shaft.

The pumps are capable of discharging against the highest boiler pressures, and can be used in conjunction with other centrifugal or reciprocating pumps. An automatic cut-in device can be supplied for use when the pump is installed as a standby.

The Weir Turbo-Feed Pump is a high speed unit with a correspondingly low steam consumption. Further, as the exhaust steam is free from oil, it may be used for feed water heating by direct contact, thereby utilising all the heat in the steam. A special nozzle heater can be supplied for this purpose.

Multi-Stage Centrifugal Pumps.

(G. & J. Weir, Ltd., Cathcart, Glasgow, S. 4.)

The Weir 'Electrofeeder,' a typical example of which is shown in fig. 3, is a multistage pump for boiler feeding, built for capacities of 4,000 galls. per hr. upwards, and for all pressures up to 2,300 lb. per sq. in. It is designed to be inherently stable in operation—that is to say, the pressure-volume characteristic curve falls continuously from zero to maximum output. The pump is therefore suitable for operation in parallel, either with similar pumps or other types possessing stable characteristics, without surging.

The type shown is of ring-section construction, the end covers and sections being bolted together by heavy tie bolts, the whole being mounted on a rigid baseplate which also carries the driving motor. The impellers are of the single inlet type, fitted to the shaft with keys at 180° to ensure ease of removal and freedom from shaft distortion. To take the end thrust

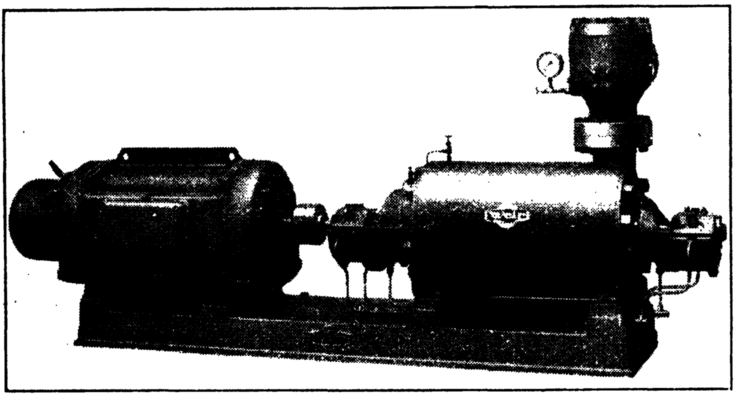


FIG. 3.—The Weir 'Electrofeeder,' Ring Section Type.

a hydraulic balance device is incorporated. The pump shaft itself is very carefully balanced dynamically, and a flexible coupling is employed between the pump and the driving motor.

Pumps of this type have been supplied for a large number of important power stations, capacities ranging up to 1,000,000 lb. per hr. Barrel casing type pumps are also built and are specially suitable for high temperature installations. Special designs, including main and booster units, electric-driven sets with turbine standbys, and complete turbo-electric sets in which the electrically driven unit operates at constant speed and at approximately boiler pressure while a steam turbine driven booster pump supplies the additional variable pressure increment according to the load, are also supplied.

Centrifugal pumps of somewhat similar design, but employing a barrel casing, and special methods of support to minimise the effects of expansion due to high temperatures, extra heavy construction throughout, special gland design, etc., are also supplied for handling hot oil in refineries, etc.

For both boiler feeding and refinery duty, the working conditions are extremely arduous, and great care has to be taken in the selection of the materials to withstand the effects of high temperatures, erosion and corrosion, etc.

SECTION XX

PART I

Ball and Roller Bearings.

TIMKEN TAPERED ROLLER BEARINGS.

(British Timken Ltd., Birmingham and Northampton.)

DESIGN.

The basis of the design of the Timken Tapered Roller Bearing (see Fig. 1) is founded on the only fundamentally correct principle for true rolling, which requires that the apices of the cones embracing the three rolling surfaces shall meet in a common point on the axis of revolution of the bearing.

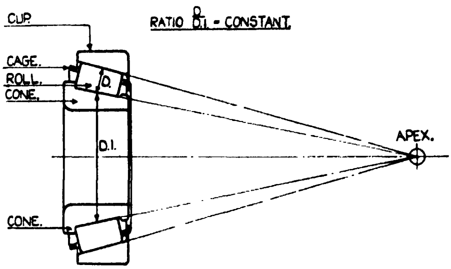


FIG. 1.—Component parts. Tapered Roller Bearings.

This principle is expressed by the equation :—

$$\text{Ratio } \frac{\text{D1}}{\text{D}} \text{ at any point along the roller} = \text{Constant.}$$

giving a constant angular velocity of the roller at any point along its length, otherwise slipping and skewing of the rollers will take place.

The main characteristics of the bearing are :—

(1) High carrying capacity for thrust and radial loads due to the long line contact between rollers and races.

(2) Ability to withstand simultaneously any combination of thrust or radial loads. A two-bearing mounting will effectively resist loads coming from all directions simultaneously.

> *N.B.*—Thrust and radial loads are both transmitted radially through the bearing at right angles to the respective lines of contact, utilising the full length of the rollers.

(3) Adjustability without interference to the geometry of the bearing, maintaining original alignment.

(4) Versatility in bearing design providing any ratio of thrust to radial capacity, determined by choice of apex position (see Fig. 2).

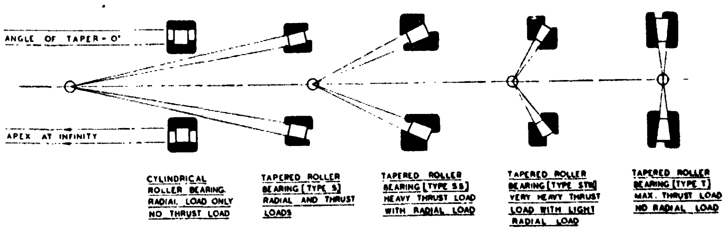


FIG. 2.—Geometric design of Tapered Roller Bearing.

(i) Apex at infinity—Cylindrical roller bearing having radial capacity only.

(ii) Apex as shown—Normal Timken Bearing (Type 'S') for heavy radial and normal thrust loads.

(iii) Apex closer to bearing—Timken Steep-Angle Bearing (Type 'SS') for heavy thrust loads and normal radial loads.

(iv) Apex in central plane of bearing—Timken Thrust Bearing (Type 'T') for thrust loads only.

(v) Low co-efficient of friction, 0.002 to 0.003 for normal loads and speeds, according to whether oil or grease lubrication is used. Value is practically constant for all speeds.

(vii) The bearing is its own oil pump. The tapered rotating elements set up a centrifugal action, which draws oil through the bearing from the small to the large ends of the rollers. Used in practice on machine tool spindles, railway axleboxes, etc., to provide an automatic oil circulation system (see Fig. 9, p. 1136).

(viii) Can be run at high speeds given an adequate lubrication system.

Types of Timken Bearings.

A general indication for the uses of each type is as follows :—

Type 'S.'	—Widely used in general engineering, automobiles and some railway axleboxes.
Type 'SS'	—General thrust load applications.
Type 'SF'	—Machine tool spindles. Facilitates accurate machining of headstock bearing housing bores, by eliminating internal shoulders.
Type 'DI'	—Widely used in railway axleboxes.
Types 'DO' and 'NA'	—For heavy radial loads, *e.g.* large reduction units.
Types 'DOS' and 'NAS'	—Alternating thrust loads, *e.g.* wormshafts.
Type 'QO'	—Used on work roll and back-up roll necks in all classes of rolling mills.
Type 'T'	—Pure thrust applications, *e.g.* crane hooks, swing bridges, automobile steering pivots.
Type V.P. Airscrew	—Special two-row bearings designed as sole support of blades of Variable Pitch Airscrews. Very high thrust capacity, with sufficient radial capacity for positive centring of blade, and races and rollers arranged to offer maximum resistance to tilting loads set up by air pressure, blade inertia and gyroscopic effects.

The multi-row bearings are combinations formed from single-row bearings of their respective types, and all conform to the basic principle of design—Fig. 1.

Bearing Mountings.

Timken bearings are practically always mounted in pairs, two fundamental methods being used (see Fig. 3).

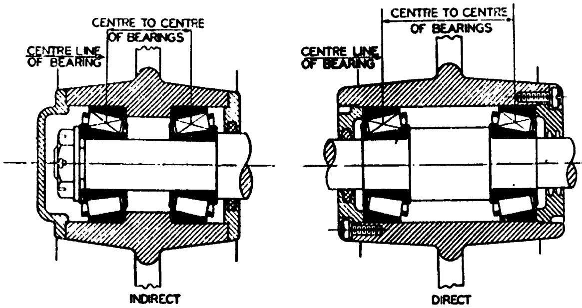


Fig. 3.—Method of mounting Timken Bearings.

Indirect Mounting.—Small ends of rollers point inwards. Used for limited bearing spacings and where the maximum resisting moment against tilting loads is required, *e.g.* machine tool, spindles, wheels.

Direct Mounting.—Small ends of rollers point outwards. Used for wide bearing spacing, and where adjustment of bearings by the cups is desired.

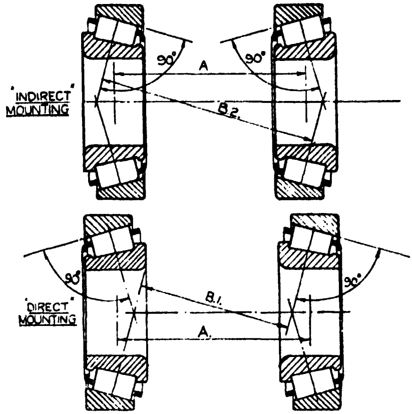


FIG. 4.—Effective Bearing Spacing.

The greater stability of the Indirect mounting is shown by a comparison of the actual effective spacings, B1 and B2 for equal axial spacings A, Fig. 4.

FITTING PRACTICE.

A good general principle is to make the revolving member a press fit to prevent slip, the stationary member being a close sliding fit for bearing adjustment. Operating, manufacturing or assembly requirements may conflict with this principle, when a compromise solution should be sought to take care of the most important requirements.

SHAFT AND HOUSING DESIGN.

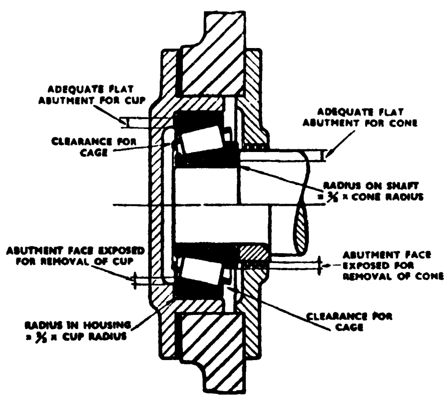


FIG. 5.—Details of Shaft and Housing Design.

Essential points in the detail design are shown in Fig. 5, which is self-explanatory.

ADJUSTMENT.

It is usual to allow a few thousandths of an inch endplay in a pair of bearings, according to the type of application and the operating requirements. A light preload of a few lb. ins. is sometimes given in special applications. Adjustment is made by moving one member of a pair of bearings. Two common methods are shown—by nut in Fig. 3 and by shims under the cover flange in Fig. 5. Fig. 5 shows the cup press-fitted into the cover spigot, and allows easy adjustment.

DOUBLE BEARING MOUNTINGS.

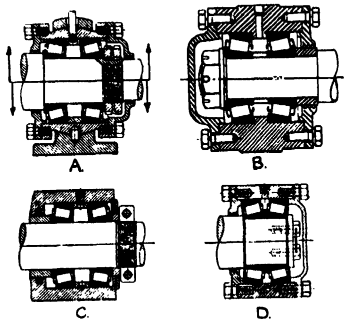


FIG. 6.—Double Bearing Mountings.

With long shafts or where thermal effects are present one double or two single bearings are mounted in separate housings at each end of the shaft. One pair axially locates the shaft, and the other pair is allowed to float. Housings can be rigid or self-aligning (see Fig. 6).

FOUR-ROW BEARING AND MOUNTING.

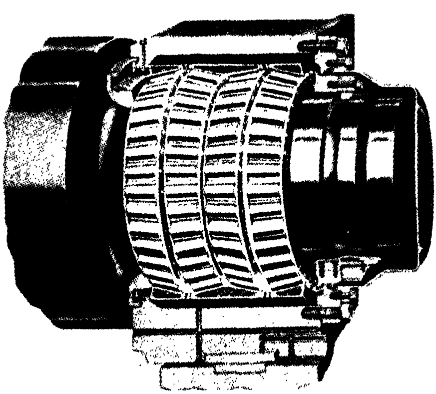


FIG. 7.—Four-row Tapered Roller Bearing Roll Neck Mounting.

Fig. 7 shows a roll neck mounting for the back-up rolls of a large 4-High rolling mill. The bearing has two double cones, one double cup and two single cups. Spacers are used between all cups and cones, ground to correct length during manufacture to give suitable running clearances in the bearing.

FLAT THRUST BEARING MOUNTING.

The following principles should be followed :—

(1) A radial bearing must be provided to centralise the revolving shaft.

(2) Races and rollers must be held in contact at all times.

(3) Faces supporting the races to be truly flat and square with axis of revolution, and robust enough to maintain this condition under maximum load.

(4) The outer rib of the revolving race to be mounted concentric with axis of revolution, and stationary race to have slight radial float to allow of self-centring.

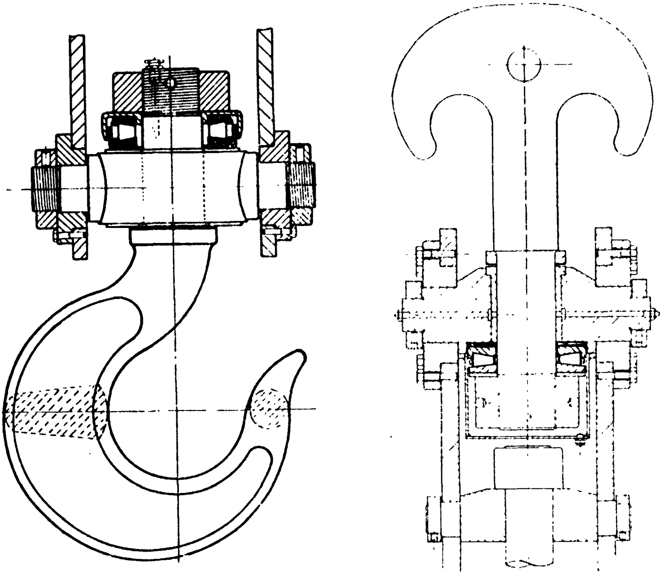


FIG. 8.—Crane Hook Mountings.

Fig. 8 shows (L.H.) light crane hook mounting, and (R.H.) shows 200 ton crane hook mounting.

LUBRICATION.

Oil or grease is used, the choice depending on bearing size, r.p.m., operating temperature, loading and working conditions.

(a) *Grease.*—Grease can be used for bearings of 6 in. o.d. up to 1,000 r.p.m. and for bearings between 6 in. and 12 in. o.d. up to 500 r.p.m., where temperature does not exceed 200° F. Housings should allow an adequate volume of grease on both sides of bearings and be fitted with grease nipple. Fill housing about ⅔ full. Grease lubrication often assists the seal under wet or dusty conditions.

(b) *Oil.*—Oil lubrication is usually necessary for high speeds. For oil bath lubrication maintain the level to immerse lower half of bottom roller under normal r.p.m., a visual indicator being provided.

With splash lubrication, provide collecting pockets above bearing (Fig. 9) with hole communicating to the SMALL-ends of the rolls, so that centrigufal force will assist circulation. Do not feed oil to the large ends of rollers as centrifugal force will resist circulation through the bearing.

With Timken automatic circulatory system, oil channels provide a complete circulatory path through and round each bearing. The oil volume and circulation holes should be as large as

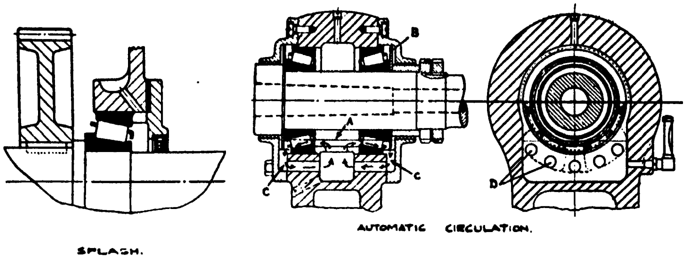


FIG. 9.—Lubrication Systems.

possible to obtain the maximum cooling effect (Fig. 9). This system is suitable for high speeds and saves cost of oil pump.

Other lubrication systems as the spray, oil pump circulation, and drip feed can be used.

Suitable oils are of the refined mineral type, free from acids and moisture and have a high flash point.

Light oils are used for high speeds, and *vice versa.*

Greases should be compounded from similar oils and high grade soap and contain no abrasive fillers, and oil should not separate on standing.

Water, acids, grit, etc., must be kept out of bearing housings to protect the highly finished bearings from corrosion and abrasion. Good seals are necessary, and can be of the rubbing or non-rubbing types.

Rubbing seals are available in a number of proprietary makes, the sealing elements being one or two rings of leather or synthetic rubber. Felt rings and stuffing boxes are also used.

Non-rubbing seals depend for their efficiency on close running clearances, labyrinths of various forms and flingers, combinations in many forms being possible. Some labyrinths are provided with nipples to ensure all spaces being filled with grease.

Applications of Tapered Roller Bearings.

There is practically no limit to the applications possible, these including automobiles, tractors, machine tools, locomotives, railway coaches, mining and steel works machinery, marine engines and thrust blocks, all types of war vehicles and vessels, gun mountings and fighting equipments, and all types of industrial machinery, etc.

Railway Axlebox and Crank Pin Applications, etc.

In addition to bearings, the Company manufactures railway axleboxes for steam, electric and Diesel locomotives, tenders, passenger coaches, electric motor coaches, wagons, etc. A few typical examples are shown (see Figs. 10, 11, 12 and 13). The boxes are made from high grade steel castings, arranged for Timken automatic oil circulation, and with hornguides faced with 11 to 14 per cent. manganese steel liners, which eliminate wear at these points.

Timken axleboxes are used on railways throughout the world, and require little attention beyond topping up with oil two or three times a year. Numerous examples are known of mileages considerably exceeding a million, with the bearings fit for further service. The starting effort is reduced by 85 per cent. compared with plain bearings, and the availability of locomotives and rolling stock greatly increased, as hot boxes are unknown, and oil consumption and maintenance costs are very low.

Timken Roller Bearings are also applied to the crank pins of steam locomotive driving axles, and to crossheads. Special lightweight reciprocating parts made from high quality alloy steel have been developed in connection with this type of application . which permits of better balancing of the locomotive, while reducing hammer blow at rail, and the destructive shaking and racking forces on the locomotive, and at the same time permitting an increase of 20 miles per hour to the maximum safe speed.

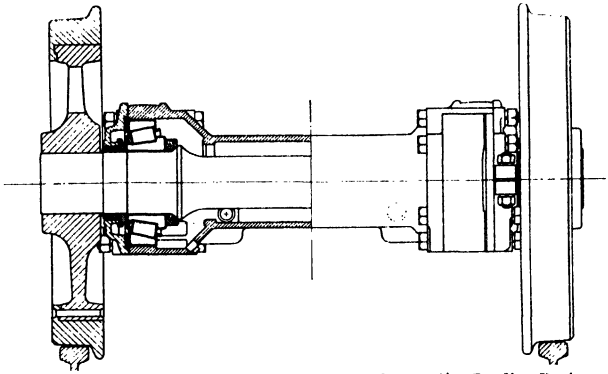


FIG. 10.—Solid type Cannon Axlebox—Steam Locomotive Leading Bogie.

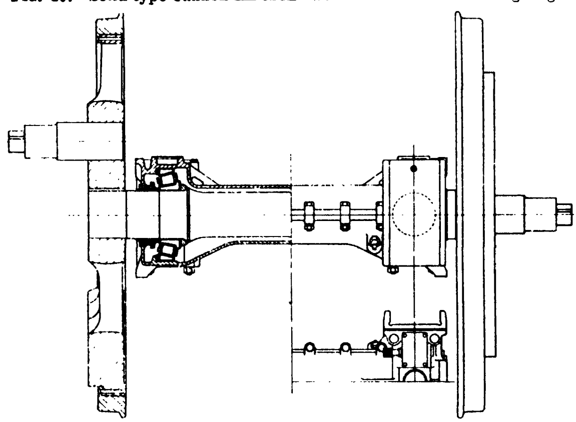


FIG. 11.—Split type Cannon Axlebox—Steam Locomotive Driving Axles.

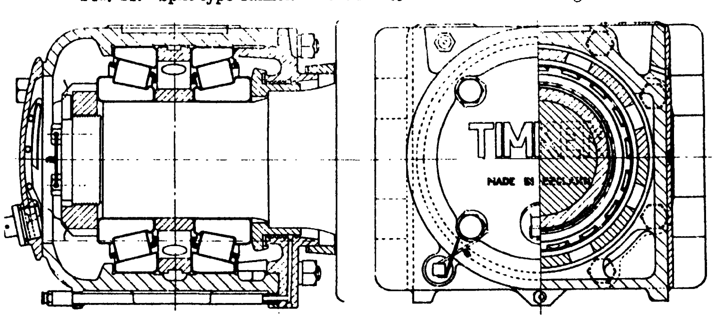


FIG. 12.—Outboard Axlebox—Steam Locomotive Trailing Trucks and Tenders.

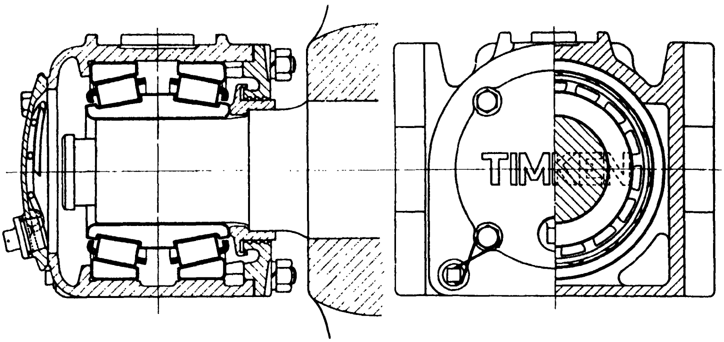


FIG. 13.—Outboard Axlebox—Electric and Diesel Locomotives, Motor Coaches, Passenger Coaches and Wagons.

ADDITIONAL DATA.

Designers and others requiring further information should refer to the technical literature (particularly the Timken Engineering Handbook) published by the Company, which includes also dimensions and data of the numerous types and sizes of bearings manufactured.

OIL RETAINING BRONZE BEARINGS.

(Bound Brook Bearings, (G.B.) Ltd., Trent Valley Road, Lichfield, Staffs.)

'Compo' oil-retaining bronze bearings are self-lubricating and suitable for practically all purposes for which ordinary machined bronze bushes and bearings are employed. These bearings are die-pressed from powdered metals and their composition is 90 per cent. copper and 10 per cent. tin—an alloy well known for its anti-frictional properties. They are porous to the extent of 25 per cent. to 35 per cent. by volume. The pores which are interconnected and distributed throughout the body of the bearing, are impregnated with a high grade mineral lubricating oil. As the result of capillary action between the oil and the cellular structure a thin film of oil is always present on the bearing surfaces. The application of bearing pressure, or rotational movement to the spindle, promotes a flow of oil from the pores to the bearing surfaces. This augmented oil film is maintained automatically during operation in proportion to any variation in the load, speed and temperature of the bearing.

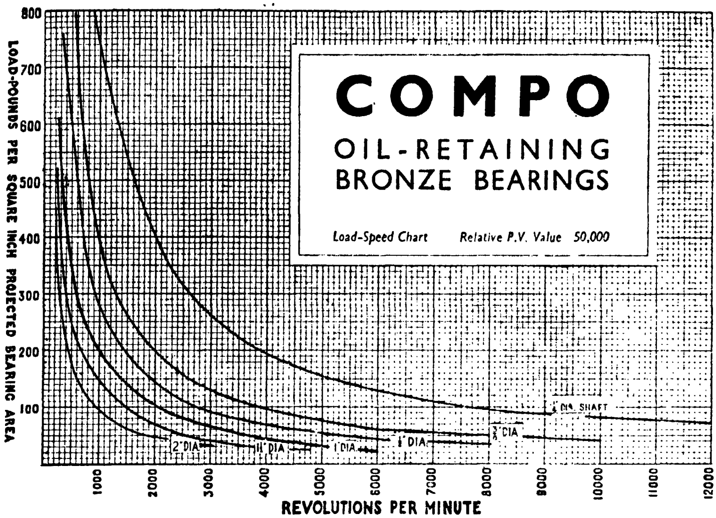
In calculating permissible loads, the conditions of operation, housing and its construction should be considered. Under ideal conditions of good heat dissipation or additional lubrication the PV factor of 50,000 utilised in the following chart may be exceeded. Where extreme conditions prevail as, for example, at high temperature, shock loading, dusty atmosphere, and lack of additional lubrication, the PV factor of 50,000 may be too high.

Because Compo bearings maintain an oil film between the shaft and the bearing, there is no theoretical limit of velocity. There are a number of installations in continuous day-to-day operation at 20,000 to 25,000 r.p.m. with small shafts. Where the PV factor approaches the maximum, bearing life expectancy will naturally be shorter than where the minimum factor is the basis. If, however, a provision is made to replenish the oil supply, the bearing will function indefinitely.

In general, it may be said that Compo bearings will carry a load equal to a normal bronze bearing, and because of their method of lubrication, will maintain an oil film to give a greater factor of safety.

In the following permissible load tabulation and chart, sufficient data are given to enable a rapid, general determination.

Velocity in Feet per Minute.	Permissible Loads.
Slow and Intermittent.	4,000 lbs. per sq. in.
25	2,500 " "
50–100	500 " "
100–150	325 " "
150–200	250 " "
over 200	per chart.



This chart may be used for general determination of permissible loads and allowable speeds for various shaft diameters. The figures are based on an empirical formula.

$$PV = 50{,}000.$$

in which formula P = the load per sq. in. of projected area.
V = shaft velocity in feet per minute.

Compo bearings are approved by the Air Ministry for use in aircraft for bearing loads up to 3,000 lb. per sq. in. for a slow and intermittent movement. Special impregnation with anti-freezing oil can be provided for applications involving very low temperatures.

Besides plain sleeve bearings, flanged bearings, spherical self-aligning bearings, thrust washers, strips and solids are available in sizes ranging from ⅛" diameter up to 3" diameter and up to 3" in length.

A comprehensive range of bearing sizes and lengths have been standardised for use with stardard bearings and shaft dimensions (see B.S.I. Specification 1131). Compo bearings are usually supplied to the finished dimensions required and ready for installing, hence no subsequent machining is normally required. The makers' recommendations should be followed where it is necessary to machine or ream the bearing surfaces of these bearings.

Compo bearings can be moulded directly to rubber, plastics and die-castings. In such cases, they are supplied without oil and impregnated after being moulded.

For the majority of applications the oil content of the bearings itelf is adequate to last the life of the machine. In special cases of high speed, high load or arduous conditions it is advisable to provide an additional supply of lubricant from a reservoir in the housing. The lubricant will then feed itself from the reservoir through the wall of the bearing to the bearing surface without the necessity of an oil-hole.

'Oilless' Graphited Bearings.

This is an accurately machined bronze bearing with grooves in it which are filled with a special graphite composition. The most suitable applications for the oilless bearings are those where oil cannot be employed and where conditions of load, high temperature or corrosive influences preclude the employment of an oil lubricated bearing.

This type of bearing is specially suitable for mechanisms on heating and drying ovens, also pulleys and rollers which have to operate at temperatures at which ordinary lubricants become carbonised. Other applications include bearings for chemical plant, food machinery and inaccessible parts of machine.

For continuous service a PV factor of 30,000 with a maximum speed of 500 ft. per minute or a maximum load of 1,000 lb. per sq. in. is suitable. For intermittent operation the PV factor may be as high as 60,000 with a load of 4,000 lb. per sq. in.

Bearing sizes ranging from ¼″ I/D up to 3″ I/D can be produced including flanged and self-aligning types.

Hoffmann Bearing Practice

(The Hoffmann Manufacturing Co., Ltd., Chelmsford, Essex.)

There is no need now-a-days to enlarge on the special place of ball and roller bearings in engineering practice. Their use in all mechanisms calling for free movement is taken for granted, and the general principles of their functioning are largely understood. Their purpose is to substitute rolling for sliding movement.

About the details of bearing application, some instruction may be helpful, as the rules governing the application of these bearings differ so much from those affecting plain bearings.

Basically, a plain bearing is merely a hole in a machine member in which a shaft revolves; or one in a wheel or pulley, carrying a shaft or axle on which it revolves. The necessary features are size, finish, correct alignment, provision for end location, and efficient means of lubrication. These essentials may be elaborated to any required extent as may be called for by the class of machine in question.

In ball and roller bearing practice things are different. Not only are such bearings required to operate with an almost total absence of power loss, but the rotating parts carried by them are to be retained in precise relation to one another throughout the machine's life.

There is still need for the above mentioned features, but in a greatly refined degree, and, in addition, complete protection from grit or dirt in any form. Sizes, finishes, etc., of the bearings themselves are the concern of their manufacturers, who provide full information on the tolerances of the bores and outside diameters, and make recommendations for seating and housing limits. These recommended figures provide suitable interference fits for rotating races of journal bearings, and push fits for stationary races.

It may be appropriate to give an extract from 'The Hoffmann Portfolio' on the various points to be kept in mind when mounting these bearings:—

'It should be remembered that all seatings on shafts or in housings must be perfectly cylindrical, that the surfaces forming abutments must be quite smooth and square with the shaft, and all recesses provided in housings for thrust races must be concentric with the shaft. Split housings should be avoided wherever possible, as a badly fitting cap may distort the races and cause local overload. Where conditions demand a split housing the cap must be registered or dowelled, and the seating for the bearing machined with the cap bolted in position. If the bearings are to be fitted in an aluminium housing it is desirable, where heavy loads are concerned, that the outer race should be housed in a liner of some harder metal. This is essential if the bearings are to be subjected to any considerable shock load.

'One of the most common causes of trouble with ball and roller bearings is creep or slow rotation of the races relative to the parts on or in which they are mounted. This creep cannot arise from friction in the bearings; it is purely the result of the conditions of working and can be eliminated if the following instructions are heeded.

'In thrust bearings creep is always due to the shoulders or abutments provided for the races not running perfectly square with the shaft. If the shoulder for the revolving race is not true it is the stationary race which creeps. Vice versa, if the stationary race is not square the revolving race will creep. Assuming the revolving race of a thrust to be out of square with the axis of the shaft, the whole of the load is carried by a small part of the race and consequently on one or two of the balls. As the shaft revolves this point of maximum load travels round the stationary race, giving it a tendency to creep forward on its seating. There is really an imperceptible wave movement, which becomes more pronounced the greater the inaccuracy. No matter how heavy the load or how high the speed, if the abutments or shoulders are square there should be no creep on thrust races.

'Whilst the underlying cause of creep (i.e., the travelling of a point of maximum load round the race) is precisely the same in journal bearings, it must be dealt with in an entirely different way. The conditions to be fulfilled by a journal bearing are such that the load can never be evenly distributed round the race. Taking the ordinary horizontal revolving shaft with a downward pressure due to the weight supported, the load will always be on one portion (of about 120 degrees) of the outer race, whilst on the revolving race this point of maximum load will be constantly changing as the shaft rotates. It will therefore be clear that it is the rotating race of a journal bearing which will have the tendency to creep.

'There should be no tendency for the stationary race of a journal bearing to creep, and it is for this reason that the stationary race of a ball journal bearing is chosen to be free endways to

permit of its aligning itself opposite the revolving member and thus preventing permanent end thrust. Where such a tendency to creep is pronounced in a stationary journal race, it is an indication that there is a recurrent change in the direction of the load, such as might be caused by lack of balance in the revolving mass. If the point of greatest load travels round the stationary race, the latter will roll round its seating, and at each complete revolution creep or lag will result to the extent of the difference in length of the two circumferences.

'The effect on the bearings of a machine resulting from lack of balance is not always fully appreciated. To secure steady running with a minimum of vibration, high-speed machines should be in balance at their normal rate of running—static balance is not sufficient to give the best results. In those cases, such as an eccentric, where an unbalanced load is inevitable, a roller journal bearing should be employed and both races made an interference fit.

'Given reasonable care in the machining of the seatings and in the fitting of the bearings, there is no difficulty whatever in avoiding 'creep' even with the heaviest loads. Keys and similar devices are quite unsuccessful and unnecessary, and quickly wear away under the constant chafing. They also introduce the risk of distortion of the races.'

This book, The Hoffmann Portfolio, gives much useful bearing information. Differing types of closure designed to give ample protection under varying conditions of service are illustrated. There is a long article on 'Methods of Calculating Loads on Bearings.' But the greater part consists of illustrated specimen mountings of ball and roller bearings in a wide variety; and these serve to show on the one hand the various points that need attention, and, on the other, the simple means by which all requirements may be met. For it should be emphasised that success in the use of these bearings is not difficult of attainment. The underlying principles are few and simple, and the machine equipped with intelligently selected and properly mounted ball and roller bearings is not only a more efficient machine at the outset—it remains so throughout its useful life.

But these subjects appeal more particularly to machine designers and makers. For machine users, who are concerned about the efficient running of the bearings. An article on 'Lubrication and Protection' is of special interest. It says, *inter alia* :—

'Oil is undoubtedly the most efficient lubricant, but the natural advantages offered by grease, which possesses both lubricating and permanent coating properties, render this form particularly suitable for use with ball and roller bearings. Grease also makes an effective closure between the shaft and housing, thus retaining the supply and excluding all moisture and dust, whereby the utmost economy and general cleanliness in operation are insured. The use of grease as a lubricating medium is to be recommended wherever practicable.

'A suitable grease should have a mineral base (lime or soda), but should contain no mineral acid, and be free from alkali or foreign matter. No filling agent should be employed, for the presence of such substances as graphite, talcum, etc., although in an extremely fine state of subdivision will give rise to lapping of the bearing parts, and so cause wear. Stability is of the utmost importance and the grease should show no tendency to gum, thin out, or separate into its constituents on standing or in service.

'For normal conditions, a good quality lime soap grease of medium consistency, and having a melting point of approximately 200° F. is recommended, and such a lubricant should withstand a continuous running temperature of 100/110° F.

'For conditions involving high speed or high temperature, a soda soap grease should be used. These lubricants usually have a melting point of 300/350° F. with a proportionately higher consistency, and considerable care should be exercised in their selection to secure the best results. The softer grease of this type will be found suitable for working temperatures up to 150° F., whilst those of higher consistency, which are usually termed high melting point greases will operate satisfactorily at temperatures up to 220° F.

'Oil lubrication should be applied in the following cases :—

(1) For light running parts where the resistance must be reduced to an absolute minimum.

(2) Where the bearings are completely enclosed in casings containing other parts for which oil lubrication is necessary, e.g., gear boxes, or when required to work in conjunction with plain bearings.

(3) Where the speed is unusually high, or the maximum is to be obtained. (For the above, use good quality mineral oil of light to medium viscosity.)

(4) For temperature conditions in excess of 220° F. (Use steam cylinder oil.)'

Attention to the above details is amply repaid. Proper mounting, proper protection, and proper lubrication will preserve ball and roller bearings almost indefinitely.

'Oilite' Self-Lubricating Bronze Bearings.

(*Manganese Bronze and Brass Co., Ltd., Handford Works, Ipswich.*)

'Oilite' Self-lubricating Bronze Bearings (oil-retaining bearings) are a product of powder metallurgy being manufactured from copper and tin powders, die-pressed and alloyed to form superior tin bronze with a microporous structure capable of retaining 30 to 35 per cent. of high-grade mineral lubricating oil. They are produced to precise dimensional accuracy and in addition to their good physical properties they possess the ability to supply a regulated flow of oil to the

journal according to the varying load, speed and temperature demand, and of maintaining at all times a uniform film of oil at the surfaces re-absorbing the oil as the load or temperature returns to normal.

For the majority of applications, the original oil content of an 'Oilite' Bronze Bearing is sufficient to outlast the life of its component, but where high speeds or other special conditions prevail, supplementary lubrication may be advisable which can readily be provided by means of a reservoir in the housing, so that as the original oil is used up, the bearing is replenished by capillary action, from the reservoir.

These bearings are suitable for practically all applications where ordinary bronze bearings are used but with the added advantages of a greater factor of safety and no lubricating problems to the user. They may be used for applications ranging from slow or intermittent movement up to high-speed rotation, and have been approved by the Air Ministry for loads of up to 3,000 lb. per sq. in. on oscillating movements, while in actual service they have proved successful at loads of over 4,000 lb. per sq. in. on slow or intermittent movements; at the other extreme they have been employed with good results in conjunction with spindles running at 30,000 r.p.m.

Tests are being carried out continuously under varying loads and speeds, and the following are actual results obtained, employing hardened and ground journals loaded on the lower surface and using no other lubricant than that originally contained in the test bearings.

Particulars of Test.		No. 4/3.	No. 1/12.
Bearing dia. × bore × length	(inches)	$1 \times \frac{3}{4} \times 1$	$1\frac{5}{8} \times 1\frac{1}{4} \times 1\frac{1}{2}$
Journal running speed	(r.p.m.)	2,100	1,100
Bearing pressure	(lbs./in.^{2})	80	60
Total running period	(hours)	1,500	4,010
Percentage oil content—initial	(by vol.)	28·0	27·3
„ „ „ —final	„	22·7	20·6
Total wear on bearing	(inches)	0·0005	0·0012
„ „ journal	„	nil	0·0002

The tests were concluded at the end of the predetermined period and not because of failure, for it will be observed that there was an adequate supply of oil still available.

The load carrying capacity of a bearing is a function of speed and load, and is a product of the two variables—feet per minute and pounds per square inch—usually expressed as P.V. For 'Oilite' this P.V. value for normal conditions of operation and without supplementary lubrication, is 50,000, but must be modified where shock loading is encountered and can be substantially increased with supplementary lubrication.

These bearings can be supplied for operations at temperatures down to minus 60° C. and up to 100°C. and impregnated with suitable lubricants to withstand such conditions. They are delivered fully impregnated and in a finished state, ready for assembly without any subsequent machining; the 'Oilite' Size List gives full particulars of the range of types and sizes available, including cylindrical, flanged, thrust and self-aligning bearings and washers, and, in addition, solid round rods, tubular blanks, rectangular strips and plates from which customer can machine special requirements.

Continued expansion in research and investigation in the field of powder metallurgy has produced new products including 'Oilite' bronze in the form of solid and cored bar and plate stock for machining, 'Oilite' lead bronze for seal rings and similar applications, and 'Oilite' super load, a material for bearings subject to extreme pressures, heavy shock load and severe atmospheric conditions. It should be noted that these new products possess similar self-lubricating (oil-retaining) properties as the original 'Oilite' Bronze.

A further new product is the 'Oilite' sintered metal filter, suitable for all types of filtration, flame arrestors, etc., possessing satisfactory physical strength properties. Structural parts in sintered iron and super load are now being developed where oil retaining properties are not a requirement, but where denser structure is necessary to give desired strength and to eliminate expensive machining operations.

R. & M. BALL AND ROLLER BEARINGS.

(Ransome & Marles Bearing Co., Ltd., Newark-on-Trent.)

The wide range of R. & M. ball and roller bearings and the multiplicity of sizes, offers a comprehensive range of bearings suitable for every application. These are fully described in the catalogues, which will be gladly supplied on request by the makers. The sectional drawings

shown below illustrating various types of bearings, and indicating their suitability for thrust or journal load, are only inserted as a guide.

In the development of these various series of bearings Ransome & Marles Bearing Co., Ltd., have played a major part, and the adaptability of some of the series to speeds as high as 90,000 r.p.m. is due to the investigations which this firm have carried out. High speeds such as that quoted are only obtainable by the use of carefully balanced cages made of lighter materials than is the normal accepted practice. The experiments revealed that success was obtainable when cages made of duralumin or bakelite were employed.

Additionally, although the normal tolerances for ball or roller bearings are fine, for these exceptionally high speeds the limits of accuracy must be approximately half those accepted for standard bearings. This, together with methods of preloading the bearings, whether this is carried out in the bearing itself, or within the application, has resulted in the wide adoption of R. & M. bearings for grinding spindles and other machine tools, where precision and high speeds are essential to modern practice.

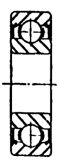
Another feature of R. & M. bearings is their ability to withstand exceptionally heavy loads Rolling mills, lifting bridges and rolling stock, being a few instances where these bearings are working under arduous conditions, and are in many instances subjected to severe shock loads.

Experience shows that wherever a shaft revolves or oscillates R. & M. bearings can be used to advantage, and this firm would be pleased to demonstrate this by supplying, on receipt of details, application drawings showing guaranteed arrangements. This service is, in common with all the technical service of Ransome & Marles Bearing Co., Ltd., placed at the disposal of users without obligation.

TYPES OF R. & M. BEARINGS.

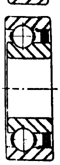
SINGLE-ROW BALL JOURNALS

For journal loads but will withstand end thrust.



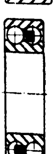
BALL JOURNALS WITH SHIELDS

A range of ball journal bearings are supplied fitted with shields to one or both sides of the bearing.



BALL JOURNALS WITH SEALS

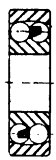
A range of ball journal bearings are supplied fitted with seals to one or both sides of the bearing.



DOUBLE-ROW BALL JOURNALS

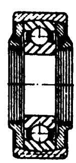
(Rigid type)

For journal loads.



DOUBLE ROW SELF-ALIGNING BALL JOURNALS

For journal loads where misalignment is present.



SELF-ALIGNING BALL JOURNALS IN HOUSINGS

For journal loads but will withstand end thrust.

Other types can be supplied fitted with spherical seatings.

SINGLE-ROW DOUBLE-PURPOSE BEARINGS

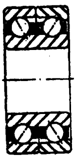
For combined journal and thrust loads (thrust in one direction only).

DUPLEX DOUBLE PURPOSE BEARINGS

For thrust in either direction—can be used for combined journa and thrust loads.

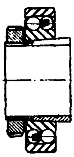
Types of R. & M. Bearings continued on page 1144.

TYPES OF R. & M. BEARINGS—Contd.



DOUBLE-ROW
DOUBLE-PURPOSE BEARINGS

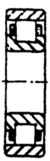
For combined journal and thrust loads (thrust in either direction).



ADAPTER BEARING
DOUBLE ROW BALL

For journal loads.

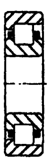
Other types can be supplied fitted with adapter sleeves.



ROLLER JOURNAL
BEARINGS

(Two shoulders in inner race and plain outer race.)

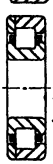
For journal loads only.



ROLLER JOURNAL
BEARINGS

(One shoulder in outer and two shoulders in inner race.)

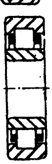
For journal loads and location duty in one direction only.



ROLLER JOURNAL
BEARINGS

(With lip and loose side plate on outer and two lips on inner.)

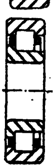
For journal loads and location duty in either direction.



ROLLER JOURNAL
BEARINGS

(With two shoulders in outer and plain inner race.)

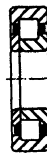
For journal loads only.



ROLLER JOURNAL
BEARINGS

(Two lips in outer and one lip on inner.)

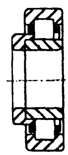
For journal loads and location duty in one direction only.



ROLLER JOURNAL
BEARINGS

(With two lips in outer and one lip and loose side plate on inner.)

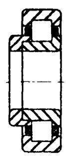
For journal loads and location duty in either direction.



ROLLER JOURNAL
BEARINGS.

(Two lips in outer, parallel inner and flanged side plate.)

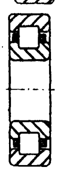
For journal loads and location duty in one direction only.



ROLLER JOURNAL
BEARINGS

(Two lips in outer, one lip and flanged side plate on inner.)

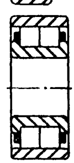
For journal loads and location duty in either direction.



ROLLER JOURNAL
BEARINGS

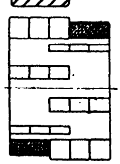
(With two lips in outer and two lips on inner.)

For journal loads and location duty in either direction.



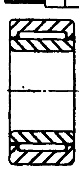
DOUBLE-ROW
ROLLER BEARINGS

For journal loads only.



ROLLER BUSHES

For journal loads only.



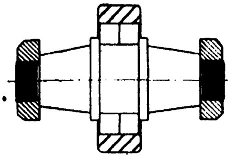
NEEDLE ROLLER
BEARINGS

For journal loads only.

(Needle rollers can also be supplied in cartridge housings.

Types of R. & M. Bearings continued on page 1145.

TYPES OF R. & M. BEARINGS—Contd.



ROLLER BEARING CRANKPINS

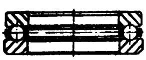
For internal combustion engines.

SELF-ALIGNING ROLLER BEARINGS

For journal loads where misalignment is present. Other types can be supplied fitted with spherical seatings.

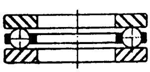
SELF-ALIGNING ROLLER JOURNALS IN HOUSINGS

For journal loads where misalignment is present and for location duty in either direction.



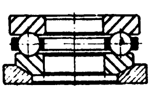
SINGLE THRUST BEARINGS

For thrust in one direction.



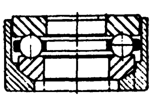
FLAT TRACK THRUSTS

For thrust in one direction.



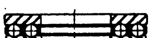
SINGLE SPHERICAL THRUSTS

For thrust where misalignment is present.



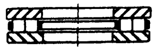
SINGLE SPHERICAL THRUSTS IN HOUSINGS

For thrust where misalignment is present.



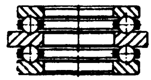
DOUBLE ROW BALL THRUSTS

For exceptionally heavy loads.



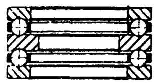
ROLLER THRUST BEARINGS

For exceptionally heavy duty at slow speeds.



DOUBLE THRUSTS

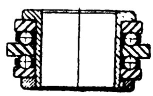
For thrust in either direction.



DOUBLE THRUSTS

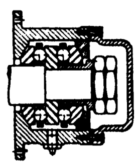
(centre washer locating).

For thrust in either direction.



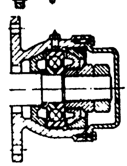
DOUBLE THRUSTS WITH SLEEVES

For thrust in either direction.



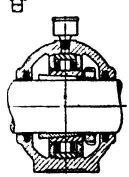
DOUBLE THRUSTS IN HOUSINGS

For thrust loads in either direction.



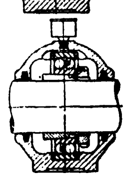
DUPLEX DOUBLE PURPOSE BEARINGS IN HOUSINGS

For thrust in either direction.



ROLLER LINESHAFT BEARINGS

For the heaviest loaded lineshafting.



BALL LINESHAFT BEARINGS

For light or medium loaded lineshafting.

The Selection of Ball and Roller Bearings.

(The Skefko Ball Bearing Co. Ltd., Luton, Bedfordshire.)

In selecting a bearing for a particular application it is necessary to have a thorough knowledge of the conditions existing in each specific case and of the characteristics of the different bearing types. Although no hard and fast rules can be laid down concerning the matter of bearing selection certain observations can be made about selection with regard to type. For lightly loaded high speed applications, ball bearings are generally most suitable. For large and heavily loaded applications, only roller bearings are satisfactory. If there is any likelihood of misalignment between shaft and bearing housing, a self-aligning bearing is desirable. If completely free axial displacement of the shaft between certain limits is necessary, a cylindrical roller bearing should be used. For applications where high speeds and relatively heavy thrust loads are encountered, a deep groove ball bearing may be the best choice. Used extensively in the automobile industry, taper roller bearings offer certain advantages in dealing with combined radial and thrust loads, especially where shock loads are met with.

The most important factors influencing bearing selection are discussed briefly in the following paragraphs.

The magnitude of the load, together with considerations of shock load and available space, has a primary influence on the bearing size, and also influences to a great extent the choice of bearing type. For small loads a ball bearing is generally used; for heavy loads a roller bearing is frequestly the only type possible. In different bearing types the ratio of radial and thrust load capacity varies considerably; selection must therefore be made with due regard to direction of loading. For pure radial load, any radial bearing could be used to advantage, and selection would then be made after consideration of other factors. With combined radial and thrust loads, after determining the magnitudes of the respective loads, consideration can be given to whether a single bearing can carry the combined loads or whether a bearing should be fitted to deal with each load component.

The speed of rotation in most cases only influences the bearing selection if it is unusually high. The speed at which a bearing may be run is determined, as a rule, by the temperature rise which is dependant on several factors. Inasmuch as radial ball bearings and cylindrical roller bearings have very low frictional characteristics under radial load, they can be used at higher speeds than other types. In addition, the material and design of the cage has some influence at high speeds. Therefore, in exceptional cases it may be necessary to employ cages of special material and design.

Fits, mounting and maintenance do not affect bearing selection in a direct way, but indirectly are of importance. For example, if the shaft of an electric motor expands and contracts owing to any considerable temperature variation, and the operating conditions are such that the bearings require a tight fit in the housing as well as on the shaft, one bearing must be of a type that permits axial movement. A common arrangement for such an application is a deep groove ball bearing at one end, and on the other end of the shaft (usually the drive end) a cylindrical roller bearing with rings that can be displaced relatively to each other. If the machine is of a type that is frequently assembled and dismantled, open bearing types such as cylindrical roller bearings, taper roller bearings or separable ball bearings (magneto bearings) may be necessary. In addition, tapered adapter and withdrawal sleeves help to solve some mounting problems. From the practical point of view, machines that operate under conditions of infrequent lubrication and inspection, should preferably be fitted with ball bearings as they are usually less demanding in these respects.

Standard bearings should be adopted as far as possible, and special types avoided. Apart from questions of cost and delivery time, special bearings present a very real difficulty when replacements become necessary.

The range of SKF bearings is such that the correct type and size of bearing for practically every application can be supplied. In the following are mentioned the types in most common use, together with brief details of the conditions for which they are most suitable.

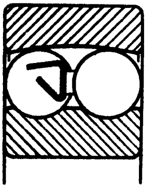


Fig. 1.

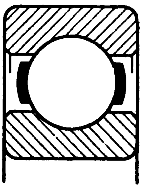


Fig. 2.

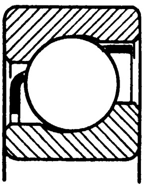


Fig. 3.

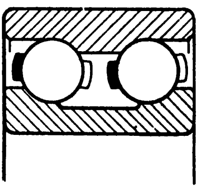


Fig. 4.

The Double Row Self-Aligning Ball Bearing (fig. 1) has two rows of balls which run on a common spherical track in the outer ring. This design allows the inner ring, together with rolling elements, to align itself freely about the bearing centre, and to accommodate a certain degree of misalignment caused by frame distortion and shaft deflection.

The Deep-Groove Ball Bearing (fig. 2) gives the maximum possible support to the balls by virtue of its deep uninterrupted race grooves, thus providing comparitively great radial and thrust load carrying capacity. This bearing is suitable for applications where combined loads within its capacity are encountered, for high speed work, and as a locating bearing.

The Single-Row Angular Contact Ball Bearing (fig. 3) has large diameter balls, a definite contact angle and high shoulders on the rings, giving good radial and one-direction thrust capacity. It can be employed with advantage where the thrust is greater than can be dealt with by a single-row deep-groove bearing, but where the use of a taper roller bearing is hardly justified.

The Double-Row Angular Contact Ball Bearing (fig. 4) is similar to the single-row type and is characterised by its large diameter balls, high shoulders on the rings and contact angles giving maximum radial and thrust capacity in both directions. It is outstandingly suited to positions where radial and axial rigidity is required.

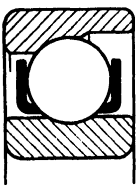


FIG. 5.

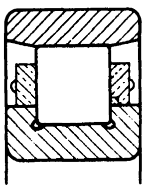


FIG. 6.

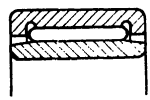


FIG. 7.

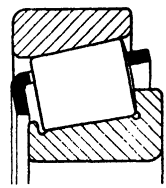


FIG. 8.

The Separable Ball Bearing (fig. 5), or magneto bearing, is a single-row type with a somewhat shallower groove on the inner ring than the deep-groove ball bearing. The outer ring has only one shoulder so that the bearing is ' open ' or separable. The bearing capacity is rather low but it offers certain advantages for easy mounting in magnetos, etc.

The Cylindrical Roller Bearing (fig. 6) has large diameter rollers, with rings having high guiding flanges on the inner or outer ring. It is suitable where radial loads are heavy and where end thrust is taken up separately. Certain types are available with lipped outer ring for location purposes where the axial load is small and does not warrant the use of a separate thrust bearing.

The Needle Roller Bearing (fig. 7) can be considered as a variant of the cylindrical roller bearing. Though less efficient than normal cylindrical roller bearings, this type may be used with advantage particularly where there is little room diametrically.

The Taper Roller Bearing (fig. 8) has straight tapered rollers, a sphered surface on each roller head, and a corresponding spherical flange face on the cone ensuring perfect guiding of every roller. This bearing is suitable for dealing with combined radial and thrust loads where good alignment is assured.

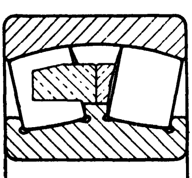


FIG. 9.

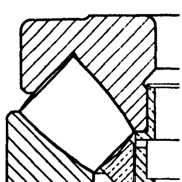


FIG. 10.

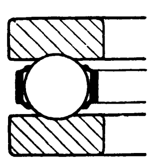


FIG. 11.

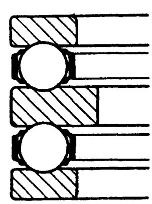


FIG. 12.

The Spherical Roller Bearing (fig. 9) is a double-row design, both rows of rollers having a common spherical track in the outer ring, so that the bearing is fully self-aligning in the same manner as the self-aligning ball bearing. Since the load is always evenly distributed between the two rows of rollers, and both rows have a maximum number of rollers, this bearing has the greatest possible capacity in a given space. It is suitable for the heaviest radial loads, has a considerable thrust capacity, and can withstand very severe shock loads ; it is, in short, the pre-eminent bearing for all heavy duties.

The Spherical Roller Thrust Bearing (fig. 10) is a variant of the spherical roller bearing with an exceptionally large contact angle. Unlike most thrust bearings it is fully self-aligning within itself, and is also capable of carrying a degree of radial load. It is the ideal bearing for heavy thrust duty.

The Single Thrust Ball Bearing (fig. 11) is designed for dealing with end thrust in one direction only, and is used in cases where the axial load is too heavy to be carried by a radial double-purpose bearing. Certain types are made with spherical seatings and seating rings to compensate for errors of alignment.

The Double Thrust Ball Bearing (fig. 12) is similar to the single thrust ball bearing, but can carry thrust in either direction. Again, types are made with spherical seatings and seating rings to compensate for misalignment.

Practically every bearing problem can be solved satisfactorily by careful selection from the SKF range of ball and roller bearings of a type and size to suit the conditions of load and speed. Engineers are aware, however, that the correct choice of bearing does not in itself ensure the high efficiency and long bearing life it is possible to obtain. Shaft and housing designs, limits and fits, methods of handling and fitting, lubrication, and other relative factors, must be thoroughly understood and the knowledge applied

SECTION XX

PART II

Lubricating Oils and Lubrication.

Graphite Lubrication.

(*Graphite Products, Ltd., 52 Battersea Church Road, S.W. 11.*)

Graphite is, chemically, carbon, but unlike the diamond, which is also carbon, the crystalline structure consists of hexagonal plates, which readily break apart, forming flakes. These flakes will range from ¼-inch or more across, when exceptionally large, to those of ultra-microscopical dimensions. Generally it may be said that under suitable grinding conditions the flaking process will continue indefinitely.

If graphite is squeezed between two adjacent surfaces, such as a journal and a bearing, the smaller flakes will be caught and fixed in the interstices, leaving the larger flakes to form a highly polished and unctuous surface. Conventionally, it may be illustrated as in figs. 1 and 2 (p. 1149).

From observation of the diagrams it is clear:

(1) That in the case of fig. 1 *starting friction is heavy.*
(2) That if pressure between faces is sufficient *seizing will take place.*
(3) The presence of oil or grease in the interface *does not necessarily prevent these results.*
(4) The presence of flake graphite forms a *smooth, greasy* and *unctuous surface.*
(5) *Starting friction is very small.*
(6) Prevents *metallic contact, so seizing cannot take place.*
(7) *Stops wear of metal.*
(8) If oil is used with the graphite *less oil is needed.*

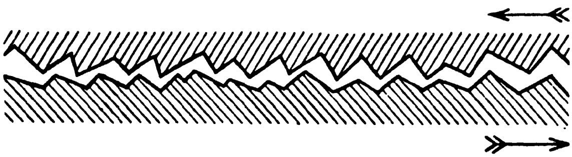


FIG. 1.

Another advantage of graphite not shown by the foregoing, is that graphite has a conductivity of 0·0117 against mineral oil of 0·0003, so that:

(1) Bearings are kept cool.
(2) Greases which without graphite might melt and flow out of the bearing are prevented from doing so by addition of graphite. It should be added that capillary attraction between the flakes also helps to this end.

Proof in experiments of these points is of interest:
Friction between dry, smooth and clean metal surfaces before and after graphiting.

STEEL AND BABBITT METAL.	
Before graphiting	$\mu = 0{\cdot}515$
After graphiting	$\mu = 0{\cdot}140$
STEEL ON STEEL.	
Before graphiting	$\mu = 0{\cdot}365$
After graphiting	$\mu = 0{\cdot}157$

Authority: Koethen.

These are very high coefficients in any case, but the reduction is remarkable. It must be remembered, too, that, except where oil or grease are impracticable, dry graphite is not used alone but always as a means of improving the properties of the usual lubricants.

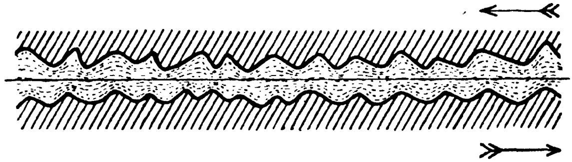


FIG. 2.

The effect of the use of 'Foliac' Lubricating Flake Graphite with a lubricating vehicle such as an oil or grease is:

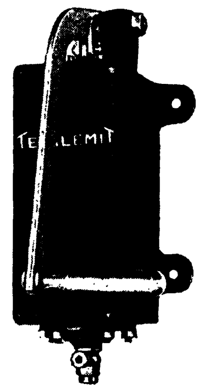
(1) To increase the efficiency of the lubricant and reduce friction. (10 per cent. to 40 per cent. reduction of friction are not unknown.)
(2) To increase the permissible bearing pressures. (Sometimes as much as 100 per cent.)
(3) To enable easy starting.
(4) To enable less lubricant to be used for a given service.

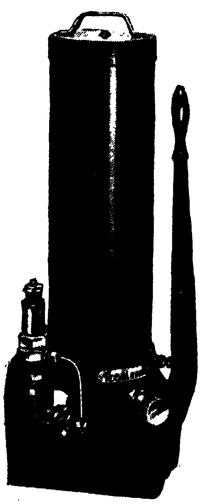
The proportions of 'Foliac' Pure Flake Lubricating Graphite to oils and greases to give the optimum results will vary considerably, and consequently it is advisable for those who desire the best results to buy 'Foliac' Graphite Lubricants, which are made for all classes of work, rather than buy the graphite alone, unless they have had previous extensive experience in the making of graphited lubricants.

Tecalemit 'Bijur' Metered Lubrication System for Use with Oil.

(Tecalemit, Ltd., Great West Road, Brentford, Middlx.)

This is claimed to be an essentially up-to-date method of central lubrication for almost any type of machine. Its features are characteristic of Tecalemit industrial equipment and comprise a 'Bijur' oil pump, hand operated or driven from the machine itself, and a single supply line which can distribute to a considerable number of points fitted with meter valves of widely varying capacities to satisfy the requirements of individual bearings. Installation is thus simple and inexpensive. The correct oil film is maintained automatically, making this system particularly applicable to textile and other plant where over-lubrication is detrimental, yet ensuring that the bearing furthermost from the pump is fully supplied. Clean oil of practically any viscosity may be used with 'Tecalemit-Bijur' equipment without adjustment, metered lubrication being automatically maintained under all operating conditions.





Tecalemit 'Line-o-Matic' Lubrication System for Use with Grease.

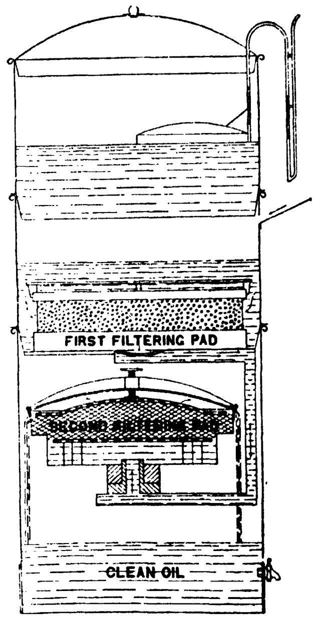
The makers state that this is a simple centralised system applicable to almost any machine requiring heavy oil and soft grease lubrication. The essential components include the Line-o-Matic Pump, the single supply line and the bearing automatic injection. The pump can be manual, as illustrated, or an automatic power-driven model. The single supply line provides easy and economical installation, and is capable of distribution to a considerable number of bearings. The automatic injector situated at each lubrication point is calibrated to supply the correct injection to suit individual requirements, thus the possibility of excessive lubrication or partial-starvation are each eliminated. The Line-o-Matic system, it is claimed, ensures complete lubrication of the machine in one pump-cycle, giving both positive assurance and complete safety to operators.

Recovery of Lubricating Oil.

(*A. C. Wells & Co., Ltd., Mount Street, P.O. Box 5, Hyde, Cheshire.*)

These filters consist of three compartments, any of which can be readily removed from the other to facilitate cleaning.

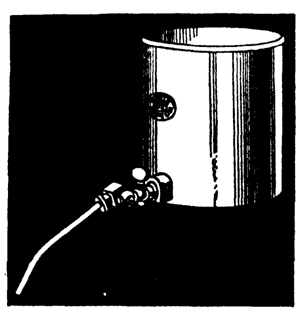
The top chamber receives the dirty oil, which after admission is allowed to settle for about a day, the time varying with the degree of foulness attained by the oil. The top chamber is provided with a floating syphon which when set in operation delivers the settled oil drop by drop into the second compartment, whence it passes by gravity through a special filter pad into the lowest compartment. The descending oil is collected off the underside of the filter pad in the second stage and led into a tube which terminates in a further filter pad. This cleans the oil by upward filtration, the recovered oil falling into the bottom of the third, or lowest, compartment, whence it is drawn off for re-use as required. By heating the oil with steam coils or electric heaters the capacity of the filter can be practically doubled.



Wells' Lathe Cans.

(*A. C. Wells & Co., Ltd., Mount Street, P.O. Box 5, Hyde, Cheshire.*)

Wells' lathe cans are available in two sizes: 2-pint and 1-gallon. Made of iron, and soundly constructed, these lathe cans will give years of hard service. Both sizes are fitted with brass universal fittings.



SECTION XXI

PART II

Belting—Belt Driving—Clutches.

Belting.

R. & J. Dick, Ltd., Greenhead Works, Glasgow, S.E.)

During the present restriction of the supply of raw materials, it is of the utmost importance that the care and maintenance of belting should be attended to.

The following points should enable belt users to get longer and more useful life from their belting.

Never allow dirt or other matter to collect on the surface of the belt. Such dirt forms into lumps, tearing or distorting the driving face of the belt, and preventing it from forming proper contact with the pulley.

Care should be taken to prevent oil from getting at the belt. It will cause loss of power by slip, and it is particularly harmful to Balata and cotton belts, tending to separate the plies. It will also rot a leather belt and rapidly lessen its efficiency.

Chemical action caused by bad air in close and badly ventilated rooms will damage Balata belts. An occasional application of castor oil used sparingly on both sides of the belt will combat deterioration from this cause.

Belting should not be run too tight, it reduces power and wears it out.

See that belt is running true, if not check up the alignment of shafting and pulleys.

Never run a thick belt over a small diameter pulley ; thin belts grip best and transmit more power.

An overloaded belt will cause it to slide from side to side of pulley. Cure—thicker or wider belt, or fit extra belt on top of existing one.

Slip is shown by polished pulley. In the case of a Balata belt a few drops of castor oil applied to the face of pulley will probably cure this. If the belt still slips it should be shortened.

Clutches.

Coil Clutch.

(The Coil Clutch Co., Ltd., Phœnix Works, Johnstone, Renfrewshire.)

The clutch shown in fig. 1 consists of a bushed driving plate (*c*) mounted on a shaft and having lugs cast on its face to engage the transmitting coil (*a*). A drum (*b*) of chilled material

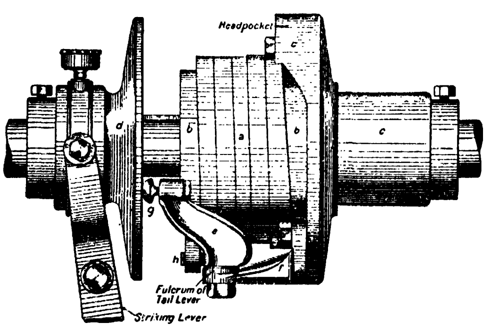


FIG. 1.

is keyed to the shaft adjacent to the driving plate. The transmitting coil (*a*), of special steel, having a highly-finished internal surface, is mounted on the drum and has one end engaged with the driving plate lugs, whilst the other is free to be operated. A sliding member (*d*) is mounted

on the shaft beyond the drum, its function being to pull the free end of the coil through a steel lever (*e*), which is pivoted on an extension of the driving plate (*c*), and engages the free end of the coil.

In engaging the clutch, the plate (*d*) is moved towards the drum, thereby tilting the lever (*e*) and tightening the free end of the coil, which is contracted and gradually tightens on to the drum (*b*), causing the different members of the clutch to be locked together. The action is similar to that of a man tightening the free end of a rope on a revolving capstan.

The clutch, made by the Coil Clutch Co., Ltd., Johnstone, is simple in construction and compact, and is used on transmissions such as :—haulage gears, cranes, rolling mills, wire and tube drawbenches, and bending rolls and presses.

Brakes and Brake Linings.

'FERODO' BRAKE LININGS.

(*Ferodo Ltd., Chapel-en-le-Frith, Stockport, England.*)

Friction linings can be divided into two main types, cotton and asbestos base. The latter can again be sub-divided into metallic and non-metallic.

Ferodo Fibre linings are produced from a solid woven cotton base, specially impregnated; this treatment imparting exceptional qualities of wear resistance and a high coefficient of friction to the finished lining. Impregnation also raises the naturally low resistance to heat of the cotton base to the maximum recommended operating temperature of 220° F. The value of the coefficient of friction for design purposes is 0·5. This lining can be supplied up to 18 ins. wide and from ⅛ in. to 1 in. thick. It is applicable to all types of brakes and cone clutches in engineering practice, subject to the temperature limitation already specified.

Ferodo Fibre H.D. friction lining is similar to the material described above, but will satisfactorily resist working temperatures up to 350° F. Its coefficient of friction for design purposes can be taken at the same value of 0·5 and it is manufactured in the same range of sizes. It is, however, a rigid product and therefore cannot be supplied in roll form. For the same reason Ferodo Fibre H.D. linings must be formed to the final radius during manufacture and this is particularly necessary when this material is fitted to post brakes where each end is given a comparatively sharp reverse bend for anchorage purposes.

Asbestos base, metallic and non-metallic, linings are available in a variety of types to meet the requirements of particular applications.

Ferodo Bonded Asbestos lining is produced from solid woven asbestos, brass wire being introduced during weaving. It is bonded by an infusible process which gives it its characteristic coefficient of friction and wearing qualities. The coefficient of friction for design purposes should be taken as 0·33. It is a lining suitable for running against all types of cast-iron and against steels having a Brinell hardness of 200 and over. It is available in thicknesses from ⅛ in. to 1 in. and in any width up to 18 ins. and in rolls up to 50 ft. lengths or in die-pressed segments. These latter segments have the same friction properties as the roll material from which they are manufactured, but, as they are highly compressed, they give a considerable increase in lining life and the segments are mechanically stronger and more rigid. A large range of sizes of die-pressed segments are available from stock.

Ferodo M.Z. brake lining is an asbestos base solid weave metallic friction material, zinc alloy wire being introduced in the weaving operation. It is preferably supplied in segment form, shaped to definite dimensions as to length, width, thickness and radius. It can, however, be supplied in strip form for special applications. The coefficient of friction of Ferodo M.Z. lining for design purposes should be taken as 0·3. This material is intended to meet the requirement of a heavy duty brake lining having a constant medium coefficient of friction throughout its temperature range. It is suitable for running against all good quality cast-irons and cast alloy irons, and similar quality steels having a Brinell of 180 and over. Ferodo M.Z. lining is available from ¼ in. to 1 in. thick and in widths up to 18 ins.

Ferodo VG.91 is a rigid moulded material containing random asbestos yarn spun on brass wire. This friction lining has exceptionally high temperature resisting qualities and will function satisfactorily up to 1,000° F., providing the mating surface will withstand this very high level. Ferodo VG.91 has a moderate but slightly rising friction value with temperature, the cold μ for design purposes being taken at 0·25. This friction lining is an extremely rigid product, and must therefore be manufactured to its final dimensions with respect to length, radius, width and thickness. Preferably it should be used only against high quality cast or alloy irons, or if it is imperative to employ steel, the latter should have a Brinell value of not less than 200.

Ferodo M.R. friction lining is a solid weave, non-metallic, asbestos base material. In its normal form it is a hard, rigid brake lining and definite dimensions as to length, width, thickness and radius should be specified when ordering. In special circumstances this material can be supplied in flexible lengths to be fitted to band brakes, etc. Ferodo M.R. lining has a higher coefficient of friction than other asbestos materials, and for design purposes this value can be taken as 0·35. It is available in standard ranges of thicknesses from ⅛ in. to ⅝ in. and in widths up to 18 ins. It is suitable for use against all good quality close-grained cast and alloy cast-irons, and against similar quality steels of 150 and over Brinell. In common with all non-metallic

friction linings, Ferodo M.R. lining requires that the initial finish of the mating metal surface should be as good as possible; this is best attained by grinding.

The brake lining thicknesses given represent standard sizes, but materials beyond these dimensions are available for certain applications.

'Ferodo' Clutch Linings.

Cone Clutches.—Ferodo Fibre and Ferodo Bonded Asbestos linings are both available in cone form, the material being radially woven facilitating formation into cone linings and ensuring uniform density across the face of each lining. For clutch design purposes the coefficient of friction of Ferodo Fibre should be taken as 0·48 and that of Ferodo Bonded Asbestos as 0·28. The minimum included angle for Ferodo Fibre should be 36° and for Ferodo Bonded Asbestos 24°. Ferodo Fibre should not be adopted where the maximum operating temperature is likely to exceed 220° F.

Plate Clutches.—Ferodo Fibre linings are not suitable for fitting to plate clutches, as the temperature of the plates is generally above the safe maximum for this class of material. Several types of asbestos base linings are available in disc form for plate clutches, their construction differing according to their dimensions and the duty and conditions under which they are to operate.

Ferodo R.A.D.11 facings are manufactured from solid, radially woven, metallic (brass wire) asbestos, and are available in a large range of diameters, bores and thicknesses. There is no visible joint in these facings, this joint being cemented during manufacture, which obviates the weakness and other disadvantages of the more usual wire stitched butt joint. When dimensional requirements, particularly wide face widths, preclude the above facing construction, similar material is available in cut out form up to 21 ins. outer diameter. Beyond this dimension, each facing is supplied in four or more equal segments which are riveted to the metal carrier plate to form a continuous disc. These clutch facings should be run dry (unlubricated) and for design purposes the coefficient of friction should be taken as 0·28.

Ferodo R.A.D.5 facings are similar to Ferodo R.A.D.11 discs with respect to material and construction and are intended for use in oil immersed clutches. The coefficient of friction of these facings for design purposes when running in oil should be taken as 0·1.

Ferodo L.M. facings are manufactured from a metallic asbestos base, rigid moulded material which can be run dry or lubricated. The coefficient of friction dry should be taken as 0·22 and lubricated as 0·06. This material can be supplied in one piece discs up to 36 ins. diam., any bore and up to 1 in. thick. Being a rigid, moulded material, gear teeth or splines can be cut in the periphery, when it will function satisfactorily as both friction facing and clutch plate.

'Ferobestos' Products.

Ferobestos products are manufactured by Ferodo Limited for a very large range of applications which have little or no connection with the brake and clutch lining field. They are made from a rigid moulded material having an asbestos base and given the availability of suitable moulding tools, are produced in a variety of forms and sizes. Large quantities of non-metallic bushes are made in this material which has a relatively low dry friction, and therefore is particularly suitable for oscillating and sliding motions at comparatively slow speeds. This material is also completely resistant to the solvent action of any normal lubricant, including water, and a large range of chemicals and is therefore particularly valuable as a bearing material where it is completely submerged in a liquid which will attack orthodox bushes. There is a large range of Ferobestos materials, each type having its own particular outstanding qualities such as chemical or temperature resistance, high structural strength, low friction, etc., etc., and it is suggested that the makers should be approached for particulars of these new products wherever it is thought they may be usefully employed.

SECTION XXI

PART III

Wire Ropes.

Whitecross 'Contra-Lock' Non-Rotating Ropes.

(*The Whitecross Co. Ltd., Warrington.*)

With the considerable advance made in recent years in the production of mechanical handling plant, there has arisen an increasing demand for steel wire ropes which possess no tendency to rotate under varying loads.

Such ropes must necessarily be constructed of multiple layers of strands so balanced as to counteract the tendency to tighten or slacken on the core as the load is imposed or released with consequential rotational movement. Many such designs have been evolved, but most ropes of this type were liable to become distorted in service owing to uneven rate of wear on the various parts, which had the effect of upsetting the exact torsional balance essential to a strictly non-rotating rope.

To obviate these difficulties, the Whitecross Co. developed the 'Contra-Lock' Patent Construction which by imparting a self-tightening action to the rope automatically takes up the wear on the respective parts of the rope and thus maintains correct balance throughout the life of the rope.

Sections of standard non-rotating ropes to which the 'Contra-Lock' Patent principle has been applied with success are shown below.

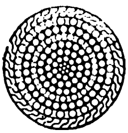


FIG. 1.

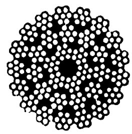


FIG. 2.

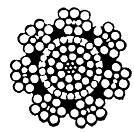


FIG. 3.

Fig. 1. 'Contra-Lock' Locked Coil Rope for high speed colliery winding.

Fig. 2. 'Contra-Lock' Multi-Strand Non-Rotating Rope for large balance ropes, sinking ropes and large crane ropes.

Fig. 3. 'Contra-Lock' Multi-Strand Non-Rotating Rope for light cranes and hoisting appliances.

In addition to the above, several special designs have been evolved incorporating the same principles, but adapted to meet unusually arduous or difficult conditions which the standard ropes proved unable to withstand. Included in these special deisgns is the Whitecross Co. Ltd.'s Koepe Winding Rope, which is in use at many of the largest collieries in Belgium. Special provision is made in this construction to obviate slip on the Koepe pulley and to reduce elongation under load to a minimum.

For small standard round or flattened strand ropes required to work, as is so oftent he case, in mechanical handling plant, round pulleys of small diameter and subjected to several reverse bends, it has been found that the application of the Whitecross preformed process has effected a material improvement in the life of the ropes. In this process the wires are first of all formed to the exact helical pitch of the strand, each strand then being treated in the same way to give the desired pitch before closing into the finished rope. This process eliminates all tendency to liveliness so that the load is evenly distributed on the individual strands and bending stress is alleviated. Further, should any of the outer wires break owing to excessive wear, they remain in place and therefore minimise wear on the other wires and on sheaves, pulleys and drums.

SECTION XXI

PART IV

CHAIN DRIVES.

(Morse Chain Co., Ltd., Letchworth, Herts.)

The essential difference between the Morse inverted tooth rocker joint chain and other forms of inverted tooth chain is the joint construction of the former. Briefly, instead of the round pin in bushes or liners utilised in other constructions, the Morse joint consists of a two-part pintle so shaped that the one part rocks or rolls on the face of the other as the chain goes on or off the wheel teeth. The pin elements are so held in alternate links that sliding friction, either between link and pin or between the two pins, is eliminated, and hence a very high degree of mechanical efficiency is assured, while, as a further consequence, chains with this joint construction are not so dependent on lavish lubrication as are those comprising the old form of round pin with its sliding surfaces between pin and bush.

The latest form of a Morse rocker joint inverted tooth chain is illustrated in section, fig. 1; while fig. 2 shows the position of the respective pins in a driving link plate, and fig. 3 shows the pins in their position relative to each other. It will be noticed that the bearing surface extends

throughout the width of the chain. The pins for all practical purposes form an integral part of the link, and hence the only motion which takes place at the joint is a rocking movement. These chains are made to work with wheel teeth of a shape to permit of exceptionally quiet running, even at high speeds; the drives complete being furnished by the Morse Chain Company, Ltd., to specific requirements which should indicate the h.p. to be transmitted, the speed of driving and driven shafts in r.p.m., the distance between shaft centres and the type of load. The introduction of the Morse rocker joint construction has rendered available the benefits of the inverted tooth chain to large powers, and such drives up to 5,000 h.p. in one transmission have been

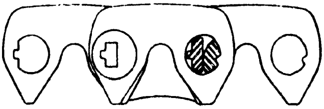


FIG. 1

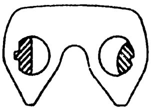


FIG. 2.

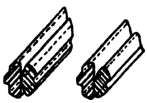


FIG. 3.

FIG. 1.—Rocker Joint Chain Section.
" 2.—Position of Pins in Link.
" 3.—Relative Position of Pins.

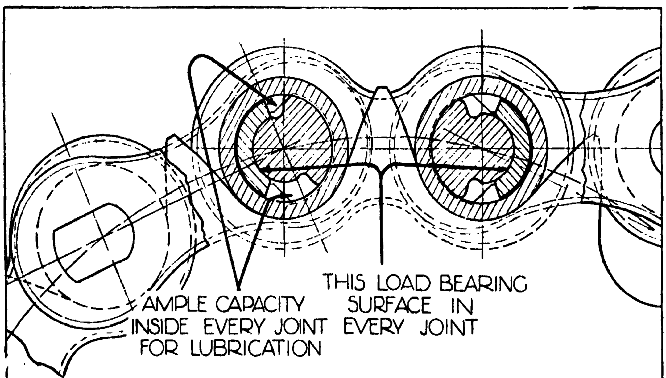


FIG. 4.—Morse Segmental Bush Roller Chain Section.

successfully furnished. These chains are manufactured up to 2-in. and 3-in. pitch to take care of such high power drives. The general applicability of a Morse Rocker joint chain is shown by the fact that it is made in numerous combinations of widths and pitches, the latter ranging from ⅜-in. to 3-in. pitch, suitable for rotary speeds ranging from 4,000 r.p.m. downward.

A further important feature in the field of power transmission by chains is the Morse patent roller chain. There are of course great varieties of power transmission for which, on account of

their slower speeds or the absence of demand for a high degree of quietness in running, the roller chain is eminently suitable. The distinguishing feature of the Morse roller chain again lies in the joint construction. As will be seen from fig. 4 the ordinary round bush has been eliminated and its place is taken by a segment secured to the inside links, while the pin is of such shape as to form a segmental portion, leaving a semi-circular portion to bear against the segmental bush. This pin is riveted into the outside links.

It will be seen from the above figure that the special features of this new roller-chain are as follows:—

1. *No Frictional Movement under Load, against Inside of Roller.*—As the roller and tooth engagement fixes the pin (or bushing) at that particular point, the bearing pressure is always taken on the *inside of the bush*, as the adjacent link with its pin (or bushing) flexes in engaging with the sprocket.

2. *Uniform Joint Action.*—In view of note 1, the frictional bearing surface is the same for every joint, *i.e.*, always between pin and bushing, consequently the rubbing velocity is proportionately less, owing to the smaller radius of contact.

In the conventional (round pin) roller chain, with the roller link fixed on sprocket, and the pin link about to engage, movement takes place between the *pin and bushing*; but with the pin link on sprocket and the roller link about to engage, movement takes place between *pin and bushing and bushing and roller*; consequently there is a different joint action during the engagement of each alternate link.

3. *Bearing Surface.*—In the round pin chain the bearing surface is virtually 'line contact' (owing to the necessary working clearance) whereas in the Segmental Bush Joint, a uniform arc of contact is secured, thereby reducing greatly initial 'bedding down' and chain elongation.

4. *Lubrication.*—In the case of the conventional round pin chain oil can only reach the pin via the small clearance between the pin links. In the case of the Morse Roller Chain, however, additional lubricant reaches the joint at the point of greatest clearance (between roller and link plate), and is at once in direct contact with the bearing surface. Moreover, the flow is greatly facilitated by the pumping action of the relative movement of pin and bushing sections.

The manufacture of both inverted tooth and roller chains, it will be appreciated, renders feasible the production of chain couplings consisting essentially of two sprockets, one on either of the shafts to be coupled, the sprockets themselves being connected by either an inverted tooth chain or a roller chain. Such couplings have outstanding advantages in that, while retaining a measure of flexibility, they offer exceptional ease of installation and entire dependability. The finished outer faces premit ready check for alignment, and wrapping the chain round the sprockets connects the coupling; disconnecting is equally simple, consisting merely in the removal of one of the chain pins, when the chain can be taken off. Fig. 5 shows the 50 h.p.

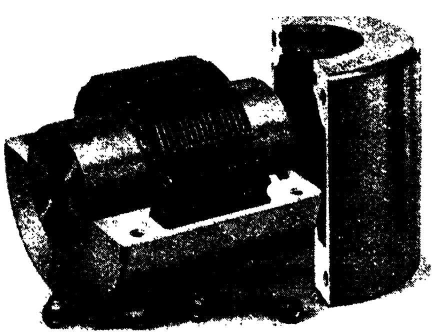


FIG. 5.—Morse Inverted Tooth Chain Coupling.

coupling at 1,440 r.p.m. Coupling sizes range from fractional h.p. to 1,000 h.p. or more, depending on the speed of shafts.

For protection, and to ensure lubrication, cases can be furnished totally to enclose the coupling, the case being filled with grease, thus assuring ample lubrication while excluding grit and dirt. Such a case is shown in fig. 5.

Renold Roller Chain.

(*The Renold and Coventry Chain Co., Ltd., Manchester.*)

The design of the Renold Roller Chain is based upon the use of simple cylindrical sections for the bearing pins, bushes and rollers: as a consequence it is possible not only to manufacture the parts themselves accurately, but the tools required for their production also lend themselves to a maximum degree of accuracy. For example, the holes in the plates are circular, and the bearing pins can be ground entirely on their working faces.

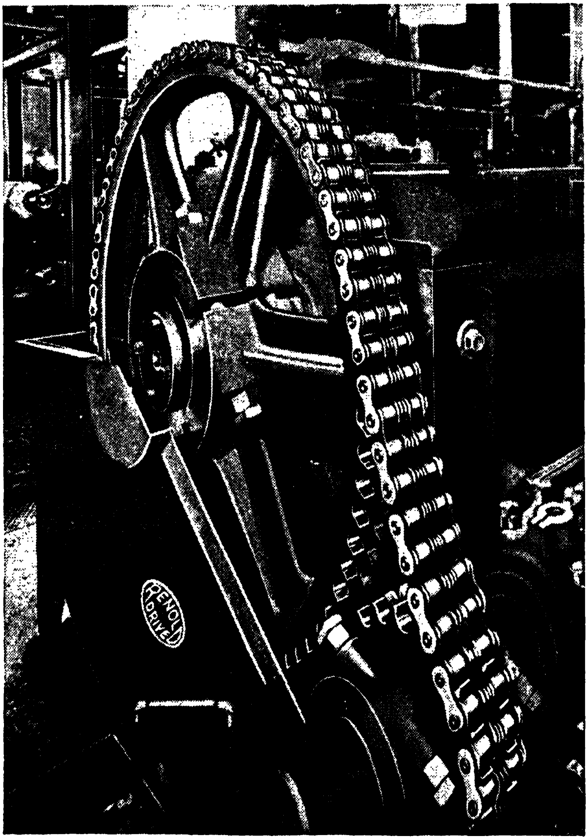


Fig. 1.—100 h.p. Renold Chain Drive 86·5 r.p.m. to 22·2 r.p.m. at Short Centres.

A major problem associated with the roller chain is how to obtain the necessary stability or fixation of the inner plates to the bushes, particularly in the case of the smaller pitches which

are required to run at very high speeds. Here again the advantage of the circular bush, which is the basis of the design, in allowing the necessary degree of force fit to be used to prevent lateral movement of the plates, will readily be appreciated.

Even so, the provision of the maximum amount of uninterrupted contact between the bush and the inner plate, which the circular bush achieves, requires to be supplemented by some positive method of attaching the plates if the chain is to operate successfully under ultra high-speed conditions. It was to deal with such conditions that the 'Mark 10' feature was introduced some

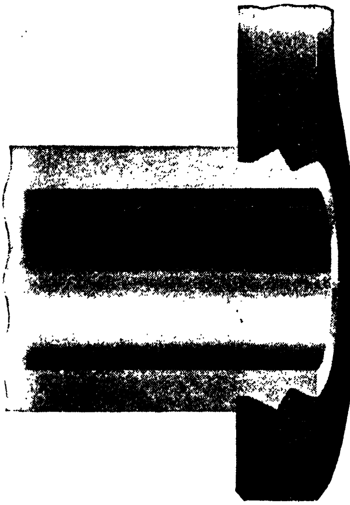


FIG. 2.—The keyed bush forced into the chain plate. The projecting keys prevent turning of the bush in the plate, and the metal filling the depressions prevents the plate being forced off endways.

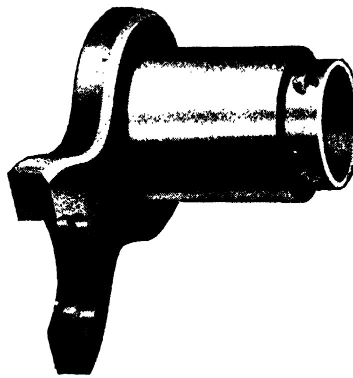


FIG. 3.—Keyed bush and plate, one end of plate cut away to show projections in the bore formed by the keys. These projections show the moulding of the metal by the keys.

few years ago. In the Renold 'Mark 10' chain the bushes are made with small keys pressed up on them and in the process of forming these keys small depressions are made in the surface of the bush in front of them. Hence, when the bushes are pressed into the plates, the metal removed by the keys is forced into the depressions in the bush and a compound lock is provided, making it impossible for the plates to be forced off.

The results of a series of tests of chain-drive efficiency carried out by the National Physical Laboratory show that the efficiency of a Renold chain drive, taken from stock, with no previous running, was 98½ per cent. The chain tested was a 1·0 in. pitch simple roller transmitting 25 h.p. from a shaft running at 900 r.p.m. on which a 23-tooth pinion was mounted to a shaft running at 363 r.p.m. carrying a 57-tooth wheel.

Before the advent of steel chains the choice between ropes, belts or gears was governed roughly by whether the deciding factors were positiveness, and perhaps compactness, in which case gears had to be used, or whether they were smoothness and elasticity, which pointed to the use of belts or ropes. The great achievement of the transmission chain is to combine to a remarkable extent these two sets of hitherto separated features. Chains are in fact as positive as gears, and as smooth running as belts and ropes, and have the highest maintained efficiency.

SECTION XXII

PART I

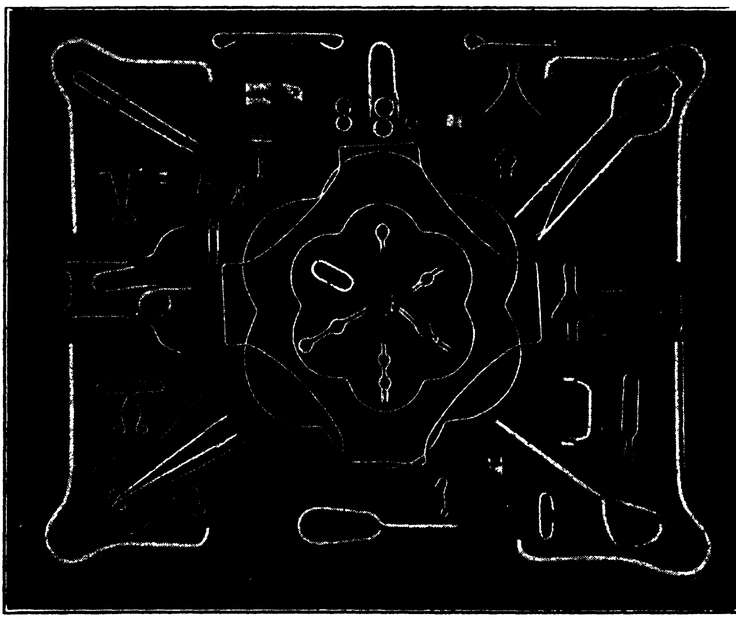
Machine Tools:

'HEENAN' AUTOMATIC MACHINES FOR THE WIRE AND STRIP INDUSTRY.

(Heenan and Froude Limited, Worcester, England.)

A number of machines are available for producing direct from the coil of wire or strip, a variety of components. These machines are :—

1. A four-slide Wire and Strip Forming machine (see illustration of various productions below).
2. A High-speed Nail Press.
3. A Chain Former, with separate Chain Welder.
4. A Ring Coiling machine, also capable of producing short coil springs.
5. A Hair Grip machine.
6. A Mattress Chain machine.



A small selection of the components which can be produced on 'Heenan' Wire and Strip Forming Machines.

The 'Heenan' Wire Forming and Strip Forming machines are designed on the basis of universal application and, when suitably tooled, can produce a very large variety of components direct from the coil of wire or strip. There is practically no metal industry which does not employ a variety of products which could profitably be produced on these automatic machines.

A large range of articles is at present being manufactured automatically on 'Heenan' machines, and the universal nature of these is such that one machine is capable of manufacturing automatically components which would normally require up to seven separate press operations.

In cases where production rates are such that a machine can be run for long periods uninterruptedly, the saving in production costs is frequently enormous, while where the consumption of one particular article can be catered for by running for a comparatively short period, the cost of installation is usually recouped very quickly by reduced production costs and floor space required, especially if sufficient different forms require to be made to keep the machine employed for a reasonable proportion of the year.

Each of the two types of machines is made in a range of sizes capable of producing components from 50 to 400 pieces per minute.

The working principles of the 'Heenan' Automatic Wire and Strip Forming machines are exceedingly simple and the tooling for different forms is fundamentally similar to that for ordinary power or hand presses ; each machine being a combination of several presses unified in a simple whole.

The 'Heenan' Automatic High-speed Nail Press produces headed nails from round or oval wire, the rate of output being very high.

The 'Heenan' Chain Former makes linked chain of the normal 'short-link' type, as used for cranes, etc., the operation being entirely automatic. The ends of the links are subsequently welded in the Chain Welder. Output is extremely high, the links being particularly well formed, and the welds of unusually good quality.

The 'Heenan' Ring Coiling machine produces rings with close-butted or overlapping ends, and by a simple attachment coil springs (compression or tension) can also be produced up to the maximum wire length which the machine will feed.

The 'Heenan' Automatic Hair Grip machine manufactures a standard form of grip, either plain or crimped, and with the ends sheared either square or rounded.

The 'Heenan' Automatic Mattress Chain makes the special form of chain used for spring mattresses, etc., and produces these in continuous linked-up lengths. A device is incorporated by which a link can be left unclosed at intervals, and the intervals themselves can be adjusted over a wide range.

'Rapidor' Sawing Equipment.

(Edward G. Herbert, Ltd., Manchester 19, England.)

'Rapidor' machines available in capacities from 6 ins. × 6 ins. to 12 ins. × 12 ins. are built for continuous high speed sawing. Using High Speed steel blades they will saw 28/30 tons tensile steel and brass at a cutting speed of 200 ft. per minute. Outstanding features are

Saw Frame is rigidly guided by two square slides placed diagonally for strength and to maintain alignment.

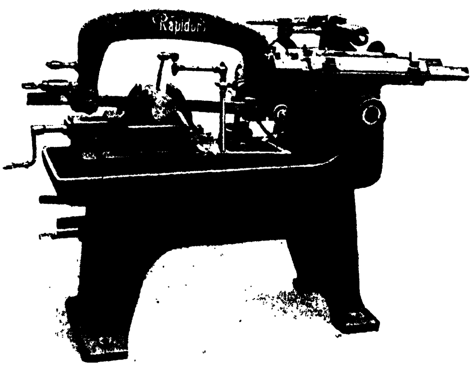
Saw Holders are rectangular, permanently in alignment and adjustable for different lengths of blades.

Tension.—The blade is quickly tensioned simply by turning a handle and a patented tension indicator shows when the blade is correctly strained.

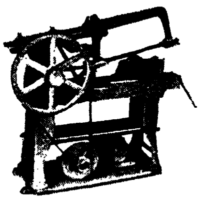
Pressure on the blade is regulated by a spring and a pressure indicator is fitted. Once the pressure is adjusted to suit the work being sawn no further alteration is required.

Starting.—A patent starting valve on the dashpot enables cuts to be started on a sharp corner without damage to the blade teeth.

Automatic Lift.—The saw blade is lifted automatically on the return stroke by means of a simple oil dashpot in conjunction with the angular setting of the saw.



'Rapidor No. 1 Sawing Machine.'
6″ × 6″ capacity arranged for single speed Belt Drive.



'Rapidor Manchester' Motor Driven Sawing Machine.

Length Gauge.—**An adjustable gauge is fitted on all 'Rapidor' machines.**

An adjustable automatic stop is incorporated to stop the machine at the finish of each cut.

Vice **is of the heavy screw type with roller thrust bearing on the screw. Loose jaw swivels to grip taper or irregular work and the whole vice can be adjusted on the T-slotted** table. Graduated swivel vice is supplied as an exrra.

Suds Pump **is of the gear type and has a positive roller chain drive. Iron piping with strong brass joints is provided for the suds.**

Lubrication.—**Grease gun lubrication ensures that all parts are perfectly lubricated.**

The' Rapidor Manchester' Sawing Machine **is a reliable and** inexpensive light **hacksawing machine, with a capacity of 6 ins. × 6 ins., and was introduced for use in small shops, maintenance and repair shops, etc., where sawing is not undertaken on a production basis. The essential and proved features of the heavier production 'Rapidor' machines—namely, double square steel slides, automatic relief on return stroke by oil dashpot, and the 'cutting on a corner' method of applying pressure—have been incorporated to ensure straight and trouble-free sawing.**

The firm is again manufacturing a full range of testing machines covering tension, compression, transverse, fatigue hardness and other tests; also static-dynamic balancing machines.

H.S.S. hacksaw blades, saw blade sharpening machines are also included in this firm's products.

SCREWING MACHINES (PRODUCTION).

(*Charles Winn & Co., Ltd., Granville Street, Birmingham,* 1.)

The illustration (fig. 1) shows the latest model heavy pattern high speed Tube and Bolt Screwing Machine, No. 4 size, the capacity being 1-in. to 4-in. tubes, and 1-in. to 4-in. Whitworth bolt

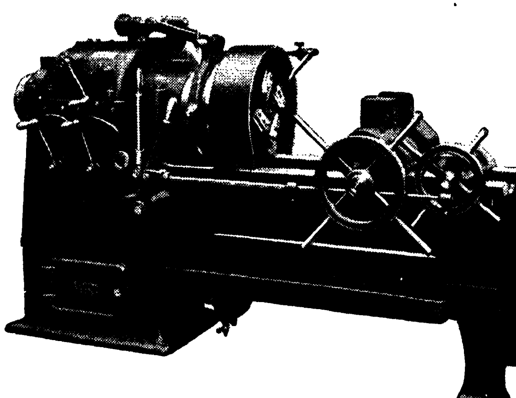


FIG. 1.

threads, and provided with tangential die head, with automatic opening and closing attachment also powerful open-jaw duplex-grip vice, with geared right- and left-hand vice screw, specially for gripping heavy work, and rack and pinion traverse with two machine-cut racks. A large oil tray and reservoir are fitted under the bed of the machine, with a self-priming rotary gear type oil pump at the back driven off the main driving shaft at constant speed.

The machine is fitted with single pulley drive and four-speed gearbox with forward and reverse movements, so that right-hand and left-hand threads can be cut with equal facility, or alternatively the machine can be provided with eight speeds for right-hand screwing only, the gears

being in heat-treated steel and running in an oil bath. A multi-disc friction clutch is provided for stopping and starting the machine, with clutch lever shown at the front of the headstock.

This machine is also made fitted with 5 h.p. four-speed A.C. motor and provided with two-speed gear, giving a total of eight spindle speeds. A reversing switch can be fitted to enable the eight spindle speeds to be used for either right-hand or left-hand screwing.

An interesting feature is the patent thread length indicator, fitted to the saddle, giving a clear visual indication of the length of thread being cut. This can be used independently or in conjunction with the automatic opening and closing diehead attachment.

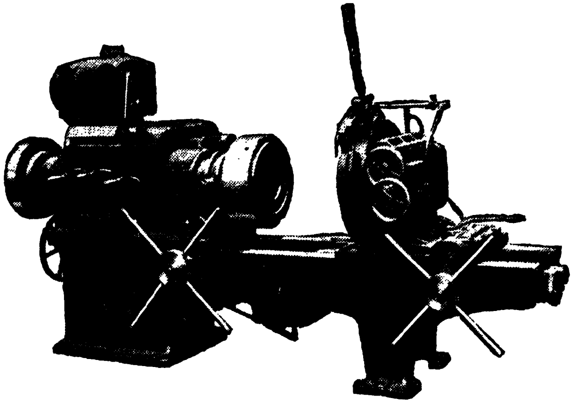


FIG. 2.

Fig. 2 shows an electrically driven high speed combined Screwing and Cutting-off Machine, No. 6 size, for tube works requirements having Winn's patent quick-grip self-centring chuck and hollow spindle for gripping and releasing the tube without stopping the machine. The chuck is operated by means of a geared capstan lever at the front. A self-centring steady chuck is fitted at the rear end of the spindle. The headstock is fitted with the latest type eight-speed change gearbox, with heat-treated gears running in an oil bath, and the multi-disc friction clutch with the operating lever at the front is provided for stopping and starting independently of the motor, which is 5 h.p. and is mounted over the gearbox.

The bed is fitted with a large oil reservoir and a rotary gear type oil pump is provided for lubrication to the screwing and cutting-off tools.

The machine is also provided with a leading screw, and the necessary change wheels with an operating lever on the saddle complete with gunmetal split nut, the saddle carrying a stationary die head of the tangential type, with an efficient cutting-off slide at the back, and has a capacity of 2½-in. to 6-in. bore tubes, although extra equipment can be provided if necessary for increasing the capacity from 1 in. to 6 ins. An electrically driven oil pump can be fitted in place of Rotary pump.

Patent thread length indicator, fitted to the saddle, giving clear visual indication of the length of thread being cut.

This machine is also made fitted with four-speed A.C. motor instead of gearbox.

The No. 2 size, high speed combined screwing and cutting-off machine shown in fig. 3 is driven by a four-speed A.C. motor and fitted with an air operated, double acting friction clutch, giving the machine two ranges of four speeds. One set of speeds is arranged for screwing and the other range, at increased speed, for cutting off and frazing.

The forged steel spindle, running in Timken taper roller bearings, is fitted with Winn's patent quick-grip self-centring chuck which is operated by means of a capstan lever at the front. The rear end of the spindle is fitted with a steady bush and ring steadies bored to suit tube sizes.

The saddle carries a stationary tangential diehead with a cutting off and frazing arrangement at the back. The safdle is moved by means of machine cut rack and pinion and is arranged to operate the double acting clutch.

An electrically driven oil pump is fitted to supply lubricant for screwing and cutting off.

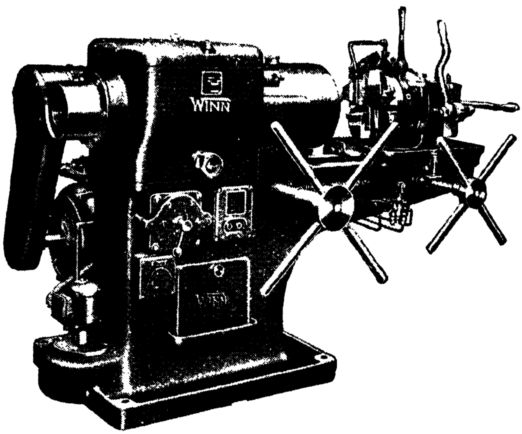


Fig. 3.

SCREWING MACHINES (GENERAL PURPOSE).

Charles Winn & Co., Ltd., Granville Street, Birmingham, 1.)

The illustration (fig. 4) shows a No. 3 size, fig. 6197–BE, machine for screwing up to 3-in. bore tubes and 2-in. bolts.

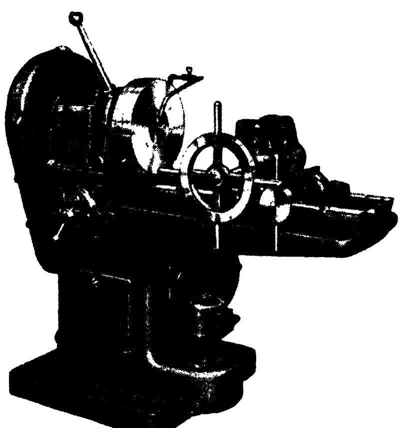


FIG. 4.

This machine is directly driven by an A.C. four-speed motor and has a built-in change-pole controller.

The motor is mounted on slide rails at the back of the machine so that the chain tension can be maintained. A split bearing is provided for the headstock and radial dies are used. The diehead is of the type opened by a bow lever, operating through gun metal slippers and a sliding head.

An interesting feature is the patent thread-length indicator which gives visual indication of the length of the thread being cut, thus enabling the operator to release the dies immediately the required length has been screwed.

The machine is of robust and compact design, and although primarily intended for maintenance and repair work, can be used for production work where a full-production machine is not warranted, but where there is considerable continuous repetition work.

Adjusting strips are fitted for taking-up wear on the saddle and vice. An independent constant-speed motor-driven pump provides an ample flow of cutting lubricant to the work. When required, a cutting-off plate can be fitted to the machine. Necessary chuck, nut holders, and taps for tapping can be supplied with this machine when required.

Other sizes made are No. 1, No. 2 and No. 4, all fitted with electrically driven oil pumps.

This type of machine, when fitted with tangential diehead and automatic opening and closing attachment, becomes a full production machine, and these features can be embodied upon request.

TUBE CUTTING-OFF MACHINES.

(Charles Winn & Co., Ltd., Granville Street, Birmingham, 1.)

The illustration (fig. 5) represents the latest model high speed cutting-off machine for modern tube works requirements, being fitted with Winn's patent Quick Grip self-centring four-jaw chuck

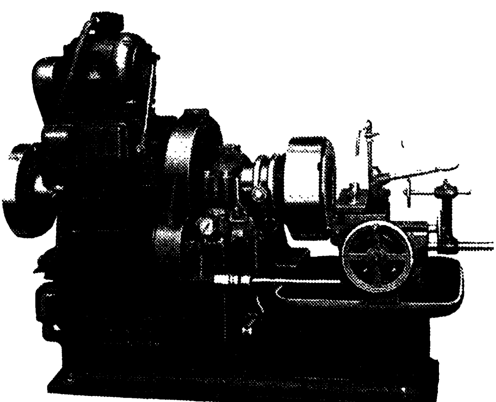


FIG. 5.

operated by a pneumatic cylinder, complete with control valve mounted on front of machine and a three-jaw self-centring steady chuck at the back end of the hollow spindle.

The cutting-off slide has power feed and frazing attachment. The machine is driven by an A.C. four-speed motor. The driving shaft is fitted with a multi-disc clutch operated by a lever in front of the machine.

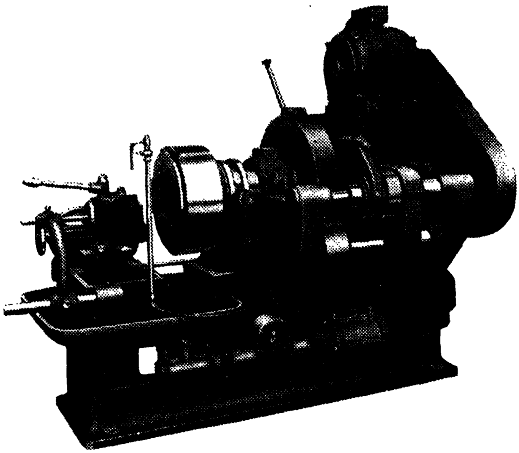


FIG. 6.

The illustrations (figs. 5 and 6) show a No. 6 size machine, suitable for tubes 1-in. to 6-in. bore Other sizes made are No. 2, No. 4, No. 8 and No. 12.

SECTION XXII

PART III

Steam and Power Hammers.

Drop Hammers.

(Brett's Patent Lifter Co., Ltd., Coventry.)

The modern drop hammer provides for extreme rigidity and strength of guiding effect of the tup. Accurate and permanent adjustment is a vital necessity in the use of multiple impression dies, which has superseded the old method of separate dummying. The side shock created by one sided impressions has to be resisted by guide rods capable of withstanding heavy stresses in all directions.

The Automatic Board Drop Hammer possesses the merit of a perfectly free fall. Operating with multiple impression dies, the blows of the hammer are so synchronised that they provide just the time necessary to move the bar from one impression to another. It can be said that the board hammer works at the maximum drop hammer speed. There is no class of drop forging appliance capable of greater production.

Clipping Presses are essential for all drop forging work. They should be of solid construction and capable of withstanding heavy overloads without risk. In some cases it is preferable to use the three-start clutch which gives almost instantaneous application of the slide to the work.

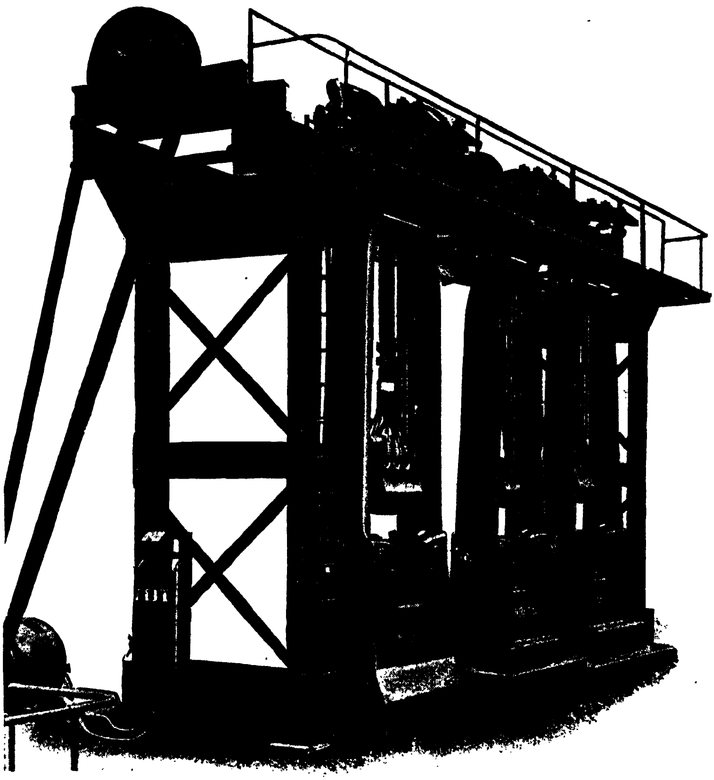


Fig. 1.

FIG. 2.

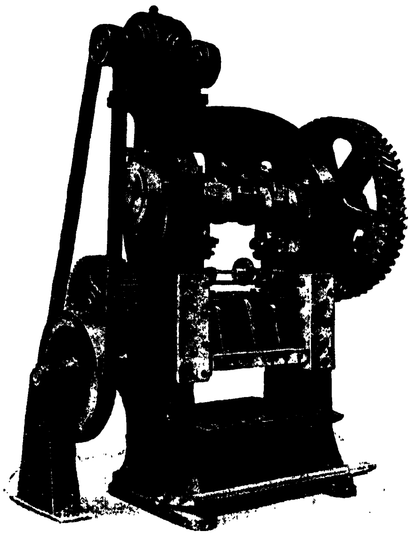


FIG. 4.

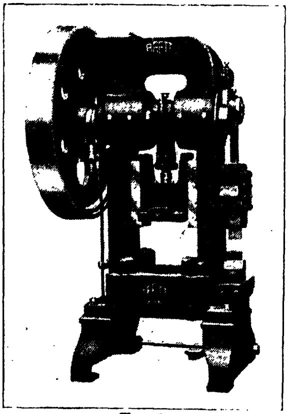


FIG. 3.

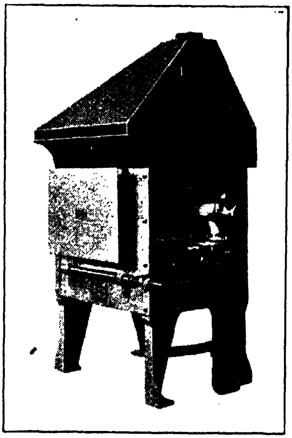


FIG. 5.

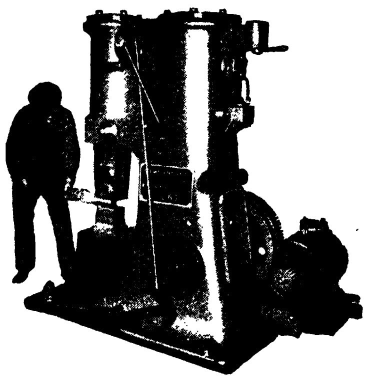
In a general way clipping presses for drop forging are ungeared. Clipping the stamping is a momentary operation and the heavy flywheel stores up sufficient energy to actually do the work, with the advantage that the job is perfectly clean. For very large stampings, such as 4 and 6 throw cranks for motor cars, front axles, etc., gearing is necessary. Fig. 4 shows a typical press constructed for the purpose. It is a very powerful machine and is in consequence capable of dealing with very heavy work.

Oil Fired Furnaces possess many advantages. They give a very intense heat with very little sulphur. Bars are heated quickly and uniformly. There are many types of burners but the one shown is constructed on special lines and provides for combustion chamber, pre-heating of the air supply, positive atomisation and perfect control. The design of furnace shown is suitable for bar heating. Where separate sections are used, naturally these must be placed on the floor of the furnace for which a different design is necessary, but the same burner is utilised.

PNEUMATIC POWER HAMMERS.

(*Alldays & Onions Ltd., Great Western Works, Birmingham.*)

These hammers, which are designed for either belt or motor drive, are made in six sizes, ½ cwt., 1 cwt., 2 cwt., 3 cwt., 5 cwt. and 7 cwt., having strokes of 8", 12", 14", 18", 21" and 24" respectively. As may be seen from the accompanying illustration their design is particularly simple, comprising a hammer and an air pump combined into a single unit.



Pneumatic Power Hammer.

The body of the machine, consisting of the hammer and air pump cylinders, crank case and the foot, through which the anvil body passes, comprises a single rigid casting. The upper part of the hammer or tup forms a piston which reciprocates in the hammer cylinder. An adjustable guide bar prevents the tup from turning and the height of forging is automatically set to any distance within the range of the machine. The air pump consists of a piston with a large bearing surface reciprocating within a cylinder which communicates with the upper part of the hammer cylinder through a port at the top. Interposed in the air port between the two cylinders is a rotary valve which controls the air supply to the hammer cylinder and all movements of the tup. The setting of this control valve is done by means of either a hand lever or foot treadle gear and it may be arranged for the hammer to be held up or down or for the striking of continuous heavy, medium or light blows. When the valve is set for the holding up position, air is drawn from the hammer cylinder on the downward stroke of the pump piston and the tup rises to the top of its stroke by virtue of the vacuum formed above it and the atmospheric pressure on its under surface. On the following compression stroke the piston expels the air in its cylinder into the atmosphere and, with this setting, so long as the piston reciprocates, it continues to exhaust air from the hammer cylinder. When set for holding down, in order that a workpiece may be gripped between the tup and the anvil, the valve allows the air from the pump to be continually fed into the hammer cylinder where it is compressed and builds up a load on the upper surface of the hammer. The valve can also be set for the tup to give full, or heavy, blows or light blows. When giving heavy blows the valve setting permits a free passage for air between the pump and hammer cylinders and as the pump piston reciprocates the tup also moves up and down striking its heaviest blows. In order to strike light blows the air from the pump cylinder is partly exhausted and striking of the tup is proportionally light. Single blows are obtained without any special fitments or gear. The mechanism of the valve is such that when the control levers are released, the tup returns to its highest and inoperative position and remains there so long as the pump crankshaft revolves.

SECTION XXIII

PART I

Metallurgy.

Non-Ferrous Alloys.

DURALUMIN

(Registered Trade-mark.)

(James Booth & Co., Ltd., Nechells, Birmingham.)

'Duralumin' is a registered trade-mark applicable to a group of important engineering alloys whose mechanical properties are capable of wide variation by suitable heat treatment. The alloys contain 90 to 98 per cent. of aluminium, the remainder being made up of copper, manganese, magnesium, silicon, iron, zinc, etc., according to the type of alloy.

'Duralumin' was introduced by James Booth & Company in about 1912, and was thereafter developed and improved steadily. Much of its development was carried out in conjunction with the aircraft industry, which was quick to take advantage of the low density and the high strength/weight ratio of the new material. This phase is reflected in the existence of numerous Air Ministry DTD and British Standards Aircraft Specifications for this class of alloy such as the well-known BSS. 6L1 and 5L3, and DTD. 390. In recent years, however, the advantages of using aluminium alloys have been appreciated in many other spheres of engineering, and nowadays the greatly increased availability and reduced price of these alloys, have enabled them to take their place as second only to ferrous materials in general engineering importance. A comprehensive series of B.S.I. Specifications are available for this class of product and a new series will shortly be published.

There are now four main types of 'Duralumin,' all of which respond to heat treatment:—

'Duralumin "B"' is the modern form of the original 'Duralumin,' and it has been in wide use for many years. It is essentially an alloy of aluminium with copper, manganese and magnesium. The heat treatment given to this allow consists of a solution treatment at about 495°C. to take the copper and magnesium into solid solution followed by a rapid quench which retains this condition for a limited period at room temperature. For an hour or so after quenching the alloy is quite soft, but precipitation of submicroscopic particles causes a hardening, rapid at first but becoming slower with time; from a strength of 17 tons per sq. in. as quenched, the alloy reaches 25 to 28 tons per sq. in. in five days and retains this strength indefinitely.

The soft condition immediately after quenching provides a very convenient opportunity for carrying out a moderate amount of forming, the subsequent aging then taking place without distortion. For more drastic deformation the material is annealed at 380° C., which treatment agglomerates in precipitate, thereby removing its hardening effect and produces the softest condition of the alloy. As with most other metals, difficult forming may be done in stages with inter-stage annealing, with the usual proviso that the amount of work at each stage should be above about 5 to 10 per cent., or coarse grain may be produced at the next anneal. For full mechanical properties, of course, the material must finally be solution treated and quenched to allow of age hardening.

'Duralumin "S"' is a more recent development of 'Duralumin "B"' and it is characterised by a higher silicon content. As solution treated and aged at room temperature its properties are very similar to those of 'Duralumin "B,"' but if it is given further 'precipitation' heat treatment at about 180° C., it undergoes a second precipitation which affects particularly the 0·1 per cent. proof stress of the material. The more refined methods of design, based largely on

proof stress, thus allow a higher working stress in 'Duralumin "S,"' by virtue of its relatively high proof stress. This alloy was widely used during the last war in such highly important aircraft members as wing spars, and its use as high strength sheet is steadily increasing. Solution treatment is carried out at 505° C. and requires somewhat more critical control than 'Duralumin 'B."'

'Duralumin "Q"' is a higher strength grade of 'Duralumin "B,"' and is mainly used in the form of clad sheet and tube. This alloy has a solution treatment temperature of 495° C.

The second type of 'Duralumin' is typified by :—

'Duralumin "H"' which is essentially an alloy of aluminium, manganese, magnesium and silicon. It is fully heat treatable, but in the annealed state it is very readily workable and may be spun and pressed into the deep pressings with ease. Solution treatment produces a moderate strength alloy with good ductility and excellent corrosion resistance. It is not susceptible to natural ageing. If the solution treated alloy is precipitated at about 200° C., its strength is very greatly improved, and in this condition it gives a very high ratio of proof stress to ultimate stress. With this alloy considerable forming may be done after quenching without any time limit being imposed by age hardening, as in the case of 'Duralumin "B"' or 'Duralumin "S,"' and of course, it may be subsequently fully hardened by precipitation heat treatment. A variant of this alloy is 'Duralumin "E"' which consists of a 'Duralumin "H"' type alloy to which copper has been added for increased strength. The behaviour of this alloy in its heat treatment lies between that of 'Duralumin "H"' and 'Duralumin "B."'

The third type of 'Duralumin' is the newly developed :—

'Duralumin "K"' which contains zinc and is the strongest aluminium alloy yet produced. It is intended for use in applications requiring particularly high strength with the high proof stress/ultimate stress ratio given by precipitation at about 135° C. Solution treatment of this alloy is conducted at 460° C., followed by quenching in water. The aircraft industry is steadily exploring the possibilities of this alloy and considerable experience is being gained in its manufacture and fabrication. A somewhat softer alloy of this series is 'Duralumin "L,"' which contains a lower proportion of hardening elements, and is used for the manufacture of sheet and tube.

The fouth group of 'Duralumin' is :—

'Duralumin "T"' and 'Duralumin "J"' which are intended for specialised service at elevated temperatures and possess good creep strength at temperatures up to 250° C.

Other Aluminium Alloys.

In addition to the 'Duralumin' range of alloys there is an interesting series of alloys of aluminium and magnesium which are not heat treatable but which have interesting properties in the annealed condition, and whose strength can be increased by cold working. These alloys which are sold under the registered trade names of 'MG2,' 'MG5' and 'MG7' contain 2, 5 and 7 per cent. of magnesium respectively. These alloys give an interesting combination of workablity and strength, they are weldable and have a high resistance to corrosion. In addition, they possess the peculiar property of steel in that they have a fatiguel imit. In general the heat treatable alloys do not have such a limit.

Aluminium alloys possess good resistance to corrosion, in general far better than structural steels. In most applications to structures when frequent inspection is not practicable, paint coatings are naturally applied. For the best possible protection an undercoat containing zinc chromate is applied to a surface specially prepared for adhesion by anodising or by a simple chemical dip. Almost any kind of paint can then be put on top of the undercoat.

Sheet material, particularly for aircraft use, is often used in very thin gauges, and corrosion cannot be tolerated, nor can heavy paint coating. In these cases, sheets are used in which a strong alloy core has an integrally bonded pure aluminium coating rolled on during manufacture. This gives the composite sheet a very high reistance to attack, and as is well known, the stressed skin of modern aeroplanes is of such material quite unpainted. Clad sheets of this description are sold under the registered trade name of 'Aldural.'

Actual structures have demonstrated that aluminium alloys may be attached to steel without detriment, provided an ordinary paint cover is applied. Contact with most other non-ferrous metals, however, sets up an electrolytic action and has to be avoided particularly in the case of copper and brass.

Practically all the alloys described above may be welded, but in general the heat treatable alloys containing copper are difficult to weld and when welded the properties of the welded joints are not attractive. ' Duralumin " H " ' welds very well, but the highest properties in the welded joints are given in the case of the aluminium magnesium alloys. Some specialised experience is desirable in order to produce the best type of welds in these alloys, but the strength of such joints is upwards of 90 per cent. of the strength of the parent annealed metal. The alloys may also be jointed by soldering and if the correct types of solders are used resistance to corrosion may be obtained, but users are not advised to embark on operations involving soldering of aluminium alloys without expert advice.

MACHINING.

The machining of the various ' Duralumin ' alloys does not present any great difficulty, they may be machined at considerably higher speeds than many other so-called free machining alloys, and normally they take a fine finish. In the softer conditions, however, and in the case of the 'MG2' and 'MG5' alloys, machining will not be so easy. It will generally be found that tools for cutting aluminium alloys should have more top and side rake than those which are used for free cutting brass or steel, and it is advisable to keep very keen edges on the tool with fine emery wheels, and care should be taken to keep the edges of the tools free from metal which may adhere to them by fairly frequent stoning.

Roughing cuts may be made dry, but with finishing cuts cutting compound is necessary. Suitable compounds are soluble cutting oil and mixtures of paraffin and lard oil in about equal proportions, will also be found to be suitable. If parts during rough machining are found to heat up, they should be cooled before the final finishing to size, as the co-efficient of expansion of ' Duralumin ' is higher than that of brass or steel, and some slight inaccuracies in dimensions may be noticed after cooling if this precaution is not taken.

USES OF ALUMINIUM ALLOYS.

A few of the uses which these alloys have found may be indicated. It will be appreciated that in all kinds of mobile units these alloys have great value because of their high strength/weight ratio. The value of light alloys in modern military aircraft has been much appreciated in recent days and, of course, it is in aircraft that much light alloy has been used in the past. In these post-war days, however, the strong light alloys are finding considerably extended applications in general engineering, and the British Standards Institution have embarked on the production of a series of general engineering specifications for all these alloys. It is hoped that these specifications will be published in the near future. Light alloys are used extensively in modern omnibus body manufacture, roofs, floors, window-frames, seats, baggage racks, heating and electrical equipment, treat plates, rails, handles, panelling and stressed parts of the chassis are made from these alloys. A further interesting use of the alloys has been the manufacture of mine skips. While at the present time considerable attention is being directed towards the use of certain selected alloys for marine purposes, the Admiralty have recently issued a booklet making recommendations in this connection. The tables below record the physical properties of the ' Duralumin ' alloys and a list is given of the specifications for all the alloys manufactured by the company. These specifications were issued by the Superintendent for the Technical Applications of Metals, under the heading S.T.A.M., during the war for use by the services. Relevant aircraft and B.S.I. specifications are also indicated in the table.

Specific Gravity	**2·6 to 2·9**
Co-efficient linear expansion	**24·0 × 10^{-6} per °C.**
Specific Heat	**0·214 (water = 1·0)**
Thermal Conductivity	**0·3 to 0·45 CGS units at 0°C.**
Electrical Resistivity	**3·3 to 5·3 Microhms per cc.**
Young's Modulus of Elasticity	**10 × 10^6 lb. per sq. in.**
Modulus of Torsion	**3·8 × 10^6 lb. " " "**
Poisson's ratio	**0·32**
Fatigue range	**10^7 cycles. ' Duralumin "B" ' ± 9·5 tons per sq. in.**

TENSILE PROPERTIES.

S.T.A.M. Symbol	Alloy Designation (Trade-marks)	Condition	0·1% Proof Stress	Ultimate Tensile Stress	Elongation % on 2"	Related Specifications
AW4 A	'MG2'	Extrusions	—	11	18	—
B		Tubes	—	9–15	—	DTD.310B 440
C		Sheet	—	11–15	—	DTD.606 634
AW5 A	'MG3'	Extrusions	6	14	18	—
B		Tubes	7–12	14–16	18–5	—
C		Sheet	7–15	14–18	18–5	DTD.180B
AW6 A	'MG5	Extrusions	8	16	18	DTD.303
B		Tubes	8–14	17–18	18–5	—
C		Sheets	8–17	17–20	18·5	—
AW7 A	'MG7	Extrusions	9	20	18	DTD.297
B		Tubes	9–16	20–25	18–5	DTD.190
C		Sheets	9	20–23	18	DTD.182A
AW10 A	'Duralumin "H"'	Extrusions	7	12	18	—
B		„	15	18	16	—
C		Tubes	17	20	10	—
D		Sheet	7	14	15	—
E		„	15	20	8	—
F		Wire	—	13	—	—
AW11 A	'Duralumin "E"'	Extrusions	10	17	15	DTD.443
B		„	20	25	8	DTD.423A
AW12	'Duralumin "T"'	Bars and Forgings	21	27	10	DTD.130A DTD.410
AW13	'Duralumin "M"'	Wire and Rivets	—	17	—	DTD.327
AW14	'Duralumin "B"'	Wire and Rivets	—	25	—	2L37
AW15 A	'Duralumin "B"'	Extrusions	15	25	15	6L1, 2L39
B	or	„	28	32	8	DTD.364A
C	'Duralumin "S"'	Tubes	18	26	12·5–8	5T4
D		„	23	29	12–8	DTD.464
E		Sheet	15	25	15	DTD.603, 5L3
F		„	23	28	8	DTD.646
G		Aldural	15	25	15	DTD.610, 390
H		„	21	27	8	DTD.546
AW16 A	'Duralumin "K"'	Extrusions	33	38	5	DTD.363
B	'Duralumin "L"'	Sheet	27	32	8	DTD.687
AW17 A	Y Alloy	Bar and Forgings	14	24	15	4L25
B		Forgings	—	22	8	—
AW18	'Duralumin "J"'	Forgings	—	25	6	2L42

NOTE:—The registered trade-name 'Duralumin' is often abbreviated to 'Dural' which is also a registered Trade-mark of James Booth & Co., Ltd.

BULL'S METALS.

(Bull's Metal & Melloid Co. Ltd., Yoker, Glasgow, W. 4.)

Bull's Metals are strong, malleable alloys of good casting qualities, and equal to good gun-metal in resistance to corrosion. They are manufactured in two qualities, No. II and No. III.

Alloy No. II is suitable for founding purposes and extensively used for propellers and propeller blades, parts of guns and gun carriages, requiring the strength of cast steel with great toughness and resistance to corrosion.

Alloy No. III is very malleable and can be rolled, forged, stamped, or otherwise wrought at a red heat; this metal also casts well. Its composition ensures great resistance to corrosion and wear. It is absolutely non-magnetic.

Rolled in round, hexagon, angle, and section bars, or into sheets and plates; forged or stamped Bull's metal is used for pump rods, centrifugal pump spindles, rams, stern shafts, sluice valve spindles, parts of marine and hydraulic machinery generally. Forged propeller blades, propeller studs and nuts. Bolts, nuts, screws, studs, and nails. Parts of torpedoes and torpedo boats, ordnance, ammunition, etc. Condenser plates, strainer plates, doctor blades. Hull plates and angles for launches, torpedo boats, yachts, sailing and rowing boats.

The elastic and ultimate resistance to tensile, crushing, and torsional stresses are greater in the case of wrought Bull's metal than in mild steel, and higher working strains may therefore be used with Bull's metal. Where resistance to the action of water or chemicals enters as an important factor, Bull's metal, whether wrought or cast, becomes specially superior to steel.

TENSILE TESTS OF BULL'S METALS.

Dimensions.	Area. Square inch.	Reduction area at fracture.	Extension.	Ultimate Stress.	Remarks
Diam. Inch.		Per cent.	Per cent.	Tons per sq. in.	
0·749	0·4406	22·1	23·6 on 8 ins.	31·41	Forged.
0·623	0·3048	37·9	8·1 „	39·93	Rolled, hard finish.
0·690	0·3739	37·8	30·0 „	35·01	Rolled, medium finish.
0·688	0·3717	30·6	29·0 on 6 ins.	31·47	Cast medium.
0·750	0·4418	24·2	22·0 on 8 ins.	38·68	Cast hard.

'MELLOID.'

(Registered Trade Mark.)

This alloy is a malleable genuine bronze containing no inferior metal; its resistance to corrosion is very great. Its tensile strength in a rolled state when suitably finished is as high as that of mild steel; the strength is not appreciably affected at the temperature of high-pressure steam, whilst copper at such heat is very much reduced in strength and elasticity.

The alloy is perfectly malleable at *all* temperatures, from cold to blood-red heat; it has no brittle stage at a black heat like most metals and alloys, and at a red heat it is much superior in strength and toughness to mild steel under such conditions, and the alloy, under all conditions, is exceedingly tough.

'Melloid' meets the mechanical requirements referring to copper, naval brass, and rolled bronze specified by the Admiralty, the principal railway companies, and consulting engineers.

It is claimed that 'Melloid' will be found superior to all brass alloys, including naval brass yellow or Muntz metal, and also to wrought copper, wherever a combination of toughness, great strength, and a maximum resistance to corrosion is required.

'Melloid' is very suitable for condenser tubes and piston cooling tubes, boiler tubes, fire-box plates, boiling vats, beater bars, doctor blades, etc.

OIL COOLERS FOR HEAT TREATMENT OF STEEL.

(Heenan and Froude Limited, Worcester, England.)

The continuous introduction of hot steel into oil quenching tanks naturally heats up the oil, which retains heat for a long time, and rapidly becomes too hot to permit of the process being carried on.

Owing to the poor heat conductivity of oil, such arrangements as water-jacketing the tank or inserting pipe coils through which water is circulated are ineffective, the cool surfaces affecting only that portion of the oil with which they are in immediate contact, while allowing the main body to remain overheated.

The 'Heenan' system of oil cooling is to extract the hot oil by means of a suitable centrifugal pump and pass it through a 'Heenan' oil cooler before returning it to the quenching tank. This oil cooler is built on lines similar to those of the 'Heenan' water cooler described on page 1273. In effect, oil is distributed over a large cooling surface with which cool air is forced into contact. Circulating water is not used.

The use of this cooler is very widespread at home and abroad, and by its means the temperature of the quenching oil can be kept very much more uniform, thus promoting greater consistency in hardness of the heat-treated products.

SOLDERS AND SOLDERING FLUXES.

(Fry's Metal Foundries, Ltd., Tandem Works, Merton Abbey, S.W. 19.)

Soldering is a joining process for metals differentiated from welding by the fact that it is carried out at temperatures lower than the melting point of the joint members.

1. In soft soldering, lead-tin alloys melting at temperatures little above 183° C. are used. The value of the method lies largely in its simplicity and convenience. The strength of the joints obtained is not high, but soft soldering is often used for sealing joints made mechanically, *e.g.*, by lock seaming or riveting.

2. Hard solders, which are discussed later, give strong joints, but temperatures of 650°–1000° C. are required.

TIN-LEAD SOLDERS.

In soldering, adhesion of the solder to the joint members results from the wetting of the surfaces by the molten solder, which then alloys with, or 'tins,' the basis metal. For wetting to occur, clean molten solder must come into contact with clean basis metal; to ensure this contact a flux is used.

The main reasons for the use of the tin-lead alloys are their low melting point and the ease with which they 'tin' other metals. The melting point of the alloys varies with the composition. Pure tin melts at 232° C. (450° F.), pure lead at 327° C. (620° F.). In all alloys containing both metals, there is present a constituent known as the eutectic having the fixed composition of 63 per cent. tin, 37 per cent. lead. It melts at a temperature of 183° C. (361° F.), which is thus considerably lower than the melting point of either of the pure metals. Most solders contain a greater percentage of lead than the eutectic. Under the microscope, the excess lead in these alloys appears in the form of separate crystals distributed through the eutectic. The eutectic portion of the alloy melts at 183° C.; the lead crystals, however, melt only as the temperature increases beyond this, to some higher value. The alloy therefore melts not at one fixed temperature, but over a range of temperatures. The melting range increases with the proportion of excess lead present in the alloy.

Tinman's Solder, which is largely used for hand soldering, has a composition around 50 per cent. tin, 50 per cent. lead. This is sufficiently close to the eutectic to ensure that the proportion of lead crystals is small. As a result, the melting range is short and the alloy is completely molten at a temperature of approximately 210° C.

TABLE I.

Composition*		B.S.S. Grade.	Completely Solid at	Completely Liquid at	Freezing Range.	Weight lb./cu. in.
Tin. %	Antimony. %					
65	1 †	A	184° C. 363° F.	188° C. 370° F.	4° C. 7° F.	0·302
63	—	—	183° C. 362° F.	183° C. 362° F.	— —	0·302
60	0·5 †	K	183° C. 362° F.	188° C. 370° F.	5° C. 8° F.	0·309
50	2·8	B	185° C. 365° F.	204° C. 399° F.	19° C. 34° F.	0·317
50	0·5 †	F	183° C. 362° F.	212° C. 414° F.	29° C. 52° F.	0·320
45	2·5	M	185° C. 365° F.	215° C. 419° F.	30° C. 51° F.	0·325
42	0·4 †	G	183° C. 362° F.	230° C. 446° F.	47° C. 84° F.	0·332
40	2·2	C	185° C. 365° F.	227° C. 441° F.	42° C. 76° F.	0·333
30	1·4	D	185° C. 365° F.	248° C. 478° F.	63° C. 113° F.	0·351
18	0·9	N	185° C. 365° F.	275° C. 527° F.	90° C. 162° F.	0·371

* Lead balance. † Maximum.

The short melting range carries with it the very desirable advantages of ease and rapidity of melting and, conversely, of rapid solidification. Further, the alloy has excellent fluidity. These properties are of great value in bit and machine soldering. Fluidity is especially important in applications where the solder has to penetrate and flow into confined spaces, while ease of flow and speed of setting assist the production of smooth seams.

It is usual to add to solders a small proportion of antimony. This has two advantages. Firstly, the antimony increases the amount of eutectic in the alloy, being twice as effective as tin in this respect. Secondly, the solder is considerably strengthened, the improvement obtainable being as much as 20–25 per cent. The amount of antimony added must be carefully controlled during manufacture, however, or the solder will be sluggish and brittle.

For some purposes, *e.g.*, soldering of zinc, it is desirable to use a solder with a low antimony content.

The composition and properties of some B.S.S. solders are given in Table I.

Plumber's Solder.—As the lead content of the solder increases, the melting range increases. Thus an alloy containing 33⅓ per cent. of tin and 66⅔ per cent. of lead commences to melt at 183° C. and is fully liquid only when the temperature reaches 250° C. In the interval the solder is a mixture of solid and liquid metal; in this pasty condition it can be wiped around the joint.

All plumber's solders exhibit this characteristic feature. The composition varies slightly according to the type of work being carried out and the preference of the plumber. Decreasing the tin content increases the plastic range but gives a coarser 'wipe.' The following grades are widely used :—

Grade.	Melting Range.	Comments.
'Belfry'	185° C.–254° C. 365° F.–490° F.	Used for general plumbing work.
'Abbey'	185° C.–248° C. 365° F.–478° F.	Higher tin content, for best quality plumbing.
'Prior'	185° C.–245° C. 365° F.–473° F.	Extra rich grade for cable jointing.

Soldering Fluxes.

The soldering flux has two functions :—

(1) To clean the surfaces to be joined; (2) to protect them from the atmosphere during the soldering operation.

Fluxes can be divided into two main classes depending on the extent to which these two functions are fulfilled.

Active Fluxes are capable of effecting a considerable amount of cleaning. Killed spirits has been used in the past for general soldering work, but it is very corrosive and tends to leave a sticky residue in the joint itself.

Frysol active fluxes do not suffer from these disadvantages to the same extent. They are rapid in action and there are few metals which cannot be soldered by their use. They are generally employed for production work on iron, steel, tinplate, copper, brass and bronze, zinc, galvanised iron, nickel, etc. This flux is supplied in three forms :—

Frysol Soldering Paste Flux.
" " Fluid
" Tinning Salt.

All active fluxes exercise some corrosive action if they are left in contact with the members of the joint after soldering is completed. For many purposes this is not serious, as witness the fact that most of the proprietary fluxes marketed are of this type. It is desirable after soldering with these fluxes to wipe or wash the joint in order to remove any residue.

Safety fluxes are for use where cleaning of the joint after soldering is not possible and freedom from corrosion is essential. In general, these fluxes are slower and less effective in removing surface oxide; they act largely by protecting the metals from the atmosphere during soldering. As a result, it is necessary to start with clean surfaces. Their value lies in the fact that, having little chemical activity, they cannot cause corrosion of the joint.

Resin and tallow are in this class. They are suitable for soldering the less 'difficult' metals such as tin, pewter, lead and clean copper or brass. *Fry's Alcho-re* is a resin-base safety flux in the form of a liquid or paste, and is thus applied much more easily. It is especially suitable for electrical work, being non-conducting and completely non-corrosive. Further it sets rapidly after soldering, leaving a hard dry residue. This is important since most fluxes leave a sticky residue which absorbs moisture or traps dust or metal filings, thus lowering the insulation resistance of the joint.

Oleic Acid No. 9 and *Oleic Acid No. 10* are liquid safety fluxes intended for use mainly on tinplate. They are also suitable for soldering iron and steel if the surfaces are carefully cleaned before soldering.

Fryolux tinning compounds are mixtures of powdered solder and flux. They are thus convenient to use and, moreover, being very active, they give satisfactory results on metals that are normally only tinned with difficulty. The compound is simply sprinkled on the heated metal; if the latter is very dirty, rubbing with a scratchbrush assists the cleansing action of the flux. When a bright tinned surface has been obtained, surplus flux is wiped off.

Solder Paint and Solder Creams.

Solder paint and solder creams contain solder in powdered form intimately mixed with flux and provide a simple method of tinning or sweat soldering which for many purposes has superseded the use of solder in stick form or hot dip tinning.

They are best represented by *Fryolux* solder paint which carries finely powdered solder or pure tin in suspension in an active flux. The constituents do not separate and therefore stirring during use is unnecessary. The paint can be thinned down for some duties by the addition of small quantities of water and can be applied by a brush, pad, spraying or dipping. The residue left after soldering is non-greasy and can be removed by washing or wiping the joint.

Soldering by this method lends itself to mass production—the quality of the joints is less dependent on the skill of the individual operators. Many soldering operations can be mechanised, *e.g.* by passing the articles after applying the solder paint over a series of gas jets or through an oven or by the use of rotating jigs.

Soldering of Cast Iron.

Cast iron is difficult to tin owing to the presence of graphite and non-metallic inclusions. Pickling in hydrofluoric acid before tinning is sometimes recommended, but this method is dangerous and not always successful. For soldered joints, cleaning with an emery wheel may be sufficient, followed by soldering with an active flux.

For most purposes, including the tinning of bearing shells, it is advisable to use *Fryolux* tinning compound on account of its increased activity.

Soldering of Aluminium.

Soldered joints in aluminium can be made quite successfully, but their use is limited. The reason for this is that aluminium corrodes readily in moist conditions when it is in contact with other metals. Welding should therefore be used whenever possible to joint aluminium, using a filler rod of the same composition. Sometimes welding is out of the question owing to the high temperature required. Soldering is advantageous from this viewpoint, since there is much less danger of distortion or of softening the material. Soldering should not be employed for articles which are to contain water, and if the joint is likely to be exposed to moisture attack it should be protected by a coat of paint or enamel. If these precautions are observed, durable joints can be obtained.

In making the joints, a different technique is required for aluminium. The metal forms an oxide skin on the surface which is not removed by the ordinary soldering fluxes. The method employed is to remove it mechanically during the soldering operation. The solder is melted on the metal and used to exclude the air while the surface beneath is scraped with a sharp tool, such as a hacksaw blade or a scratchbrush. The aluminium can be tinned rapidly in this way if a solder of suitable composition is employed. The most satisfactory alloys contain tin, with zinc and other elements.

Fryal is an improved aluminium solder of this type. When molten, the solder is capable of penetrating the skin of oxide and therefore little scraping is required. Often it is sufficient merely to rub the alloy on the heated aluminium. Once a surface has been obtained in this way soldering can be completed with an ordinary lead tin solder. By this method, the aluminium need not be heated above a temperature of 250° C.

It will be observed that a flux is unnecessary.

Solders for Hot Dipping.

The hot dipping process is used either to provide a protective coating or as a means of soldering complicated assemblies, such as commutators or automobile radiators. The usual procedure is as follows:—

(1) *Degreasing*. If the article is greasy, treatment in an alkaline solution or in a trichlorethylene vat is necessary.

(2) *Pickling*. The surface must be free from oxide or scale. Pickle solutions containing sulphuric, hydrochloric or nitric acid are used, according to the metal being treated.

(3) *Fluxing*. After washing, the article is dipped in flux. Any of the liquid fluxes described above are suitable.

(4) *Dipping in molten solder*. The time of immersion should be no longer than that required to bring the article to the temperature of the bath. The latter depends on the alloy and the nature of the work being dipped, but the temperature should always be the lowest that gives a smooth coating.

(5) *After withdrawing the article*, the coating is allowed to set and is finally washed, dried and polished.

Soft Solders for Elevated Temperatures.

All tin-lead solders commence to melt at 183° C. (361° F.) but as they approach this temperature their strength falls rapidly. Thus the strength of a joint in brass made with Tinman's solder is reduced by 75 per cent. when the temperature reaches 150° C. (302° F.).

For soldered parts subjected to temperatures over that of the boiling point of water, it is advisable to use a special solder having superior resistance to elevated temperatures. The special solders listed in the table are much stronger than Tinman's solder, at temperatures over 100° C.

Solder.	Solder is completely Solid up to—
50/50 Tinman's solder	183° C. (362° F.)
H.T.3	236° C. (457° F.)
L.S.2	304° C. (579° F.)
L.S.T.1	308° C. (586° F.)
L.S.4	294° C. (561° F.)

H.T.3 is easy to apply, being free flowing, and can be used to replace tin-lead solders without any change being made in the soldering technique. It is suitable for electrical work since it has a high electrical conductivity and can be used with a safety flux.

Lead-silver solders have an even greater advantage in temperature resistance since they do not begin to melt until the temperature reaches about 300° C. (572° F.).

They need rather more experience and skill than normal solders as they do not possess the free running properties of the tin containing alloys. They should generally be used in conjunction with an active flux.

Amongst the applications in which the above solders are employed are aircraft cooling systems, hot water appliances and electrical machinery.

Fusible Solder.

Alloys with very low melting points are frequently used for special purposes. Among these may be mentioned solders for work which might be damaged by the temperature of application of tin-lead alloys; in safety devices, *e.g.* for operating alarms or breaking the electrical circuit when the temperature exceeds the melting point of the alloy; similarly, in fusible plugs for boilers; as fillers for the bending of thin walled tubes; and setting media for the mounting of punches and dies.

Table IV.

Fry's Alloy. No.	Alloy.	Melting Point. °C.	Melting Point. °F.
62	Tin-lead	183	362
20	Cadmium-bismuth	144	291
18	Tin-lead-cadmium	142	288
17	Tin-bismuth	138	281
15	Lead-bismuth	124	256
11	Tin-bismuth-cadmium	102	216
9	Tin-lead-bismuth	95	203
7	Lead-bismuth-cadmium	91·5	197
2	Tin-lead-bismuth-cadmium	70	158

Fry's Tube Bending Alloy, melting at 70° C., is used as a filler, providing the internal support necessary to prevent distortion in the bending of thin walled tubes. The alloy is melted in hot water and poured into the tube, which is plugged at one end. The filling is then chilled quickly by plunging the tube into cold water.

After bending, the tube is emptied by melting out the alloy in hot water.

Matrix Alloy is used for setting dies and punches in press tools. The punches are located in oversize holes in the backing plate and Matrix Alloy is poured into the clearance space. The alloy expands on solidification and so holds the part firmly in position. The alloy can be poured at about 230° C. so that the temper of the die is not affected.

HARD SOLDERS.

Hard solders were originally brasses containing a high proportion of zinc. More recently it has been found that the addition of silver to brass lowers the melting point of the solder and gives a better joint. Silver solders are now employed extensively despite their high cost.

BRAZING SOLDERS.

These usually contain about 50 per cent. of copper and 50 per cent. of zinc, and the melting point is in the region of 870° C., *i.e.* at a red heat. With this alloy it is possible to braze the commonly used brasses which have melting points of 900° C. or over, and, of course, higher melting point metals such as iron and steel.

The British Standard Specifications (No. 263—1931) cover three grades of brazing solder:

	Copper.	Zinc.
Grade AA	59–61	Balance
,, A	53–55	Balance
,, B	49–51	Balance

Grade AA is intended mainly for solder supplied in the form of wire or slittings.

With these solders, a borax type flux is generally used.

SILVER SOLDERS.

GRADE.	MELTING RANGE. Solidus.	Liquidus.	REMARKS.	SIZES.
B.S.S. Grade 'A' .	690° C. (1274° F.)	735° C. (1355° F.)	High grade silver solder especially suitable for electrical work.	$\frac{3}{16}$" × ·040" (normal stock size).
B.S.S. Grade 'B' .	700° C. (1292° F.)	775° C. (1427° F.)	Specified for general, aircraft and electrical engineering.	$\frac{1}{16}$" × ·040" 0·20 Sheet ·0625" Round.
F.E.F. . . .	595° C. (1103° F.)	630° C. (1166° F.)	Low melting point silver solder very fluid and easy to use.	Wire. No. 3 Silver Solder Flux
Standard . . .	740° C. (1364° F.)	780° C. (1436° F.)	An inexpensive, high strength alloy.	is suitable for these solders.

Of these alloys, F.E.F. quality is normally used owing to its low melting temperature. It should be used with No. 3 Silver Solder Flux, a powder which should be mixed with water into a smooth paste.

PRESSURE DIE CASTING.

(*Sparklets, Ltd., London, N.* 18.)

During recent years the business of pressure die casting has made very rapid strides both in the possibilities of economical production of intricate parts and in the development of alloys suitable for the purpose. The industry can, therefore, quite justly claim to be established on a firm basis for very great economies can be effected in the use of die castings as, following the manufacture of the tools or dies, the die-cast components are rapidly produced at a minimum of cost.

Economy is further assured by reason of the interchangeability of all die-cast parts produced from a given set of dies, and this is a constant factor since all dies are manufactured from a high grade steel suited to the requisite temperature for any particular alloy and, with accurate die-making in the first instance, mass production can be effected without alteration of form; whilst the elimination of machining operations owing to the fact that holes, slots, etc., are cast to finished dimensions in the actual die casting process is a fact far from negligible where costs are the major consideration.

Care must be exercised in the design of parts proposed for die casting, and avoidance of under-cuts on any internal form should receive consideration because, even if the production of the necessary dies is possible, such forms will generally operate against the most economic production.

Further, designs should incorporate as far as possible an equal thickness of section as this will facilitate the production of homogeneous castings. Where bosses, etc., are inevitable substantial fillets should be provided and, at the same time, advantage should be taken of the possibility of comparatively thin walls as a distinct saving of material is thereby effected. Sound advice from the specialist during the initial designing can have far reaching effect on the cost and utility of the ultimate product.

The development of die casting alloys has now made it possible to offer a product which can in many cases replace cast iron or brass, and with high grade zinc as the basic metal a tensile strength of 35,000 to 40,000 lb. per sq. in. is obtainable coupled with reasonable ductility. In addition to the above, the 'Sparklets' range of alloys includes many with aluminium, tin and lead as the basic metal—all of which can be successfully used to meet various requirements respecting weight or other service conditions.

Sparklets Woven Wire Braiding.

(*Sparklets, Ltd., London, N.* 18.)

Fifty years ago Sparklets, Ltd., introduced plaited wire in tubular form for the reinforcement and protection of rubber hose. At that time its application was limited to the armouring of rubber or rubber and asbestos tubing used principally for flexible gas and air connections. The technical developments which have taken place in all branches of engineering have resulted in a much wider application of this form of reinforcement and protection.

It has been proved by actual tests that where flexible hose has to withstand high gas or liquid pressure one coating of Sparklets Woven Wire Braiding, particularly if in the form of Multiplait, will increase the bursting strength of the tubing approximately ten times. By utilising a number of layers even greater strength can be obtained.

The great additional strength imparted to rubber or other hose by Sparklets Woven Wire Braiding is due to the method employed; the wires are closely woven and cover the surface of the tube entirely without affecting its flexibility. The wire coating will also prevent kinking of metallic tubing or flattening of rubber hose.

The high elastic limit of the wire used enables the hose to withstand sudden increases of pressure to a much greater extent than would be the case if it were more rigidly confined.

Sparklets, Ltd., pay particular attention to the corrosion-resisting qualities of the wire, especially in cases where steel wire is used.

It is not possible in the space of a short article to enumerate the multiple uses to which the plaited wire can be put, but the following will give some indication of its application: reinforcement and protection of hose for air or other gases, oil feeds, etc., ignition lead covering, casing for flexible metallic shafts, and protection for glass containers.

Special attention is drawn to the employment of wire braiding in the electrical industry, where it has been found to be a very efficient medium for armouring and screening of electrical cables, flexible electrical contacts, earthing strips for electrical apparatus, live rails, etc. For these purposes, the braiding can be obtained in tubular form or in flat strip.

SECTION XXIII

PART II

Welding and Cutting.

(*Barimar Ltd., 22/24 Peterborough Road, Fulham, London, S.W.* 6.)

Contrary to the hopes of a few years ago we have not reached a period when there has been any easement in the overload carried by so many factories and engineering works in this country. The export drive has necessarily increased this pressure and, at the same time, it has caused the withholding of a great deal of new equipment that is needed to supersede obsolescent machinery and other plant that is due for replacement on account of wear.


Sparklets
WOVEN
WIRE ARMOURING
& BRAIDING
THE WOVEN WIRES
PROTECT THE CABLE
FLAT-STRIP

In these circumstances little imagination is required to appreciate that serious situations are often threatened by the breakdown of some key machine unless the fault can be put right with complete dependability and a minimum loss of time. It is in this connection that Barimar scientific welding service has been applied with strikingly successful results.

A particularly anxious period was during the fuel crisis when the rapid repair of a wide variety of plant operated by electricity and gas undertakings called for skilful handling following upon failure through breakage, wear and corrosion. Sometimes it is impracticable to deal with these needs in the shops of the welding specialist as bulk or weight make transport impossible. In instances of this kind the welding repairs are done on the site and specially trained operators with self-contained equipment tackle the problem on the location. By this means too a great deal of time is saved.

The experience gained during the war years with emergency repairs, many of which had been regarded hitherto as beyond the bounds of practical accomplishment, has provided an invaluable background of knowledge that is being applied and extended at the present time.

Not all repair work completed by Barimar on the site is of the type associated with boilers, the massive castings of steel rolling mills, hydraulic presses and so on. In a recent instance a very large tractor broke down whilst at work, for the main casting, which is also the backbone of the chassis, fractured suddenly, fortunately there was reasonable accessibility and no subsequent machining was required. So the job was tackled and successfully completed on the spot where the breakage happened. It was during a period of extremely cold weather and screens of sacking had to be erected as a protection from the wind. Results, however, fully justified the trouble taken and no further weakness has been experienced.

A vital branch of industry in which a great deal of outstanding welding work has been done is that which is concerned with the processing of food. Some of this involves the use of stainless steel. Certain containers have been vastly improved by lining with this material which is applied in strips welded together and to the main body of the container.

A striking piece of work of a different nature concerned the repair of diffusion cells used in the production of sugar from beet. In the great factories engaged upon the important task of making sugar, when once the process is started it is continuous. That is to say it proceeds night and day, month in and month out until the autumn harvest of beet is reduced to its required components of sugar and residue dried pulp which is a valuable cattle food. In these circumstances any hitch in the smooth running of the complex plant would be serious, to put it mildly. The diffusion cells, to which reference has been made, are iron castings resembling huge cauldrons. The diameter is 6 ft., the height 3 ft. 2 ins. and each unit weighs 2 tons. Not only was the metal cracked extensively but some of the mounting lugs, which originally were integral with the main casting, were broken away entirely. As no spares were available there must be no element of doubt as to an utterly dependable result and in addition it was emphasised that the work had to be completed by a certain date not far distant. Barimar achieved a perfect and invisible repair well within the stipulated time and issued their usual money-back guarantee.

Prior to 1939 welding repair work was becoming an increasingly valuable export. A wide variety of engine components were received for attention from all over the world. The items most frequently shipped were crankshafts (some of large size belonging to powerful diesel engines) cylinder heads and blocks with a considerable proportion of precision cut gears from which the teeth had been broken away and often lost necessitating the 'growing' of new teeth by scientific welding methods.

This work for overseas users is being resumed and it is increasing steadily. It might be thought that the cost of a double shipment together with the time factor would render such a scheme impracticable except on a restricted scale. In many instances it is possible to complete the repairs and re-ship within a week of the damaged parts being received. Sometimes a great deal of time is saved because of the lengthy delay in obtaining many important components such as crankshafts, large crankcases and other major items. Although the cost of re-conditioning is very small when compared with the price of replacement, even so, the greatest saving is often that which follows upon the additional weeks and months that are frequently required to supply a new part.

Typical of this class of repair was the top half of a crankcase belonging to a three-cylinder Diesel engine operating a pump for a vital water supply. The casting is of massive proportions as it carries the cylinder liners and provides the jacketing. Across the end of the casting and running for about a third of its length on one side, was a crack. In addition to a full measure of strength being restored perfect alignment was imperative for without it the repair would be useless; moreover any error or even an increase beyond the estimated date of completion would create an extremely serious situation. The welding was completed well within the period promised, subjected to pressure tests far in excess of those reached during normal running conditions and has since fully justified the confidence that the owners had in the ability of Barimar to handle the difficult task successfully.

Equally exacting in its demands to the repair just described was the work carried out recently on the cylinder of a power hammer. The casting stripped weighs 5 tons and measures 10 ft. by 7 ft. and is 3 ft. 6 ins. across. The trouble was a crack extending more than half way up the bore

and production in a steel works was being held up pending repair. The welding was done horizontally but, if needs be, vertical and even overhead welding of cracks can be tackled successfully.

A feature of this job was that so neatly was the welding carried out that re-boring was not necessary and the casting was set up and put to work just as it was received from the hands of the Barimar operators.

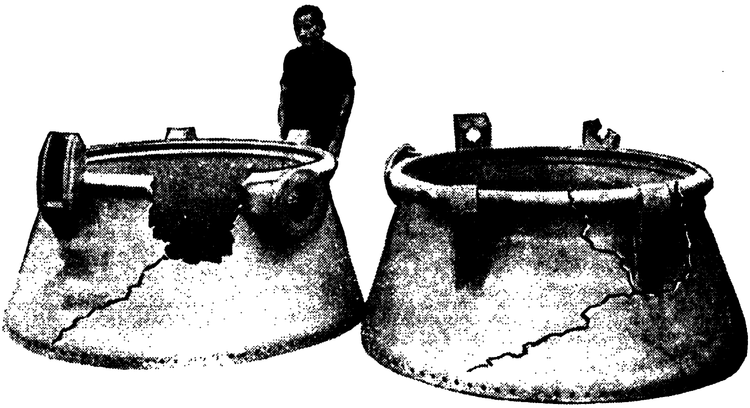


FIG. 1.—Two badly broken and cracked diffusion cells used in sugar beet manufacture. Weight 2 tons each.

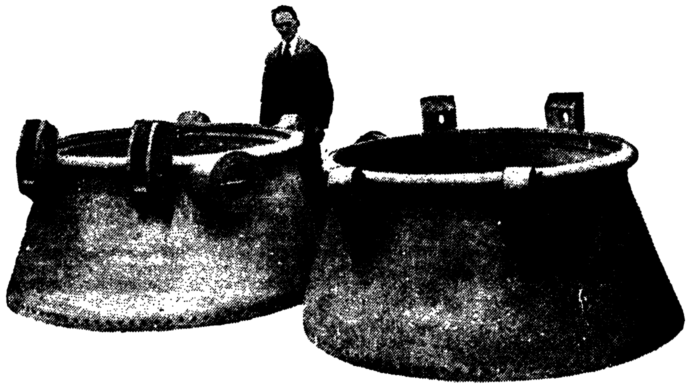


FIG. 2.—The 'cauldrons' after successful repair by scientific welding.

In these days of urgency scientific welding is accomplishing repair work the value of which it is impossible to compute. Sometimes the chances of a successful outcome seem so slender that some owners regard a breakdown as being hopeless from the start. This is not wise, for so remarkable is the progress that has been made that work regarded as being virtually impossible not long ago has now become almost commonplace.

MUREX ELECTRIC ARC WELDING EQUIPMENT AND ELECTRODES.

(Murex Welding Processes Ltd., Waltham Cross, Herts.)

A comparatively low voltage is required for electric arc welding, *e.g.* 50–60 volts if direct current is used at the arc, or 80–100 volts for alternating current supplies. In order to convert efficiently the power from an electrical supply to meet the requirements of arc welding, a motor generator equipment or a transformer must be used. These equipments, while reducing the mains voltage to that required for welding, must also have special volt/ampere characteristics to ensure arc stability. Further, a range of such characteristics is necessary so that a choice of welding current is available.

A typical volt/ampere characteristic curve of Murex generators and transformers is shown in fig. 1. The actual welding voltage in the case of metallic electrodes is generally between 20–35 volts, according to the size of electrode and current used. The curve shows a 'drooping characteristic,' *i.e.* as the load increases, the voltage 'falls' from the open circuit value to that required for

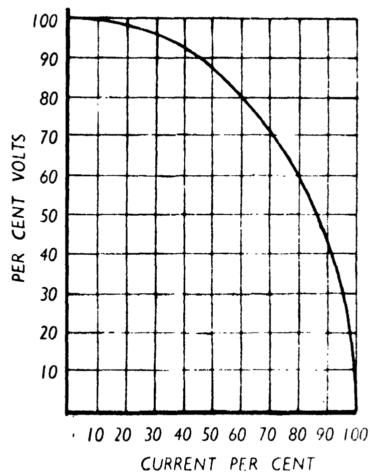


FIG. 1.—Typical Volt/Ampere Characteristic Curve of Murex Generators and Transformers.

welding. This characteristic is achieved by the design of the generator, or in the case of the transformer, by the design of the regulator or reactance.

The current required for welding is dependent upon the size of the electrode used, the following values being approximate for all makes and types :—

s.w.g.	16	14	12	10	8	6	4	5/16 in.	3/8 in. Electrode
Current	25	70	90	125	170	210	320	450	540 Amperes

The size of the electrode used is dependent on the plate thickness, the type of joint and welding technique employed. For example, a 16 or 14 s.w.g. electrode may be used on ⅛ in. thick plate, while on a 1 in. thick plate, a number of welding runs using perhaps 8, 6 and 4 s.w.g. electrodes may be necessary.

In a general engineering shop, a welding unit having an output of 15/400 amperes can therefore be considered essential.

Fig. 2 shows a typical 300/400 ampere Murex motor generator welding equipment. The motor and generator are built into one frame, the rotor of the motor and the armature of the generator being on the same shaft. The generator is of patent design having bifurcated poles which make possible a unique form of flux distribution. By this arrangement, the correct characteristic is obtained on a self-excited machine. The electrical efficiency is high and the time for recovery in voltage from short circuit to full open circuit is reduced to a minimum.

Two operator motor generator sets are also available and in this case, the two generators are built into one frame on which is mounted the control gear, the generators being coupled directly to a suitable motor. On a double operator equipment of this nature, provision is made for paralleling the two generators by means of a paralleling switch. Advantage can therefore be taken of the combined

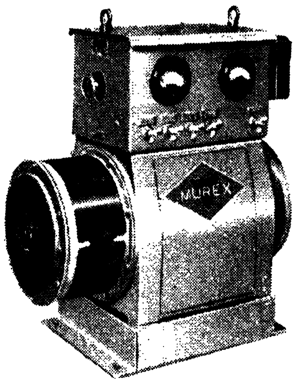


FIG. 2.—300/400 Ampere Motor Generator Equipment.

outputs of the generators to supply one operator on heavy work where large gauge electrodes are being employed for high-speed welding.

Where alternating current supply is available, the conversion of the electical power can be obtained by means of a transformer instead of a motor generator set. The portable unit illustrated in fig. 3 has an oil immersed transformer and regulator combined in one tank, space being

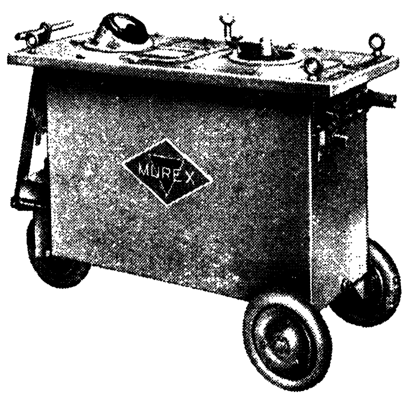


FIG. 3.—Portable 250 Ampere Transformer Equipment.

provided for inserting a capacitator for power factor correction. This equipment is designed to supply one operator with a current of 15 to 250 amperes for continuous hand welding.

Fig. 4 illustrates a three-operator transformer of the stationary type with regulators suitable for outdoor use.

All Murex transformers have a secondary open circuit voltage of 80 volts. Tappings are provided so that 100 volts can also be obtained specially for thin plate welding or for use in cases where the characteristics of the electrode require this voltage to maintain a stable arc.

On constructional work and sites where no electrical supply is available, petrol or diesel engine driven equipments are necessary. Fig. 5 illustrates a two operator unit, driven by a diesel engine running at 1,500 r.p.m., giving an output of 300 amperes per operator.

Motor generators, transformers and engine-driven equipments are available with outputs up to 600 amps per operator. All equipments can be arranged as stationary or portable units to meet the various site conditions.

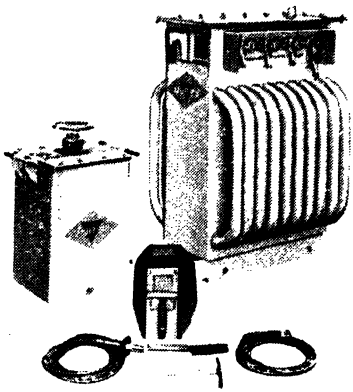


FIG. 4.—Multi-operator Static Transformer Equipment.

A comprehensive range of welder's accessories is also available including electrode holders, helmets and screens, spectacles and goggles, protective clothing in leather and asbestos, respirators, wire brushes, chipping hammers, flexible cable and tong test ammeters, etc.

Murex electrodes are manufactured for all classes of steel and non-ferrous metals. A number of special types are available for stainless steel, for bronze and for applications where hard surfaces

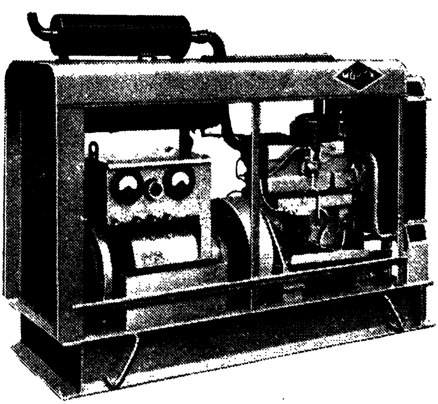


FIG. 5.—300/400 Ampere Diesel Engine Driven Equipment.

are required. This range of electrodes includes types approved by the British Admiralty and the Classification Societies for the welding of mild steel, high-tensile steels and also for pressure vessels. Special courses of instruction for welding operators, supervisors and designers are provided in the Murex Welding School attached to the manufacturing and research departments at Waltham Cross, Herts, England.

SECTION XXV

Fuels: Solid, Liquid and Gaseous.

Pulverisers.

THE CLARKE-CHAPMAN 'RESOLUTOR' PULVERISER.

(Clarke, Chapman & Co., Ltd., Gateshead on-Tyne.)

This pulveriser, see fig. 1, is a self-contained unit machine and obtains its product by the impact of renewable beaters attached to a rapidly revolving disc. The coal is fed into the machine from a hopper by a belt, the speed of which can be readily controlled. Entering the pulverising chamber the coal is caught up by the high-speed beaters, shattered to an extreme degree of fineness and

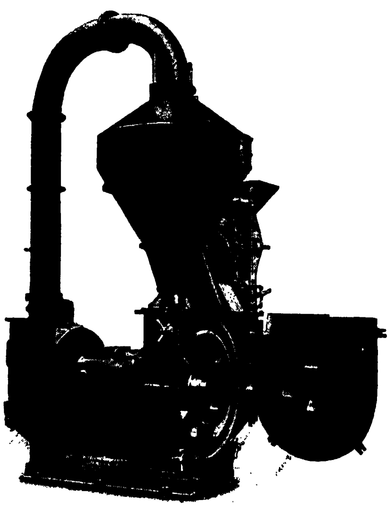


FIG 1.

drawn up by means of the action of a fan into a separating chamber. In this chamber an adjustable deflector returns any coarse coal for re-pulverisation, whilst the fine material is drawn off into the firing circuit. The 'Resolutor' pulveriser in sizes ranging up to 5 tons per hour is now successfully firing water-tube boilers, Lancashire boilers, marine boilers and furnaces of all descriptions. This pulveriser is also in use for firing cement kilns and for powdering various materials for industrial purposes.

THE CLARKE CHAPMAN RAYMOND BOWL MILL.

This pulveriser (fig. 2) comprises a milling unit and an exhauster fan both of which are driven by independent motors while the exhauster fan motor has variable speed control.

The milling unit consists of a slow speed revolving bowl upon the sides of which the coal is crushed to powder by rollers which have adjustable spring loading.

The upper portion of the mill casing embodies a classifier with adjustable louvres and deflector which allow only the finest of the fuel to pass to the exhauster fan and thence to the burners, while the coarse particles are returned to the bowl for further reduction. The raw coal being

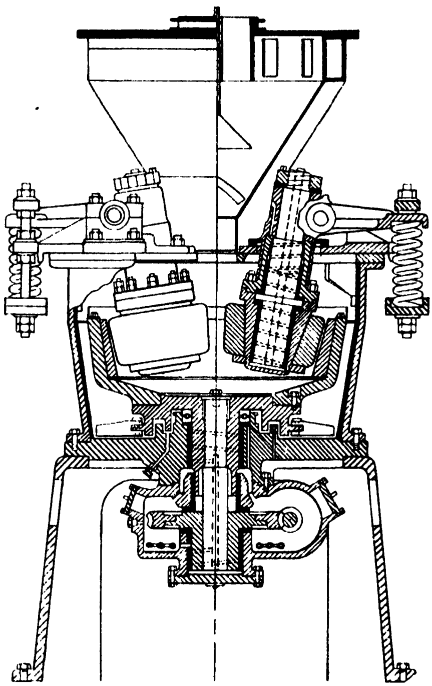


FIG. 2.

fed into the mill is mixed with the dry coarse returns from the classifier and due to this and the application of hot air, fuel containing a high percentage of moisture can be milled without difficulty.

To resist the wearing action of the coal, the bowl liner and crushing rollers are made of special metal while the casing is fitted with liners where necessary.

The exhauster fan is of robust construction with renewable volute sections.

The mill and fan units are so constructed that replacements can easily be made.

The adjustments of the crushing rollers and classifier controls can all be made from the outside while the plant is in operation.

The lubrication of the worm drive and bearings is by means of an enclosed circuit which ensures all parts being continuously lubricated.

The milling unit exhauster fan can be arranged so that both are driven by the same motor or each driven by separate motors in which case the exhauster fan motor can be of the variable speed type.

Pulverised Coal.

PULVERISED COAL FIRED BOILERS.

(International Combustion, Limited, Nineteen Woburn Place, London, W.C. 1.)

Lopulco Boilers.

The evolution of the Lopulco 'open furnace' type of boiler has taken place along with the progressive development of pulverised fuel firing and the result embodies all those features necessary to a combination of high availability and high efficiency.

Boiler and furnace are an integral steaming unit, the furnace being of such a shape as to provide maximum heat absorbing surface in relation to its cubic content. Full furnace cooling either of fin or plain tube construction is employed down to the ash hopper discharge and the burners are so positioned as to provide maximum temperature gradients without hot zones conducive to slag deposition.

The open outlet allows the use of a divided superheater, the front section of which is the semi-radiant wide pitched type and the need for baffles throughout the convection section of the boiler is reduced to a minimum if not dispensed with altogether. Dust pockets are provided before and after the lower drum so as to reduce all possibility of dust accumulation.

The circulation of the boiler is so arranged that the rear drum which is elevated in relation to the front has no active steam generating tubes entering, thus producing conditions most favourable to exceptional steam purity. It is thus possible to allow high rates of steam generation per foot of furnace width with corresponding high transfer rates and economy in construction.

The unit is suitable for the employment of the highest practicable air temperatures and is naturally suited to the high feed temperatures employed with multi-stage feed heating. High moisture and high ash content fuel can, therefore, be used to maximum advantage. This type of unit can be built for all ranges of pressure and evaporation but is naturally suited to high pressure and temperature conditions for high evaporations. Variants of this basic design can be offered with one or more drums for either pulverised or stoker firing to meeet any combination of fuel and steaming conditions which may be required.

The use of fin tubes for the furnace construction allows of the complete elimination of refractory materials on the fire side of the furnace and the minimum use of insulating medium. The Lopulco method of drum slinging is also applied where drum sizes are such as to set up binding stresses in the shell. Sling bands are forged integral with the shell in such positions as to reduce bending stresses to zero with the result that there is often an appreciable reduction in thickness and weight —a most important advantage for high pressure conditions.

Pulverised Fuel Firing.

The modern and almost universally used system of pulverised fuel firing is the 'Unit' system (fig. 2, p. 1190), where one or more mill units are direct coupled to the furnace burners and receive their drying medium direct from the boiler unit. When pulverised fuel firing was first introduced, the bunker system was used in which the fuel was dried (often in separate machines), pulverised, and delivered to a storage bunker from which it was fed to the burners by a multiplicity of feeders. This system had many drawbacks and the improvement in reliability and size development of the milling units together with the adoption of simultaneous drying and pulverising of the fuel has meant the virtual abandonment of the bunker system except for such classes of coal which make it essential to separate the burning and pulverising operation. In this country it is now only used for anthracite coal firing. In a similar process of evolution the use of the arch type of firing where the burners projected the coal and air mixture down into the furnace bottom with a U sweep up to the boiler outlet, has been virtually abandoned except for anthracite coals and replaced by horizontal burners.

In this type of burner the primary air and coal stream is surrounded by an envelope of secondary air in quantity sufficient to secure complete combustion. Centrifugal motion is imparted to these streams to give the widest possible air contact to the coal particles.

The burners can be arranged for forced draught and induced draught, and depending on the configuration of the furnace these burners may be arranged in one wall or in pairs in two opposed walls across the depth or width of the furnace.

A maximum degree of furnace turbulence can be secured by setting the burners in the corners of the furnace and directing the flame path to an imaginary circle in the centre of the furnace, With this arrangement the zone of maximum heat release and absorption is spread over the widest possible area and the minimum furnace exit temperature assured. Important advantages can be gained by making the corner burners controllable as regards elevation so that the flame zone can be raised or lowered at the will of the operator providing, in effect, an 'adjustable furnace,' in that the volume and absorption surfaces can be adjusted to the requirements of any grade of coal. By this means regulation of the furnace gas leaving temperature is possible, giving a primary control on the steam temperature and substantial control over slagging conditions both at the top and the bottom of the furnace.

Milling Plant.

The most commonly used type of mill for pulverised fuel firing is the medium speed ring roll mill.

In the Lopulco mill of this type, two large diameter rolls revolve on fixed axes against a horizontal driven table. The rolls are connected by a powerful spring gear which ensures equalisation

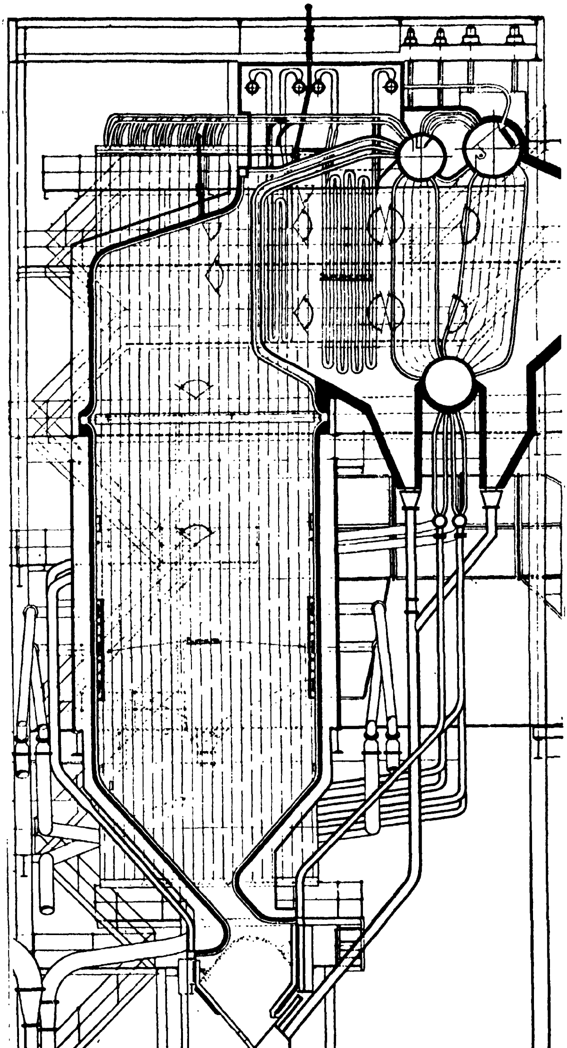


FIG. 1.—'Lopulco' Boiler.

of grinding effort and produces minimum power consumption. The flat grinding table allows for reversal after wear to obtain maximum use of metal before rejection. The driving unit is a self-

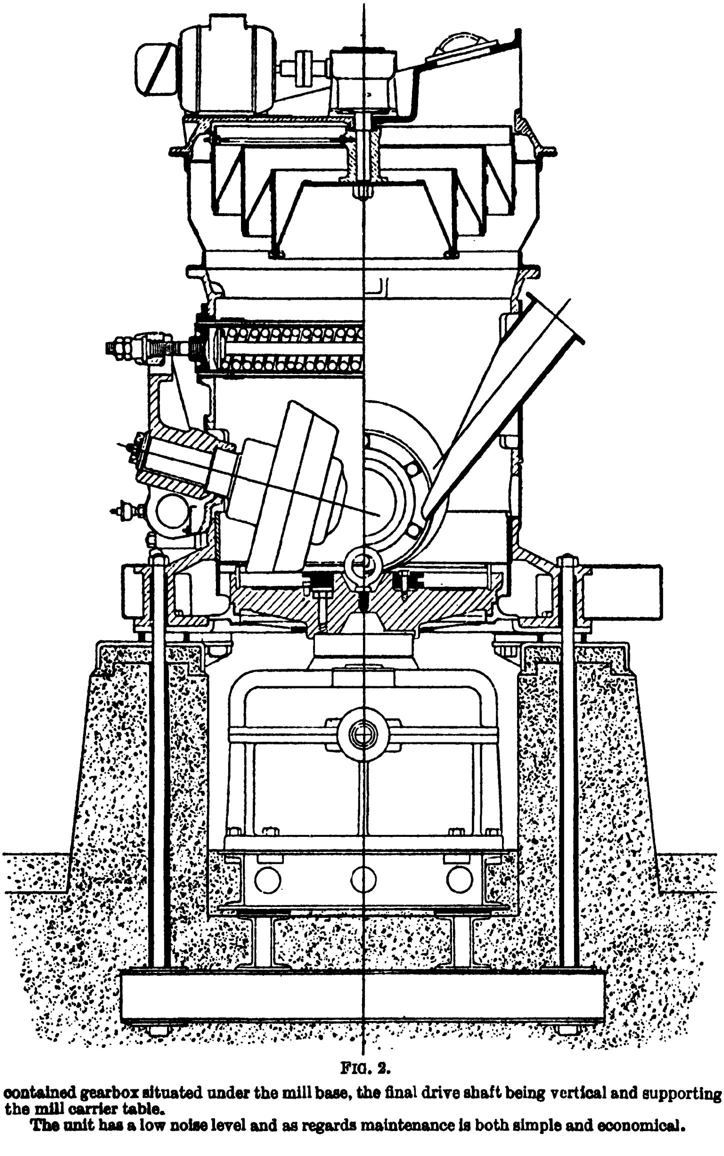


FIG. 2.

contained gearbox situated under the mill base, the final drive shaft being vertical and supporting the mill carrier table.

The unit has a low noise level and as regards maintenance is both simple and economical.

Separation is controlled by a rotary type separator housed directly over the grinding chamber and is of the variable speed type.

The grinding unit is completed by an exhauster fan which may be of the variable speed type or direct coupled to the mill drive.

Made in a number of sizes for outputs ranging between 2 and 16 tons per hour, this unit is particularly suitable for sustained peak running in power stations. Where peak loads are infrequent, high speed mills are often suitable especially for industrial boiler plants. The Impax Mill of this category is of the usual swing hammer type with coupled exhauster. A special feature is the variable speed rotating classifier, which gives a greatly improved fineness range over the static type. Higher in power and maintenance than the Lopulco Mill, it is low in capital cost and particularly suitable for firable coals.

The I.O. Ball Mill operates on the straight-through system with oversize return to the feed end. It is especially suited to grinding anthracite and refractory coals and has low maintenance. Its noise level and power consumption are high. When fitted with variable speed exhauster it can be used for unit firing.

Refuse Disposal Plants.

(Heenan & Froude Limited, Worcester.)

A few years ago the usual method of dealing with domestic refuse collected from houses, etc., was by complete incineration as an alternative to tipping on the land or dumping into the sea. A period followed in which the refuse was sorted over and materials of value salved and sold, the remainder being burnt or tipped. At the present time, in addition to these methods, screening machinery is provided which separates the dust from the refuse and enables cinders, tins, and other material of value to be picked out for sale whilst the remainder is burnt in an incinerator.

Where desirable, steam is raised for power purposes from the burning of the refuse, and electrical current may be generated, as in the large refuse destructor plant of the Glasgow Corporation at Govan, particulars of which are given elsewhere in this book.

In the diagram (p. 1192) are shown the six main stages of the 'Heenan System' of refuse disposal and various plants have been erected incorporating all or some only of the stages according to the local requirements or conditions.

Stage 1 comprises the delivery of the crude refuse from the refuse collecting vehicles into the receiving hopper, the travelling bottom of which discharges through a regulating device the right amount of refuse to supply the elevator or conveyor feeding the rotary screen.

In the screen, the dust is separated from the refuse and falls into a hopper underneath from which it is withdrawn as required and taken to the local tip or otherwise disposed of.

Stage 2 of the system comprises the removal of the cinders and small debris in the second part of the screen.

This material is also collected in a hopper and withdrawn as required. The cinders and debris may be treated to a further process in which the latter is separated, leaving clean cinder only and carrying the debris on to the incinerator for burning.

Stage 3 separates the iron and tin cans from the tailings leaving the screen, by means of electro-magnetic separation. The tins, etc., are picked up on the end plate of the screen, and, as they are carried round out of the magnetic field, drop off and fall down a chute to the baling press below where they are pressed into handy size bales weighing from 30 to 50 lb. each. These bales can be neatly stacked and stored until required for melting down at a steel works or treating otherwise.

In stage 4 the tailings from the rotary screen are conveyed on a slow-moving belt from which any further material of value is picked off by hand. This salvaged material is stored and disposed of at intervals. The remainder of the tailings pass on to stage 5, viz., to the incinerator into which they are fed by means of a movable tripper and chute for destruction. This tripper enables the cells to be automatically fed but when desired, or necessary, the tripper can deliver the refuse on to the feeding floor from which it can be hand shovelled into the top feed openings of the cells.

The incineration of the refuse produces hot gases which pass into the boiler in stage 6 of the system. A water tube type is the most suitable owing to its relatively large heating surface and free gas area.

These gases generate steam which can be usefully employed in the production of electrical power or for pumping water or sewage.

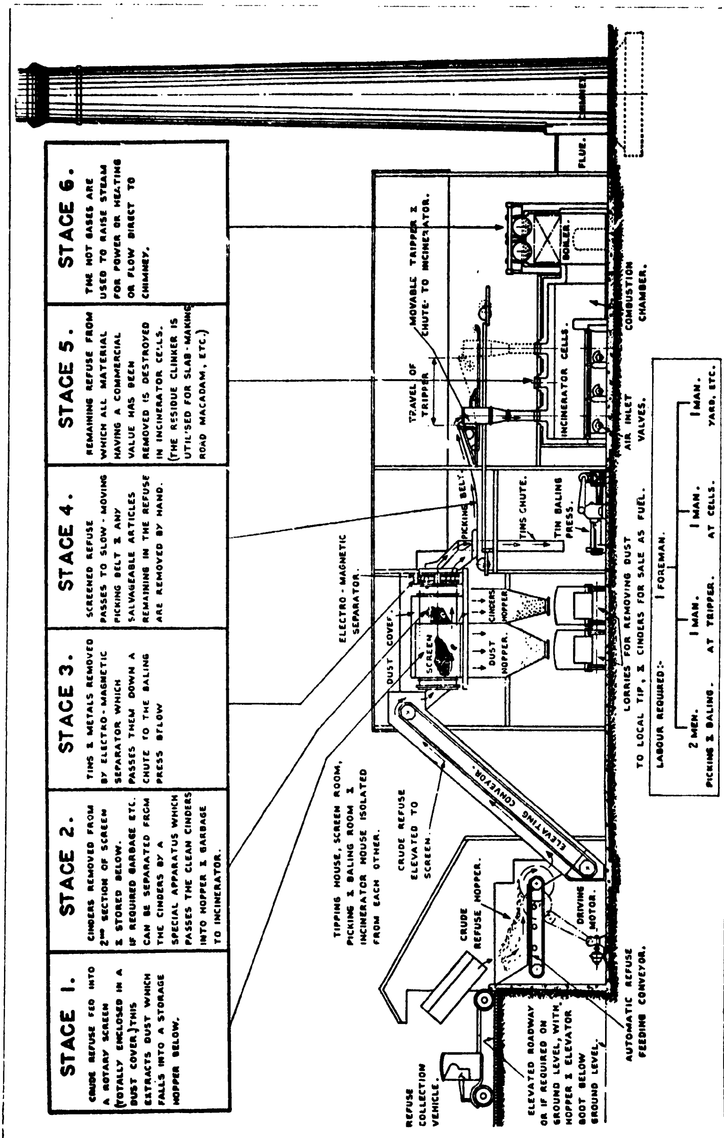
In many plants of the 'Heenan System' the first four stages are omitted and the whole of the refuse with the exception of the metals is incinerated for the full production of power.

Two pounds of steam and upwards can usually be obtained from each pound of refuse burnt and therefore an appreciable amount of power can be produced from the refuse of a fair-sized township with the added advantage that the objectionable and unstable refuse (which has in any case to be disposed of) is reduced to a valuable innocuous clinker.

From the boiler the gases pass into a chimney which has to be of sufficient height and area to overcome the gas resistance of the boiler, flues and furnace and to prevent back pressure in any part of the plant.

As each 1,000 persons produce about 15 cwts. of domestic refuse per day the plant illustrated would, if worked 8 hours only per day, be suitable for a township with a population of about 50,000 persons.

If worked for 16 or 24 hours the capacity of the plant could be correspondingly increased, but this would necessitate the provision of storage space for the refuse owing to the collection being usually restricted to one shift of 8 hours.


STACE 1.
CRUDE REFUSE FED INTO A ROTARY SCREEN (TOTALLY ENCLOSED IN A DUST COVER.) THIS EXTRACTS DUST WHICH FALLS INTO A STORAGE HOPPER BELOW.
STACE 2.
CINDERS REMOVED FROM 2ND SECTION OF SCREEN & STORED BELOW. IF REQUIRED GARBAGE ETC. CAN BE SEPARATED FROM THE CINDERS BY A SPECIAL APPARATUS WHICH PASSES THE CLEAN CINDERS INTO HOPPER & GARBAGE TO INCINERATOR.
STACE 3.
TINS & METALS REMOVED BY ELECTRO-MAGNETIC SEPARATOR WHICH PASSES THEM DOWN A CHUTE TO THE BALING PRESS BELOW
STACE 4.
SCREENED REFUSE PASSES TO SLOW-MOVING PICKING BELT & ANY SALVAGEABLE ARTICLES REMAINING IN THE REFUSE ARE REMOVED BY HAND.
STACE 5.
REMAINING REFUSE FROM WHICH ALL MATERIAL HAVING A COMMERCIAL VALUE HAS BEEN REMOVED IS DESTROYED IN INCINERATOR CELLS. (THE RESIDUE CLINKER IS UTILISED FOR SLAB-MAKING ROAD MACADAM, ETC.)
STACE 6.
THE HOT GASES ARE USED TO RAISE STEAM FOR POWER OR HEATING OR FLOW DIRECT TO CHIMNEY.
TIPPING HOUSE, SCREEN ROOM, PICKING & BALING ROOM & INCINERATOR HOUSE ISOLATED FROM EACH OTHER.
ELECTRO-MAGNETIC SEPARATOR.
DUST COVER.
SCREEN
CRUDE REFUSE ELEVATED TO SCREEN.
REFUSE COLLECTION VEHICLE.
CRUDE REFUSE HOPPER.
ELEVATING CONVEYOR.
TRAVEL OF TRIPPER
MOVABLE TRIPPER & CHUTE TO INCINERATOR.
PICKING BELT.
DUST HOPPER.
CINDERS HOPPER.
TINS CHUTE.
TIN BALING PRESS.
INCINERATOR CELLS.
BOILER.
FLUE.
CHIMNEY.
ELEVATED ROADWAY OR IF REQUIRED ON GROUND LEVEL, WITH HOPPER & ELEVATOR BOOT BELOW GROUND LEVEL.
DRIVING MOTOR.
AUTOMATIC REFUSE FEEDING CONVEYOR.
LORRIES FOR REMOVING DUST TO LOCAL TIP, & CINDERS FOR SALE AS FUEL.
AIR INLET VALVES.
COMBUSTION CHAMBER.
LABOUR REQUIRED:- 1 FOREMAN.
2 MEN. PICKING & BALING.
1 MAN. AT TRIPPER.
1 MAN. AT CELLS.
1 MAN. YARD, ETC.

Gas Recorders.

(The 'W.R.' Patent CO_2 Indicator.)

(International Gas Detectors, Ltd. (Late W.R. Patents, Ltd.), Great Wilson Street, Leeds.)

The W.R. CO_2 or Combustion Indicator is an apparatus which continually, automatically, and accurately indicates the percentage of CO_2 which is passing through the flues at any hour of the day and night. With this Indicator installed on the boiler plant the percentage of CO_2 obtained can be seen at once, and the necessary steps taken to increase the amount and thus save fuel.

Fig. 1 shows the diagrammatic arrangement of the essential parts of the device. It can be used in all cases of combustion where any class of fuel is used, and where CO_2 is a product of this combustion. The arrangement adopted differs materially from other CO_2 indicators.

The method employed in the 'W.R.' CO_2 Recorder depends on the difference of pressure generated on the two sides of a porous pot, one side of which is exposed to flue gas whilst the other contains soda lime granules contained in a paper cylinder which absorbs the coming CO_2.

An aspirator, S, worked by the natural draught which it increases, is fixed at a point as near the chimney base as possible, and continually aspirates gases from the flue it is desired to keep under observation. The path of gases is shown by the arrows. They are first drawn through a filter F, and pass from thence into a chamber containing a porous pot, inside which is a dry reagent, R. The flow of gas is shown by continuous bubbling through water in vessel

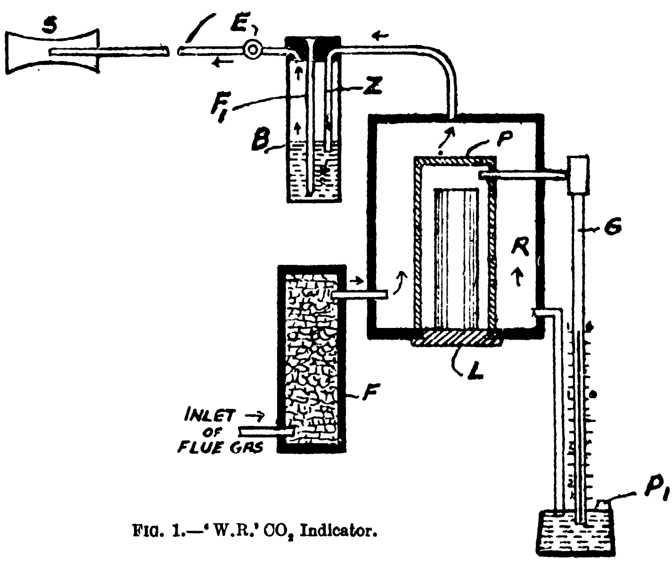


Fig. 1.—'W.R.' CO_2 Indicator.

B. A pipe connects the chamber with a vessel containing water into which dips one end of another pipe, G, the other end of which is taken into the inside of the porous pot. Some of the gases penetrate into the interior of this pot, and are absorbed by the reagent, with the result that a partial vacuum is formed. and the water is forced up the pipe G. The latter is provided with a scale so graduated that the percentage of CO_2 in the flue gases can be read.

The W.R. indicator is prompt in its action, having a lag of one minute only. Its deviation from absolute accuracy has been found to be only 0·1 per cent. It is well constructed; has no mechanical moving parts; is fireproof and foolproof, and can be worked by any boy of average intelligence. With each Indicator the manufacturers furnish a blue print showing the method of fitting to each particular boiler, also special detailed instructions for installing the apparatus, which operation can be completed in from four to six hours by any works' engineer or fitter. In cases where a recording graph is desired, one of a specially approved form can be supplied by International Gas Detectors, Ltd. The cartridges last 24 hours of continuous working.

SECTION XXVI

Electrical Engineering.

Electrical Instruments.

Insulation and Resistance Testing Instruments.

(*Evershed & Vignoles, Ltd., Acton Lane Works, Chiswick, London, W. 4.*)

The majority of electrical breakdowns and accidents are caused by defective insulation. Insulation failure can generally be avoided by systematic testing with a Megger or Meg Insulation Tester, but to guard against the dangers of a sudden accidental failure, the continuity of the earthing circuit should also be tested.

The values of insulation resistance and the tests required are laid down in the British Standard Specification for the particular apparatus, and the Institution of Electrical Engineers Regulations for the Electrical Equipment of Buildings (eleventh edition) gives the corresponding values of insulation and continuity resistance for completed installations and the tests required to verify these.

The principle of operation of the wee-Megger, Meg and Megger Insulation Testing Sets, described below, is represented in fig. 1. Each instrument contains a hand-driven direct current

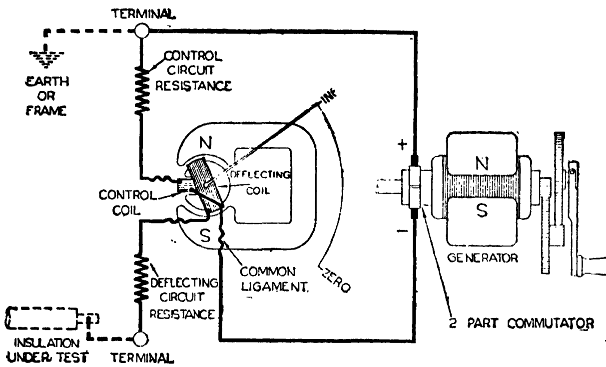


FIG. 1.

generator to supply the testing voltage, and a direct reading ohmmeter for measuring the value of the insulation resistance. The ohmmeter movement consists of two coils, the control coil and the deflecting coil, mounted at a fixed angle to one another on a common axle. Both coils are in parallel across the generator, the control coil being in series with a fixed resistance, and the deflecting coil in series with a second fixed resistance and with the resistance under test. The ratio of the currents in the two coils will therefore depend solely on the value of the resistance under test, since variations in the pressure generated, due to varying handle speeds, affect both coils in the same proportion. The deflection of the pointer depends on the ratio of the currents in the two coils and hence gives a true measurement of the resistance under test, the dials being calibrated in megohms and thousands of ohms.

The Meg and Megger Testing Sets (figs. 3 and 4) can be supplied with generators of the constant pressure type, having a special centrifugal friction clutch which ensures a constant testing voltage. Instruments fitted with this device should be used when testing the insulation of installations having considerable electrostatic capacity.

The ranges and uses for which the various types of instrument are recommended, are set out below.

Instruments for Measuring Insulation Resistance only.

The wee-Megger Tester (fig. 2) is suitable for testing house wiring, small motors, etc., operating on voltages not exceeding 250 volts.

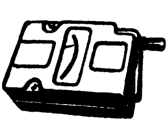


FIG. 2.

Testing voltages up to 500 volts.
Ranges up to 20 megohms.
Size 5¾ × 4 × 2½ in.
Weight 3 lb.

The Meg Insulation Tester (fig. 3) is recommended for testing power circuits, motors, etc., operating on 500 volts and for testing mains having moderate electrostatic capacity.

Testing voltages up to 1000 volts.
Ranges up to 2000 megohms.
Size $5\frac{1}{2} \times 9\frac{1}{4} \times 6\frac{1}{4}$ in.
Weight 7–9 lb.

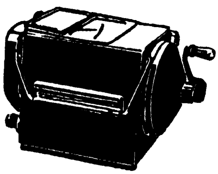


FIG. 3.

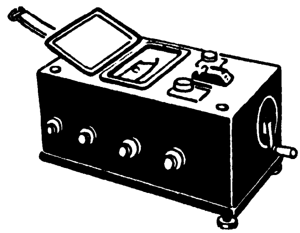


FIG. 4.

The Megger Testing Set (fig. 4) is recommended for testing high-tension equipment, transformers, mains, etc., and apparatus having a high degree of insulation and considerable electrostatic capacity.

Testing voltages up to 2500 volts.
Ranges up to 10,000 megohms.
Size $14 \times 8 \times 8$ in.
Weight 20 lb.

Special high range instrument :—

Testing voltage 5000 volts.
Range 20,000 megohms.
Size $18\frac{1}{2} \times 13\frac{3}{4} \times 10$ in.
Weight 60 lb.

INSTRUMENTS FOR MEASURING INSULATION AND CONDUCTOR RESISTANCES.

The Meg Insulation and Continuity Tester is similar to the Meg Insulation Tester, but contains a second scale of 0–100 ohms and a changeover switch enabling both the insulation resistance of an installation and the continuity of the lead sheathing or tubing to be measured.

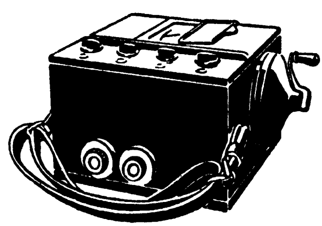


FIG. 5.

The Bridge-Meg Testing Set (fig. 5) combines in one case a constant pressure Meg Insulation Tester and a complete Wheatstone Bridge. It covers a wide range of resistance and is a suitable instrument for the maintenance engineer.

Testing voltages up to 1000 volts.
Range covered 0·01 ohm up to 200 megohms.
Size $7 \times 8\frac{3}{4} \times 12$ in.
Weight, $12\frac{3}{4}$ lb.

Mains operated Ohmmeters for routine measurements. The ohmmeter movements are mounted in switchboard pattern cases and arranged for operation on direct current mains or on alternating current mains through rectifiers.

Instruments for Measuring Earth Resistance.

The resistance of earth plates and other buried metal structures are of very great importance both from the point of view of the efficient operation of a system and of safety. The resistance must be maintained at a low value and should therefore be periodically measured.

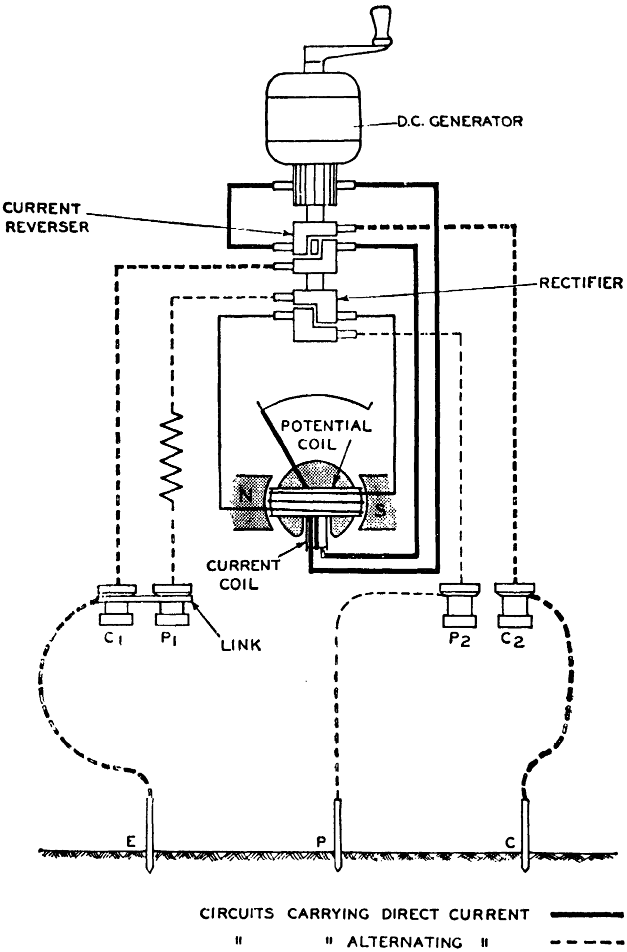


Fig. 6.

The Megger and Meg Earth Testers, which are similar in external appearance to the Megger and Meg Insulation Testing Sets, are used for this purpose.

Each instrument contains a hand-driven direct current generator and an ohmmeter of the moving coils type (fig. 6), one coil of which carries a current proportional to the testing current

and the other coil a current proportional to the potential difference between the earth electrode under test (E) and earth. The deflection of the ohmmeter which is proportional to the ratio of these currents gives the earth resistance directly in ohms. To avoid the effects of back E.M.F. and stray currents in the soil, a rotary current reverser and rectifier are incorporated so that the current in the soil is alternating whereas that in the ohmmeter is direct.

INSTRUMENTS FOR MEASURING LOW RESISTANCES ONLY.

These instruments each contain a direct reading ohmmeter of the moving coils type as fitted in the Megger and Meg Insulation Testers. but operate on low voltage.

The Megger Circuit Tester (fig. 7) which is a small direct reading portable ohmmeter operated from a self-contained 4½-volt dry battery, is suitable for measuring conductor resistance, for testing motor windings and continuity of sheathing, and for tracing circuits.

Range covered, 0–200,000 ohms.
Size 5⅝ × 4 × 2¼ in.
Weight 2 lb.

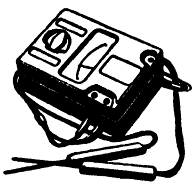


FIG. 7.

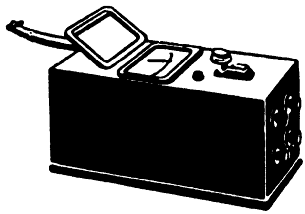


FIG. 8.

The Ducter Low Resistance Testing Set (fig. 8) is a direct reading portable instrument operating from an external alkaline battery. It is provided with a range switch giving four or five different ranges, and measures from 1 microhm to 5 ohms. It is suitable for testing switch contacts, rail bonds, windings of series motors, etc.

INSTRUMENTS FOR WATER TESTING.

The electrical conductivity of a dilute solution is proportional to the amount of inorganic impurity which is present in the water and thus can be used as an indication of the amount of the impurity.

The Dionic Water Tester is a portable equipment by means of which the conductivity of a sample of water can be quickly and accurately determined. Since this conductivity varies considerably with temperature, means are provided to correct for temperature variations.

The presence of impurities in boiler feed water condensate, etc., is of very great importance and it is often necessary to have a continuous indication or record of the state of this water.

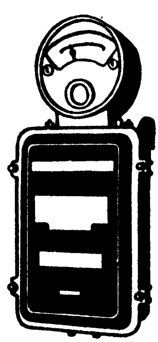


FIG. 9.

The Dionic Water Purity Meter or Salinometer (fig. 9) has been designed for this purpose, and has been extensively used both on board ship and in power stations for giving a continuous indication and chart record of the condition of the boiler feed water.

Coolers.

TUBULAR AIR COOLER.

(Heenan and Froude, Limited, Worcester, England.)

This cooler is suitable for general industrial applications calling for the cooling of air or gases without bringing them into direct contact with the cooling medium, and is widely employed in conjunction with the ventilation of alternators and motors. In the latter case the air circulates between air outlet and air inlet of alternator, in a closed circuit in which is placed the air-cooling unit. The latter consists of a nest of tubes held between tube plates, the tube joints being formed by ferrules and packing. On the air side the tubes are provided with highly efficient heat-exchange elements which are strongly soldered to the tubes, while the bores of the latter communicate with headers provided with baffles to promote the most efficient circulation of the cooling water.

For alternator work secondary tube plates are incorporated behind the main tube plates, the space between forming patented dead air spaces which trap any water which may leak from the tube ferrules and carry it away to drain, thus eliminating the possibility of loose moisture being carried through with the air into the alternator windings.

The water headers are normally fitted with inspection doors arranged so that they can be opened for access to the tubes for cleaning purposes without breaking the water piping joints.

To cover such emergencies as the cooling water supply failing, it is usual to incorporate in the ducting system inter-connected dampers by which the cooler can be isolated and the alternator caused to work on 'open circuit'; patented layouts of damper gear are available which can be incorporated in practically any installation.

Electro-mechanical Brakes.

(Elliston, Evans & Jackson, Ltd., 24 Ray Street, E.C. 1.)

This firm, which has a wide and long-established experience as makers of high-class electro-mechanical brakes for all varieties of application, considers that special attention should be paid to the following characteristics governing construction and design. They will be glad to deal with any problems arising from the use or prospective use or application of any form of electro-mechanical brakes.

Electro-mechanical brakes are used in connection with a variety of electrically driven machines such as lifts, cranes, hoists, elevators, conveyors, rolling mills, paper reeling, slipways, presses, machine tools, etc., where either a quick 'stop' or adjustable gradual 'stop' is desired, when the power is cut off, either intentionally or unintentionally.

These brakes consist primarily of an electro-magnet, which operates the brake blocks through a system of levers or other medium. The magnet is so arranged that when the current is applied the magnet is energised and lifts the brake blocks from the drum. If for any reason the current is cut off the magnet is de-energised and the brake blocks are pressed against the drum, the requisite pressure being obtained by springs, weights, or a combination of each.

The brakes are quite automatic in action; thus, in the case of an electrically driven crane, as soon as the driver applies the current to the motor to raise the load the brake is automatically released, and when he switches off the current to stop the brake is immediately applied. Should for any reason the supply current fail the brake automatically goes on and takes charge of the load.

It is desirable that an electro-mechanical brake should have the following features: compactness of design, robust construction, few and simple working parts, easy adjustment for brake shoes and pressure exerted by same. Coils should be liberally rated, well insulated, and protected from mechanical injury, and so placed that they are not likely to be damaged by oil exuding from the machine. The brake linings should be readily renewable. Brakes can now be obtained fitted with shoes which are self-adjusting for wear. These obviate the necessity of making frequent adjustments to compensate for the wear of the linings. This is a most important consideration when A.C. magnets are used, as these have a limited stroke. If springs are used, they should be of the compression pattern. Some form of dashpot, either separate from or embodied in the magnet, should be fitted which is adjustable in both directions to enable the braking effect to be gradually applied or released. Further, it is important to ensure that the shoe-linings and magnet are properly proportioned for the work to be done, *e.g.* a brake having a given torque when used on a busy dock crane would not be the most economical type when used as an emergency brake on some forms of conveyor or rubber machinery. In each case the magnet would be rated for continuous duty, but, whereas in the former case the shoes might be applied one hundred times or more per hour for several hours a day, in the latter they would probably be applied once or twice a day only.

Cable Reels.

SELF-WINDING ELECTRIC CABLE REELS.

NEW 'TRAILER' DESIGNS FOR MACHINE TOOLS

(Power House Components Ltd. King Street, Nottingham.)

For all electrically driven equipment involving position movement, including for example machine tools, the 'Wayne' self-winding electric cable reel has a number of advantages. In the standard form it consists of a metal rotating reel on which the cable winds or unwinds, under the operation of a spring mechanism in an enclosed casing, on the same principle as the spring blind. By this means suitable tension is maintained without any undue slack or sag.

These self-winding reels, a production of Power House Components Ltd., King Street, Nottingham, carrying the general trade mark 'P.H.C.' and under the direction of David Rushworth, are available in nine types, according to the length and diameter of the cable used, that is the 'Baby' (for ordinary twin lighting flex), 'Standard;' 'Major 10/3,' 'Major 12/3,' 'Super 12/515,' 'Super 15/6,' 'Super 19/4,' 'Super 19/9,' 'Goliath' and 'Mammoth.' The latter, however, being the largest, mostly used for loaders, cranes and quarry work, differs from the other designs in that the return motion, because of the great weight of the cables, 2½–3 ins. diameter, is sometimes carried out by hand-operation wind instead of a spring. For long travel machine tools the 'Super' (0·625–1·25 in. diameter) and the 'Major' (0·30–0·625 in. diameter) are chiefly used, as also for overhead cranes, runways, and magnetic cranes and welding apparatus.

In general the standard reels are attached to a ceiling, wall, girder, or convenient part of a machine by a flange plate, and when the pay-out is not in a straight line are of the swivelling type, turning on a pivot, with a cable guide. Housed at the side (or both sides) and totally enclosed is the slip-ring gear, consisting of a stout ebonite disc carrying the slip rings, varying in

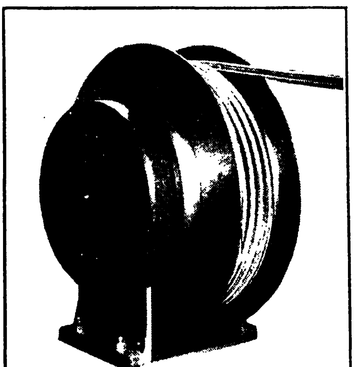


FIG. 1.

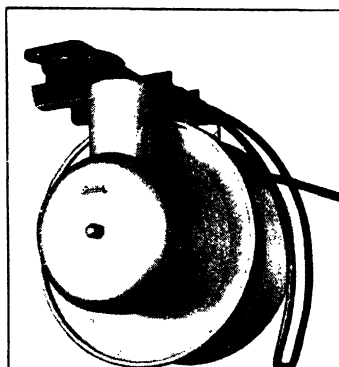


FIG. 2.

number to correspond to the cores of the cable, made of a special lead brass. The current pick-up standard plungers are of metal composition forming easily replaceable tips. They have conductivity as well as resistance to wear and facilitate maintenance work, whilst the spring, of the necessary size and strength to give ample margin for the duty, is of high-grade ribbon steel. Originally the pick-up plunger tips were of silver, but this is now used for light duty only.

Now there has been placed on the market a new 'trailer' design, available for the 'Major' and 'Super' sizes (0·5 in. and 0·7 in. cable respectively). This is a double acting or 'dual' reel, two sets of cable being accommodated and paid out in opposite directions. Consequently the new reel can be placed in the centre of the work, as in machine tools, for example, and double the amount of cable is handled because the pay-out action is in two directions at the same time, whilst electric sliprings and contact gear are eliminated. It is particularly suitable for long runways and similar applications. A modified design of roller swivel or round-about head has been

introduced which is suitable also for reels for protection against dust and combustible and explosive gases, which will take the complete range of reels up to the 'Mammoth' with 2 in. cables. This runs on a series of roller bearings with the casing attached to a flat top or cover, turning round at every angle, whilst the cable enters and leaves by a long gas-tight and dust-tight seal gland.

Much interest attaches to another modification, the totally enclosed type with a cable sealing gland in the baseplate which is suitable for operation in an explosive or inflammable atmosphere. Originally the enclosed design embodied a semi-circular steel cover totally covering the whole arrangement, the cable entering through a gas-tight gland at the top. Now, however, in the latest modification, the gas-tight gland is in the bottom cover with the cable entering or leaving from underneath, so that the cover can be removed for inspection and replacement without interfering with the cable and sealing joint.

Another specialised application is a motor turntable fire escape. For example, in one of the latest 'Merryweather' (London) escapes there is fitted for the telephone a 'Wayne' Super 19/4 type self-winding reel which handles 105 ft. of 4-core cable ⅜ in. overall diameter which operates the telephones at the top of the escape ladders, the cable winding or unwinding automatically as the ladders are elevated or the reverse, with no fear of entanglement or damage.

Finally mention may be made of still another modification, that is the 'Wayne' self-winding hose reel for dealing with rubber or canvas hose for general water supply. This is on exactly the same principle as for cables, and enables the hose always to be wound up neatly when not in use, without any trailing of long loose lengths of hose on the ground, subject to wear and tear and general deterioration. In each case the reel is designed for the specific conditions of an application, and any size of hose for use with normal pressures can be operated. A further application for these reels is the flame-proof contactless type for special 'foam' and other fire extinguishing liquid, using rubber hose up to 1½ in. diameter.

For flame-proof requirements or conditions where grit and moisture are found in such quantities as to make normal slip-ring gear unsuitable, a special range of Wayne Reels has also been introduced under the heading of 'Wayne' trailer type reel, which can be supplied in both moving and fixed designs, and also the Wayne twin-coil type reel; this is especially suitable for short lengths of cable.

With the era of post-war conditions many new demands are being set up for these reels, and a number of new designs are about to be placed on the market to cope with specialised requirements, and it is interesting to know that many of the designs will be readily adaptable to meet a wide variety of requirements which are envisaged in the next few years.

To assist Works' Safety Officers a Consulting Service is now available under the personal supervision of Mr. David Rushworth, the patentee and designer of Wayne Reels.

SECTION XXVII

PART II

Steam Generating Plant.

Boiler Design—Superheaters—Stokers—Electrostatic Precipitation—Feed Water Treatment—Scale and Corrosion—Boiler Fittings—Valves—Packing.

THE CLARKE-CHAPMAN WATER-TUBE BOILER.

(Clarke, Chapman & Co., Ltd.)

A Clarke-Chapman Water-tube Boiler is illustrated in fig. 1 (p. 1202), and consists of four drums and three banks of water-tubes, the top and bottom drums being connected together by steam and water circulating tubes. All the tubes are expanded directly into the drums.

The large and accessible superheater chamber between the first and second bank of water-tubes, contains a nearly vertical suspended superheater, consisting of a number of closely spaced multiple-loop tubes which are expanded directly into the circular superheater headers, housed in a separate easily accessible casing immediately above the boiler.

A large stoker-fired combustion chamber is provided, the side and front walls being water-cooled by means of plain tubing, the lower ends of the tubes being expanded directly into square section water-boxes, whilst the upper ends are either expanded directly into the front steam and water separating drum, or to top headers having connecting pipes leading to this drum. The lower headers of the water-walls are fed by means of downcomers from the boiler lower rear water drum. Suspended arches of refractory material are provided at the front and rear walls of the combustion chambers.

A steam receiver, or dome, running the full width of the boiler, is provided immediately above the rear steam drum, and connected thereto by a series of nearly vertical tubes expanded into each drum plate, any accumulation of moisture being permitted to drain freely back to the water in the boiler. The steam passes from the steam receiver into the superheater inlet header via a steam and water separator or dryer, fitted to the highest point inside the receiver.

At the rear of the boiler is placed the well-proportioned economiser, consisting of four banks of straight horizontal steel tubing, provided with cast iron gills or fins, giving extended heating surface, equal to about six times the surface of bare tubing. The steel economiser tubes are connected together at their ends by means of U type connecting tubes, row by row, so that the feed water fed into the economiser inlet header at the coolest end, travels onwards row by row of tubes, until it reaches the outlet header at the hottest end, before being passed on to the feed check valve at the boiler rear top drum. By-pass dampers are arranged so that the hot gases may be caused to pass directly into the fan inlet ducting when required.

The mechanical 'travelling' grate stoker is of large capacity, and provided with a forced draught fan. The stoker is so arranged that regular and even combustion can be secured over the grate surface under all combustion conditions, thus resulting in maximum output and a high combustion efficiency. The fuel is stored in a bunker just outside the boiler house and is fed to the stoker coal chutes by means of a suitable conveyor.

After combustion is completed the ash and clinker from the various hoppers is conveyed outside the boiler house by means of a water-immersed travelling conveyor.

An induced draught fan and dust collector is mounted above the boiler on the fan platform which also carries the boiler chimney for discharging the spent gases to the atmosphere.

All the pressure parts of the boiler are suspended freely on steelwork independent of the boiler house structure, with freedom for expansion everywhere. The combustion chamber brickwork is supported off the firing floor steelwork integral with the boiler structure and is free to expand in all directions thus avoiding cracking or dislocation of the brickwork. The firing floor steelwork also carries the weight of the stoker and hoppers. The large basement below the firing floor contains the F.D. fan, ash conveyor, various hoppers and turbine condensing water circulating pipes. Suitable arrangements are also provided for handling the ash and climker when the ash conveyor is not working.

By simply removing the manhole covers from the boiler drum ends, access is obtained to the whole of the boiler water-tubes, which may thus be easily and conveniently examined and cleaned.

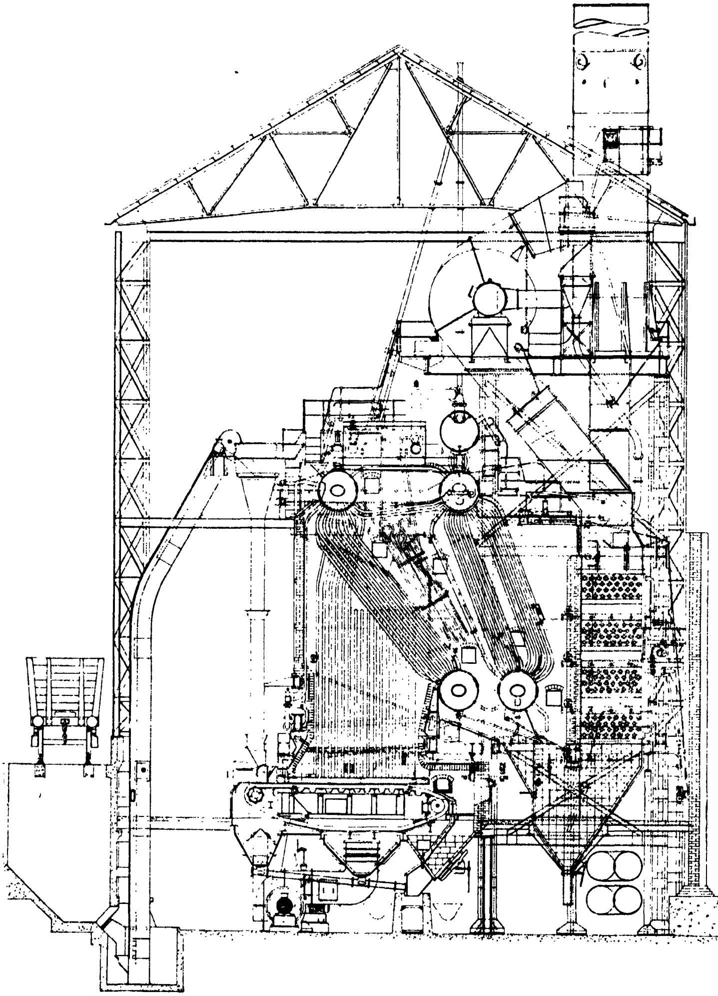


FIG. 1.—Clarke-Chapman Four-drum Water-tube Boiler.

Replacements of any individual tube when required may be easily and quickly effected. Inspection and cleaning of the whole of the gas side of the pressure parts is conveniently carried out by access to the spacious chambers surrounding the tube banks; suitable access doors being provided.

Daily shift cleaning of the whole of the heating surfaces is carried out by means of steam soot blowers, arranged at convenient points throughout the boiler, superheater and economiser tubing. Suitable soot and dust pockets of ample capacity are provided with convenient chutes for its removal.

The radiant-heat surface of the boiler is large and carries out a big proportion of the boiler evaporation. The convection heat surface is provided with suitable gas baffles, the area of the different passes being such that a gradually increased gas velocity is obtained ensuring maximum heat abstraction from the gases before entering the economiser.

Radiation of heat from the boiler is reduced to almost nil as the whole of the boiler-brickwork is covered with a steel casing suitably lagged by heat insulating material.

The multiple drums and water-tubing holds a large amount of contained water and heat reserve, and the extensive water level surface results in easy release of the steam and prevention of priming.

The boiler water has a well defined circulation; the feed water normally passes down the rear bank of tubes into the lower rear drum, and thence across to the lower front drum, rising up the front bank of tubes which comprises the main generating portion of the boiler. The steam generated passes from the front to the rear drum by way of the steam connecting tubes, whilst the water similarly passes to the rear drum by way of the water connectors.

The second bank of tubes are the downcomers, these with the front bank and connectors constituting the circuit. The gases and water flow in opposite directions, or counter flow, thus ensuring maximum circulating efficiency, and heat extraction.

It will be observed that the high hydraulic head available at the water walls is conducive to very effective steam and water circulation, the head being measured from the water level in the rear top drum to the inlet at the lower water boxes. By these means there is no possibility of the tubing in the hottest zone of the boiler being starved of water supply to make up the very rapid evaporation of the water walls. Similarly the front bank of boiler tubes is well supplied with water to keep pace with the evaporation from radiant heat surface.

The whole unit is conveniently operated and access is available to all parts of the boiler by carefully arranged ladders and platforms. A boiler control panel containing the necessary instruments for efficient operation is placed on the firing floor at the front of the boiler.

Clarke-Chapman Tri-drum Water-tube Boiler.

The Clarke, Chapman Tri-drum Water-tube Boiler is shown in fig. 2, the main features differing from the boiler shown in the previous figure consisting in the provision of an air pre-heater placed in the path of the gases immediately after the economiser, and supplying heated air for the combustion of the fuel; a secondary air fan supplying a strong blast of heated air above the stoker both from the rear and front of the combustion chamber; water-cooled walls to the four sides of the furnace the tubes being covered by Bailey cast iron blocks; primary and secondary controllable superheater; 3-tier square-finned tube economiser; coal bunker integral with the boiler house with coal weigher and travelling coal chutes to the stoker hoppers; bucket coal conveyors; separate house containing the F.D. and I.D. fans, the air to the F.D. fan being drawn immediately from above the top of the boiler and delivered by means of insulated hot air ducting to stoker; an annexe containing the steam and electric boiler feed pumps, a large basement, the ash and clinkers being removed by suitable trolleys; dual F.D. and I.D. fans and secondary fans; and external gas flue at roof leading to independent brick chimney adjacent to the boiler house.

The instrument panel containing the various pressure gauges, draught gauges, pyrometers, CO_2 meter, etc., being mounted at operating floor level and in front of the boiler.

In this case an even larger amount of radiant-heat surface is provided and the combustion chamber is of large capacity, capable of utilising a very large heat release per cubic foot. For soot blowing both gun type and multiple nozzle type steam blowers are provided and arranged to be operated in the most efficient sequence.

The water and gas circulation is similar to that of fig. 1, but additional baffles are provided at the rear bank to give additional travel to the feed water making for a very effective high temperature feed heater.

The whole plant has been arranged to give maximum thermal efficiency.

'Stefco-Wildish' Steam Separator and Dryer.

(Clarke, Chapman & Co., Ltd.)

The 'Stefco-Wildish' Patent Steam Separator is designed upon the basic principle of specific gravity, and arranged to replace the ordinary anti-priming pipe in a boiler. Solids, in the form of impurities and scale-forming matter, are heavier than steam, and in continually splitting up

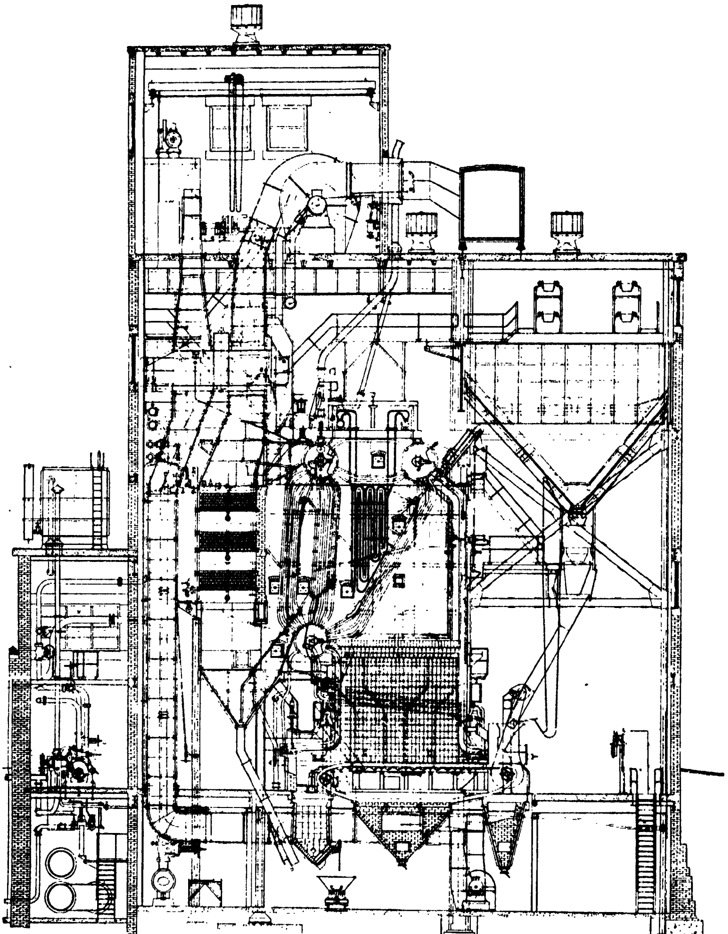


FIG. 2.—Clarke-Chapman Tri-drum Water-tube Boiler.

the volume of steam passing through the 'Stefco-Wildish' separator, as also by changing the direction of flow, the impurities, scale-forming matter and moisture content of the steam are precipitated in such a manner, and under such conditions that they may be easily discharged from the boiler.

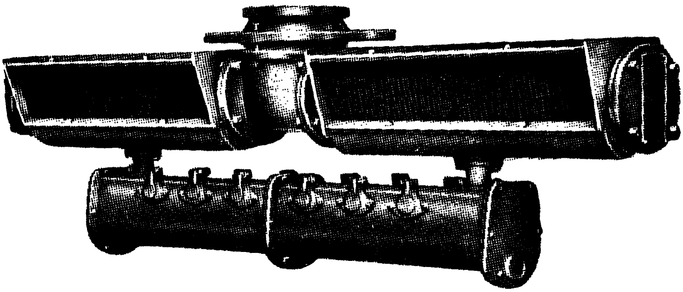


FIG. 3.—Stefco-Wildish Steam Separator as fitted in Boiler.

The design and outstanding features will be clear upon reference to figs. 3 and 4. The apparatus is so designed that by detaching the container (D) and the drain, it will pass through a standard oval manhole 15 ins. × 11 ins.

When in position the separator is attached to the boiler or drum shell immediately beneath the main steam outlet or stand-pipe. Fig. 4 shows the patented construction of the baffles, angled to the direction of flow of the steam, giving even better separation than vertical baffles due to the velocity of the steam assisting the separation. The impurities, solids and moisture are precipitated into container (D) fitted with flap valves which permit the return of the separated

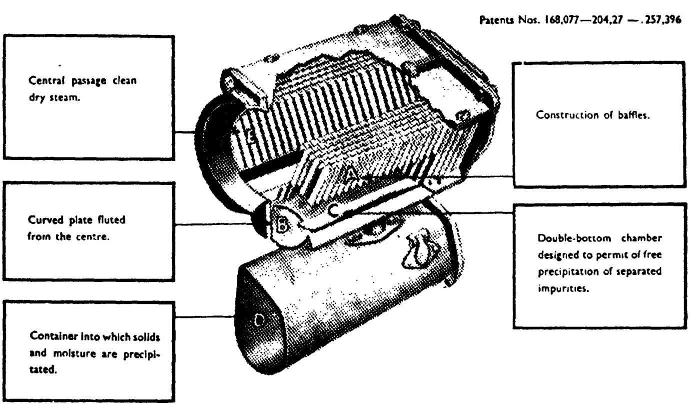


FIG. 4.—Showing Internal Construction.

water to the boiler at boiler temperature, while the heavier recovered sediment collects in the bottom of the vessel and by means of a connection through the drum shell may be blown out while the boiler is in operation.

The moisture and scale-forming matter that has been separated and collected is prevented, under surging and priming conditions in the boiler, from being drawn back and entrained along with the steam.

Other separators embodying the 'Stefco-Wildish' patents and suitable for fitting in pipe lines, are also made and are suitable for dealing with steam or air.

The Cochran Boiler.

(Cochran & Co., Annan, Ltd., Annan, Scotland.)

The boilers (fig. 1, p. 1207) are used very largely both ashore and afloat; the advantages claimed for them over other kinds for land use and export consist, principally, in great economy of space, in not requiring any brickwork setting, in having a large amount of highly effective heating surface, and in being very economical in fuel, besides being accessible for cleaning and repairs outside and inside.

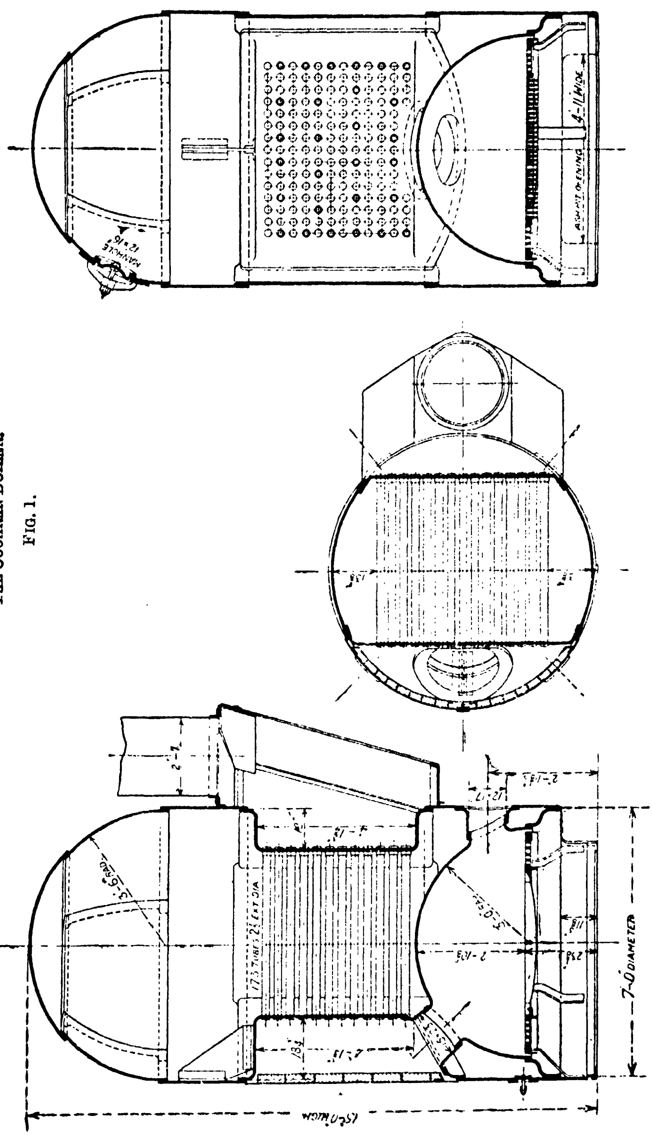
For winch boilers this type contains the advantages of the cross-tube boiler and the horizontal boiler without their disadvantages, and has been found the most suitable for large passenger, mail, and cargo steamers.

The boilers are made in a range of standard sizes, as per the following Table, which gives particulars of their standard dimensions and normal evaporations.

Evaporation of Cochran Boilers.

No.	Particulars of Boiler.							Normal Steaming.					No.
								Coal.		Evaporation.			
	Diam. of Boiler.		Height of Boiler.		Grate Area	Heating Surface.	Ratio H.S. G.A.	Lbs. per hour.	Lbs. per sq. ft. of Grate per hour.	Per lb. Coal.	Per sq. ft. H.S. per hour	Total per hour in round figures	
	1		2		3	4	5	6	7	8	9	10	
	Ft.	In.	Ft.	In.	Sq. Ft.	Sq. Ft.							
1	3	0	6	9	4·75	50	10·5	42	9·1	6·1	5·1	250	1
2	3	3	7	6	5·75	60	10·4	56	10·0	6·3	5·3	340	2
3	3	9	8	6	7·50	100	13·3	86	11·5	6·2	5·4	540	3
4	4	0	9	0	8·50	110	12·9	102	11·8	6·2	5·4	630	4
5	4	3	9	6	9·25	140	15·1	112	12·0	6·6	5·3	750	5
6	4	6	10	0	9·75	160	16·4	121	12·4	6·8	5·2	820	6
7	4	9	10	3	11·75	190	16·2	150	12·6	6·9	5·4	1,030	7
8	5	0	11	3	12·50	220	17·6	163	13·0	7·0	5·2	1,150	8
9	5	3	11	9	14·00	250	17·8	187	13·3	7·0	5·2	1,320	9
10	5	6	12	3	16·75	300	17·9	229	13·6	7·0	5·4	1,620	10
11	5	9	13	0	18·75	350	18·7	261	13·9	7·1	5·3	1,880	11
12	6	0	12	6	18·75	350	18·7	261	13·9	7·1	5·3	1,880	12
13	6	0	13	6	18·75	350	18·7	261	13·9	7·1	5·3	1,880	13
14	6	0	14	0	18·75	400	21·4	261	13·9	7·6	5·0	2,000	14
15	6	6	13	6	22·50	450	20·0	319	14·1	7·3	5·2	2,360	15
16	6	6	14	0	22·50	450	20·0	319	14·1	7·3	5·2	2,360	16
17	6	6	14	6	22·50	500	22·2	319	14·1	7·7	4·9	2,480	17
18	7	0	14	0	26·75	500	18·7	385	14·3	7·1	5·4	2,750	18
19	7	0	15	0	26·75	600	22·4	385	14·3	7·8	5·0	3,020	19
20	7	6	16	3	31·50	750	23·1	460	14·6	7·9	5·0	3,650	20
21	8	0	16	6	37·00	850	23·0	543	14·6	7·9	5·0	4,320	21
22	8	6	18	0	41·00	1,000	24·4	602	14·6	8·2	4·9	4,960	22
23	9	0	19	0	48·00	1,250	26·0	740	15·4	8·1	4·8	6,030	23

Boilers are numbered consecutively 1 (3 ft. 0 in.) to 23 (9 ft. 0 in.).



THE COCHRAN BOILER.

FIG. 1.

In addition to the range of standard sizes detailed in the above table, the 'Cochranette' Boiler, 2 ft. 6 in. dia. × 5 ft. 6 in. high × 25 sq. ft. heating surface, is built to stock in the one size only, for 100 lbs. per sq. in. working pressure. Also Cochran boilers are now constructed for the utilisation of Diesel engine exhaust gases only, and composite types for simultaneous use of exhaust gas and oil firing.

Cochran Diesel Exhaust-Gas Fired Boilers.

From the illustration (fig. 2) it will be seen that the principal advantages of the Standard Cochran design are incorporated in the exhaust gas boiler, including the feature of accessibility to all internal parts for cleaning and examination, which is essential for the maintenance of maximum heat transfer efficiency.

It will be readily appreciated that owing to the variety in sizes and types of Diesel engines available, it is not practicable to publish exhaustive tables of standard waste heat boilers, and that each individual case must be studied on its merits.

When considering a boiler for the utilisation of exhaust gases from Diesel engines, three methods suggest themselves :—

1. To use a Cochran Composite Boiler which has been designed to extract the maximum heat from the exhaust gases and can be fired simultaneously by oil fuel. For marine purposes, when the exhaust gas is used to raise steam for working a ship's auxiliary such as a dynamo or steering gear, this arrangement is essential, owing to the fact that it is sometimes necessary for the engine to be slowed down in fog, etc., with a consequent falling-off in temperature of the exhaust gases and steam raised.

2. To use a boiler designed to deal with exhaust gases only. The size of boiler to be used with a particular engine depends on the type of engine (2- or 4-stroke) and the quantity of exhaust gases. As a general rule the temperature of the exhaust gases from a 4-stroke engine will be approximately 750° F. and from a 2-stroke 550° F., and the weight of exhaust gases 14 lbs. per b.h.p./hr. for a 4-stroke engine, and 20 lbs. b.h.p./hr. for a 2-stroke engine.

3. To pass the exhaust gases through a standard boiler having tubes of diameter smaller than standard. With this arrangement the boiler is limited in use, in that it is only possible to raise steam alternatively, either by exhaust gas or oil firing. Steam raised in the standard boiler by exhaust gases will be less than with a boiler especially designed for the purpose. A special gas-tight construction is necessary for the furnace door and casings.

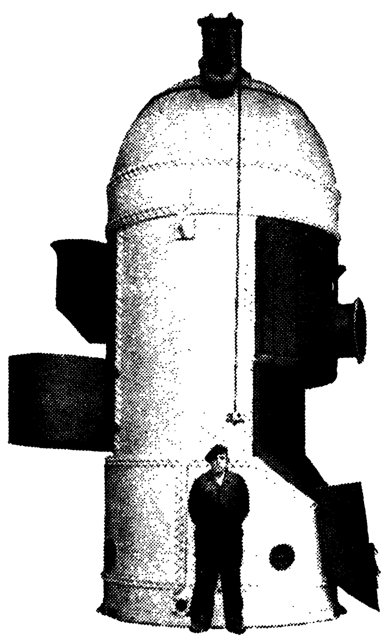
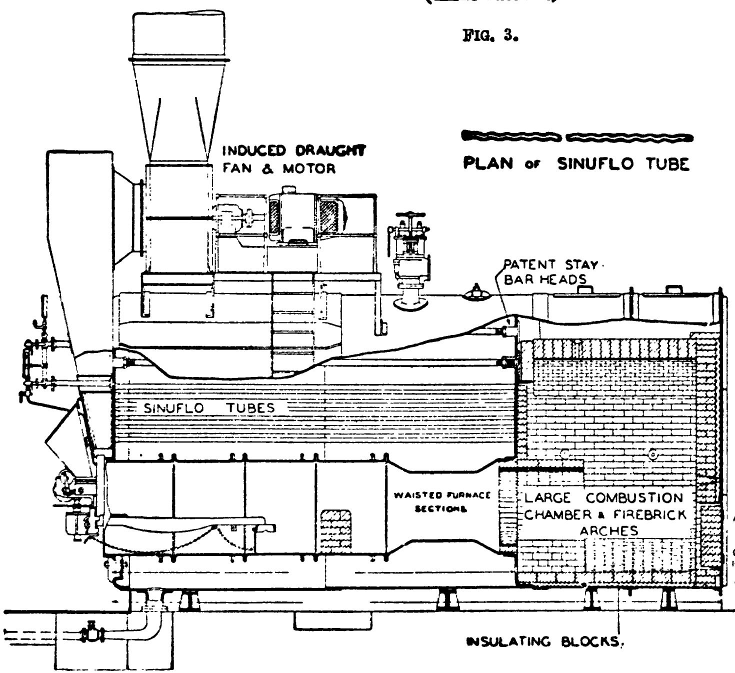


Fig. 2.—A Cochran Composite Type Boiler.

THE COCHRAN 'SINUFLO' ECONOMIC BOILER.
(Kirke Patents.)

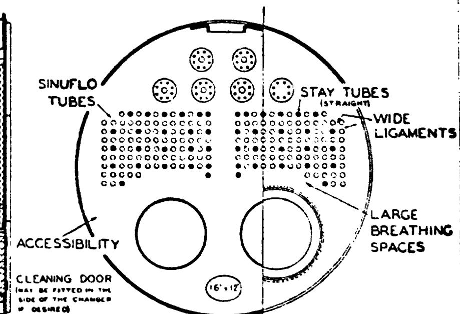
FIG. 3.



PLAN OF SINUFLO TUBE

PRINCIPAL FEATURES OF DESIGN

SINUFLO TUBES .	*To augment heat extraction.*
INDUCED DRAUGHT	*To increase boiler evaporation and to allow of sensitive control of all loads.*
ACCESSIBILITY .	*To permit inspection and cleaning of all internal surfaces.*
WAISTED FURNACE SECTIONS	*To give ready accessibility and promote early combustion.*
BREATHING SPACES	*To prevent leakage due to unequal expansion of tubes and furnaces.*
WIDE LIGAMENTS .	*To obtain effective water cooling of the back Tube Plate and to facilitate circulation.*
LARGE COMBUSTION CHAMBER AND FIRE-BRICK ARCHES	*To effect complete and smokeless combustion before the gases reach the tubes to extract and collect grit and dust.*



SECTION THROUGH
WAISTED FURNACE

VIEW ON BACK
END PLATE

THE COCHRAN 'SINUFLO' INDUCED DRAUGHT ECONOMIC BOILER.

Boiler Diameter.	Overall Length.	No. of Flues.	Diameter of Flues.	Maximum Working Pressure in lbs./sq. in.	Normal Evaporation in lbs./hour from feed at 60° F.	Normal Evaporation in lbs./hour from and at 212° F.
ft. in.	ft. in.		ft. in.			
6 6	18 7	1	2 9	250	4,800	5,800
7 0	21 7	1	3 0	250	5,700	6,800
7 6	21 7	1	3 6	250	6,700	8,000
8 0	21 7	1	4 0	225	7,600	9,100
9 0	22 7	1	4 6	200	8,500	10,200
9 0	22 7	2	2 9	250	10,400	12,500
9 6	22 7	2	3 0	250	11,300	13,600
10 6	22 7	2	3 6	250	13,300	16,000
11 6	22 7	2	4 0	225	15,200	18,200
11 9	22 7	3	3 0	250	17,100	20,500
12 6	22 7	3	3 6	250	20,000	24,000
13 0	22 7	4	3 0	250	22,900	27,500
13 6	22 7	4	3 3	250	25,000	30,000

The above ratings are based on the assumption that automatic stokers with self-cleaning grates are fitted and fired with suitable coal.

The high rate of extraction of heat from gases passing through the patent 'Sinuflo' tube has made possible the design of a high duty boiler of the simple dry-back, return-tube type in which a high efficiency can be obtained without recourse to such aids to efficiency as double passes of tubes, external ducts, air preheaters and economisers.

Owing to the high tube efficiency of the 'Sinuflo' design of tube, it is claimed to be possible to work the boiler with induced draught and very heavy loads without the temperature of the gases becoming excessive. Moreover, it has been found that the frictional resistance to the flow of gases through a 'Sinuflo' tube is less than through an equivalent straight tube, *i.e.* a straight tube that would give the same temperature drop in the gases between entry and exit. Consequently, no increase in fan power is required to give the rated outputs.

The following distinctive features are incorporated (see fig. 3, p. 1209):

(1) The flues are made up, each with a waisted section, giving on the water side access to the whole of the flues and the shell of the boiler and causing on the fire side turbulence of the gases and rapid combustion.

(2) A wide space is provided between the tube nests and the flues, which gives the necessary flexibility for breathing which is regarded by builders of Scotch marine boilers as so essential if leakage troubles due to the unequal expansion and contraction of the flues relatively to the tubes are to be obviated.

(3) The tubes are fixed at wider intervals than is customary in return-tube boilers, thus allowing ample space between them for circulation of the water, and ensuring adequate cooling of the back tube plate.

(4) To eliminate the possibility of leakage down the threads of the longitudinal bar stays, solid flanged forgings are used for attachment to the tube plates, each forging having a blind hole screwed internally to take the bar stay and having its flange riveted to the end plate.

(5) The difference of expansion between the fire tubes and stay tubes due to different thicknesses of metal reduces internal strains, as the 'Sinuflo' tubes are less rigid longitudinally than ordinary tubes.

The Combustion Chamber.

The combustion chamber casing is of mild steel plate and forms an air-tight extension to the boiler shell. It is of much larger dimension than is usual for this type of boiler, and lined with firebrick over a layer of insulating brick. Over the top of the flue ends are built two brick arches extending well into the combustion chamber which give a maximum length of travel to the flame. This ensures that combustion is completed before the gases enter the tubes, and thereby avoids the flames licking into and overheating the ends of the tubes.

These arches have the additional effect of causing an abrupt change in direction in the flow of the gases which results in a large proportion of the light ash carried over from the fire being deposited at the back of the combustion chamber, whence it can be removed at intervals.

Induced Draught.

(1) Fuel can be burnt at a much higher rate per square foot of grate than is possible under natural draught conditions, enabling larger outputs to be obtained.

(2) Combustion conditions in the flues and combustion chamber can be rapidly adjusted to ensure proper combustion and to suit varying loads.

(3) The cost of a tall chimney and its foundations is saved.

COCHRAN 'SINUFLO' WASTE HEAT BOILERS.

(Kirke Patents.)

Fig. 4 shows a boiler of the horizontal type. The design, consequent upon the adoption of the 'Sinuflo' tube, is based on consideration of the fact that gases passing through straight tubes are not directed towards the tube walls at all. Molecules flowing in a parallel direction to the tube walls pass their heat to one another until the tube wall is reached, most of the molecules themselves never reaching the wall at all. After two years' intensive experiments, numbering several hundred, it was discovered that the heat transmission could be accelerated in fire tubes

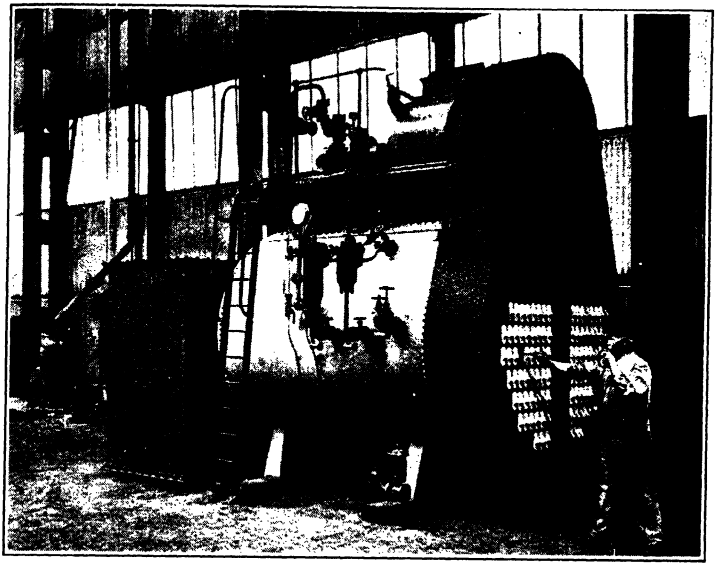


FIG. 4.

150 per cent. per unit of heating surface by so shaping the tubes without reducing their cross-sectional area that the gases were repeatedly directed towards the tube walls. Thus it became possible to reduce the tube length 60 per cent.

The frictional resistance to the flow of gases through a 'Sinuflo' tube is no more than through an equivalent straight tube.

Advantage has been taken of this discovery to increase the size of the tubes to 2 in. O.D. as compared with 1¾ in. O.D. previously popularised, thereby reducing the number of tubes to be

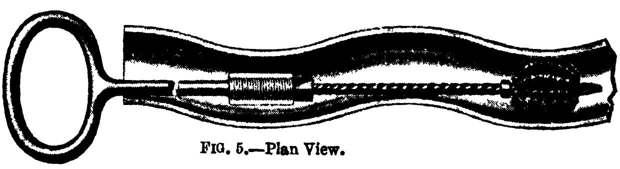


FIG. 5.—Plan View.

FIG. 6.

brushed by 26 per cent. for the same flue area. Again, the length of the tubes has been also halved and the labour of brushing more than halved.

The tubes are made very slightly sinuous, a large clear axial space (fig. 6) being left through the tube to accommodate a straight steel rod to which a spherical brush may be fixed.

Flexibility is not required between the brush handle and the brush, because the spherical brush can freely follow the waved axis of the tube.

The clear axial space between adjacent tubes is increased by 43 per cent., which renders cleaning on the water side much more easy than before.

Owing to the increased diameter of the tubes and their greatly reduced length, there is no tendency for them to sag.

The length of the clearance space for removal and replacing tubes is, of course, only half that required for the older type of boiler.

The waves in the tubes are arranged in a horizontal plane, so that the bottom of the tubes appears straight when viewed from the side, as shown in fig. 7. This prevents moisture from settling in the tubes.

The boiler in fig. 4 is suitable for working in conjunction with vertical carbonising plants, horizontal carbonising plants, water gas plants, soaking pits, and reheating furnaces and also for generating steam from the combustion of blast furnace gas in the inlet products chamber. The boilers are generally of diameter 6 ft. to 8 ft. and of length 9 ft. 6 in., the boiler shell being made in one strake. The tubes fitted are of 2-in. outside diameter and the evaporations range from about 4,000 lb. per hour to 8,000 lb. per hour. The steam raised is usually about 3 to 4 lb. per lb. of coke to producers and occasionally as much as 5 lb. when the temperature of the waste gases is high.

FIG. 7.

While retaining every one of the good points of other induced draught fire tube waste heat boilers, the following advantages are claimed for the Cochran-Kirke Patent 'Sinuflo' boiler :—

1. It occupies little more than half the floor space.

2. Only about half the space is required at the back of the boiler for withdrawing or brushing tubes.

3. The boiler may be made either vertical or horizontal.

4. The shell can be made in one ring instead of three.

5. The weight to be supported including water is reduced by about half.

6. The time required to brush the tubes is halved.

7. The heating surface is much reduced and the labour of cleaning it on the water side correspondingly saved.

8. The use of 2-in. tubes reduces the number to be brushed by 26 per cent. as compared with 1¾-in. tubes for the same total flue area.

9. The minimum space between the tubes is 1¼ in. or 43 per cent. more than with the usual 1¾ in. tubes, which renders cleaning on the water side much easier.

10. The difference of expansion on starting up between the tubes and the shell is halved which renders any form of leakage due to this cause almost impossible.

11. The tubes do not sag appreciably in the middle when the boiler is empty.

12. Owing to the length of the boiler being approximately halved, for the same evaporation, the rate of circulation in the boiler is correspondingly increased. This greatly facilitates the separation of steam from the water, ensuring very dry steam at high overloads as has been so definitely proved in water tube boiler practice.

13. The resistance through the boiler is considerably less for the same flue area owing to the substantially reduced frictional surface over which the gases pass.

For open hearth steel furnaces larger boilers are used, these ranging from 9 ft. to 13ft. in diameter with a length of 15 ft. 6 in. and with tubes of 2½-in. or 2¼-in. outside diameter. The

evaporation is from 10,000 to 18,000 lb. per hour. An average figure per ton of steel melted is 1,400 lb. Fig. 8 shows one of these boilers of 13 ft. diameter.

The initial cost of installing such boilers is generally recovered within the first year or so. The running costs are practically limited to the electric current consumed by the induced draught fan. A typical figure for fan power required is 43 kilowatts for a boiler of 12 ft. 3 in. diameter working in conjunction with a melting furnace of 90 tons capacity.

The fan power varies almost directly as the cube of the velocity of the gases through the boiler tubes, and for this reason high velocities are to be deprecated. For example, if a boiler of 11 ft. 6 in. diameter were substituted for the 12 ft. 3 in. boiler mentioned above, the fan power would be increased to 54 kilowatts and the cost of running the boiler would rise by about 25 per cent.

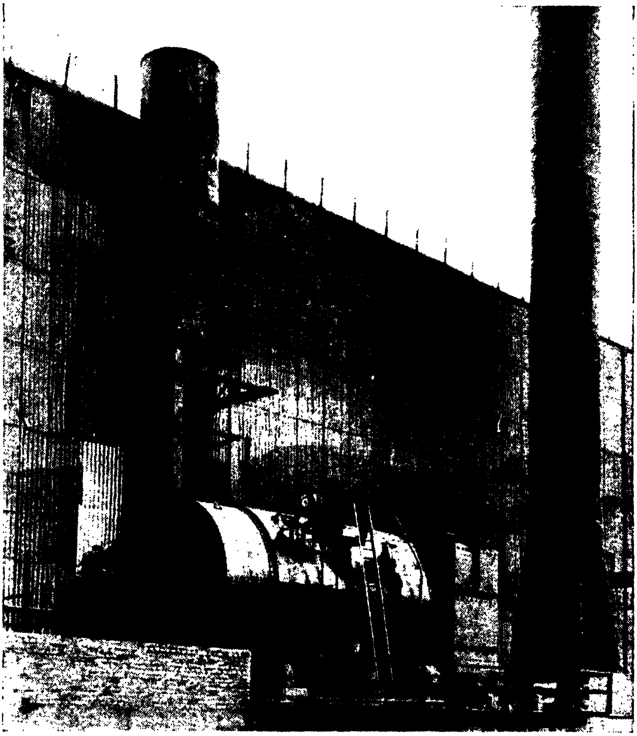


FIG. 8.—A Cochran Sinuflo Waste Heat Boiler at a Steelworks.

VERTICAL WASTE HEAT BOILERS.

For high temperature gases and where the duty is comparatively light the Cochran Vertical Boiler is adapted for use as a waste heat boiler. To obtain high tube efficiency, Sinuflo tubes are fitted, these being arranged in two nests through which the gases pass in series.

The arrangement effects a considerable saving in floor space as the boiler furnace forms the inlet products chamber and the area occupied by the boiler itself is small. So also is the space required for the brushing and replacement of tubes.

Evaporations of the order 500 lb. per hour to 5,000 lb. per hour are obtained and where fluctuations in heat input, or in steam demand, are liable to occur a steam storage effect is obtained by building the boiler with an extended top belt. The induced draught fan can be mounted on the boiler crown or at ground level.

COCHRAN 'SINUFLO' GAS-FIRED BOILERS.

(Kirke Patents.)

Fig. 9 shows a Cochran 'Sinuflo' vertical gas-fired boiler for working with natural draught, suitable for burning town's gas, water gas, clean producer gas, or sewage gas. The thermal efficiency is 85 per cent. on the net calorific value of the gas, and the following advantages are claimed :—

1. No fuel storage.
2. Minimum floor space.
3. No primary air so that back-firing is impossible.
4. Heat generated and extracted entirely within the tubes.
5. No fire-box in which explosive mixtures can collect.
6. Independent of chimney draught.
7. Complete silence due to the patent burner fitted.

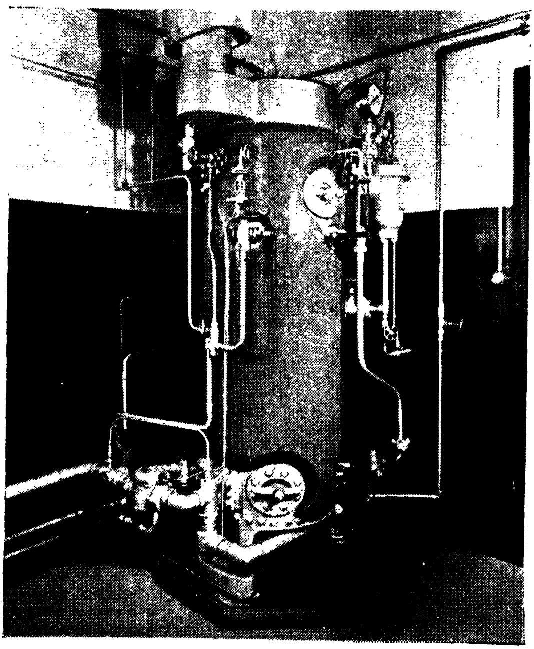


FIG. 9.

COCHRAN 'SINUFLO' EXHAUST-GAS BOILERS (KIRKE PATENTS) FOR INTERNAL COMBUSTION ENGINES.

Until recently it was customary to build Exhaust Gas Boilers of the horizontal type with straight tubes arranged either in one nest, so that the gases passed straight through the boiler, or in two nests so that the gases passed through and back again. The efficiency of the 'Sinuflo' tube has enabled these boilers with a single pass and of considerably reduced length, to be designed for the same evaporative capacities hitherto provided by the double pass of greater length. They are made in a range of sizes suitable for engines of 70 B.H.P. up to 10,000 B.H.P., and owing to the greatly reduced heating surface required, the resistance to flow of the gases through the tubes is less than that through an ordinary boiler for the same output. They are built of either the vertical or horizontal type and are economical in capital cost.

THERMAL STORAGE.

(Ruths Accumulators (Cochran) Ltd.)

One of the problems with which the engineer in charge of a boiler house is often confronted is how to cope with fluctuations in the demand for steam. The fluctuations may be irregular and violent but of short duration; regular extensive and of long duration; or of some intermediate type. The usual means available to the boiler attendant for meeting them are:

(*a*) To allow the boiler pressure to fall and to build it up again by degrees until the safety valves lift.

(*b*) To shut off the feed so that steam is raised from water already present in the boilers at saturation temperature.

(*c*) To increase the rate of firing or otherwise by attention to the fire.

(*d*) To prevail on the steam users to stagger their peak demands.

It is obvious that the efficacy of method (*a*) is largely dependent on the water content of the boilers, and that of method (*b*) on the extent to which the water level can be varied without prejudice to the safe working of the boiler.

Boilers of the same evaporative capacity but of different types may vary very considerably in their water content and in their ability to allow of water level variations, but obviously boilers with large water contents take longer to raise steam and the boilers in which the water level can be varied extensively are lacking in compactness. Method (*c*) results in inefficient working, high fuel consumption and loss of steam through the safety valves. The real solution to the problem is to provide thermal storage in a separate vessel.

Variable Pressure Accumulators—The Ruths System.

In its usual form the Variable Pressure System comprises a battery of high pressure boilers, a steam accumulator, a group of high pressure steam consumers, a group of low pressure steam consumers and a system of automatically controlled steam valves. The high pressure consumers may be power generating plant, pumps, compressors, process units requiring high temperature, etc., and the low pressure consumers dye kettles, soap boiling pans, evaporators, milk pasteurising and sterilising, etc. The system is illustrated in fig. 10. The accumulator is simply a storage

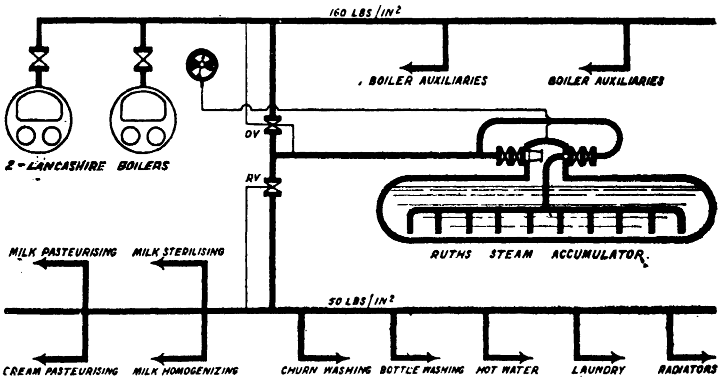


FIG. 10.—Diagram showing Interconnections and Valves between Boilers, Accumulator and Process Plant.

vessel (or several such vessels) placed *either* horizontally or vertically and generally built for a working pressure the same as that of the high pressure boiler. It is filled with water to about 90 per cent. of its capacity.

The principle is that the boilers are worked at a steady load corresponding to the total average demand from both high pressure and low pressure consumer groups, and whereas this average may vary from hour to hour or from day to day necessitating a higher or lower rate of steaming the boiler attendant has, because of the steam accumulator, ample opportunity to alter his rate of firing by a steady increase or reduction. He can, in fact, always work his boilers to give their highest thermal efficiency.

The automatic control system is such that high pressure steam is supplied direct to the high pressure consumers and the surplus steam is directed to the accumulator. Here its effect is to raise the temperature of the water and with the temperature the pressure. The heat contained in the surplus steam is thus stored in the water.

The low pressure consumer receives its steam through a reducing valve which ensures constant pressure in the low pressure main. Thus while constant pressures, with all their advantages, are maintained in the high pressure and low pressure mains, the pressure in the accumulator itself varies in accordance with the amount of heat or steam with which it is charged.

The Ruths Accumulator was developed in 1916 by Dr. Johannes Ruths, the celebrated Swedish technician, and has been used extensively throughout Europe, America and South Africa.

Constant Pressure System.

The constant pressure or feed water accumulator is represented by the Kiesselbach, the Marguerre and the Ruths Systems. The aim is to provide feed water to the boilers at or near saturation temperature, the water being heated by hot water or live steam taken from the boilers at such times as the demand for steam by consumers is less than the boiler output. An obvious effect of the use of feed water at high temperature is to raise the evaporative capacity of the boilers enabling them, up to a point, to meet peak loads without loss of pressure.

Applications of Accumulators.

Steam accumulators are largely used in conjunction with waste heat boilers in which the heat input, and therefore the amount of steam raised, fluctuates in an uncontrollable manner.

They are also used in many industries in which the low pressure consumers are dependent for efficient operation on a constant supply of steam at constant pressure, *e.g.* laundries, textile works, iron and steel works, mines and collieries, gas-works, sugar factories, breweries, soap and candle works, chemical and allied trades, rubber and cable works, hotels, food factories and dairies, paper mills.

The following results showing the improvements effected in a laundry by the installation of a Ruths accumulator are typical:—

Output greatly improved, increase up to 12 per cent. in certain departments.

Output of one 160 in. callender raised from a maximum of 800 to a maximum of 1,000 sheets per hour.

Number of laundry operating hours reduced from 52 to 48 per week.

Process water temperature stabilised at 180° F. as compared with a variation from 140° to 165° F.

Average evaporative capacity of steam generating plant increased by 26 per cent.

Evaporation of boilers increased from 9·02 to 9·91 lb. of water per lb. of coal consumed.

Ruths Accumulators have not been used extensively in Power Stations in this country, but some forty installations were in operation on the Continent in 1939. They included:—

Power Station.	Total Volume Cu. Ft.	Pressure Range. Lb./sq. in. Abs.	Capacity Kw. h.	Turbine Capacity. Kw.
Charlottenburg	176,000	206–22	73,000	50,000
Brussels	71,000	206–37	22,000	12,000
Copenhagen	58,000	169–21	24,300	10,000
Amsterdam	53,000	206–12	23,500	20,000
Hattingen	44,000	220–22	19,950	12,000
Gothenburg	38,800	206–37	16,000	10,000
Stockholm	7,760	350–147	2,400	20,000
Brünn	12,160	147	Feed Water Accumulator	

THE 'W' CENTRAL HEATING BOILER.

(Charles McNeil, Ltd., Scotland Street, Glasgow.)

Continuous automatically controlled stoking over periods of eighteen to thirty-six hours is obtained from this central heating boiler.

The principle employed is gravity feed from hoppers attached to the sides of the boiler. The fuel falls from these hoppers on to the sloping grate at a rate dependent on the amount of fuel which is being consumed.

By the employment of a suitable thermostatic control on the air intake the temperature of the water in the central heating system, or of the atmosphere in the premises in which the boiler is employed, can be maintained at the predetermined level without attention beyond the periodic filling of the hoppers and the removal of the ash.

Coke and anthracite are the fuels commonly used.

The boiler is of welded steel construction and of a design which has proved to give a high degree of efficiency. There is a long flue travel which, in conjunction with the interposed water tube elements, ensures a low temperature in the flue gases. The heating surfaces are arranged so that cleaning can be effected with the minimum of inconvenience.

The effective grate areas are large with the consequent advantages in economy in fuel consumption and in freedom from clinkering.

Independent technical authorities have carried out tests on these boilers in which the efficiency was 80 per cent. and it was proved that the rated duties are well within the capacities of the boilers.

A range of twenty-six standard sizes of these boilers, from 325,000 to 2,000,000 B.Th.U. per hour rating, is produced for installation in factories, offices, hotels, institutions, stores, etc. For smaller buildings boilers working on the same principle but of a slightly modified design are produced in a range of eleven standard sizes from 50,000 to 300,000 B.Th.U. per hour rating.

All these boilers can be used in the high temperature water and the low pressure steam system of heating as well as in the more usual hot water system.

THE YARROW LAND BOILER.

(Yarrow & Co., Ltd., Scotstoun, Glasgow, W.4.)

This boiler, fig. 1 (p. 1218), consists of three or more drums, including a saturated steam drum of large capacity, which is connected to each of the water-drums by straight tubes. The drums are of circular section, built up of pressed steel parts, the shell generally being in one ring, so that no intermediate circumferential joint is required. The drum ends are of partial spherical form, and no stays are necessary. For higher pressures, as in the case of Yarrow boilers adopted in many recent installations, the drums are made from hollow forgings, either entirely seamless, or with riveted joints only where separate ends are attached. The tubes in both the superheater and the boiler are expanded into thickened tube plates and bell-mouthed at their ends. The tube plates on the inside of the drums are, where necessary, recessed out in way of the tube holes, to enable the expanding of the tubes to be perfectly carried out. The Yarrow boiler is supported by the steam drum, which leaves the tubes and water-drums perfectly free to expand independently.

As one of the water-drums is placed at the front of the furnace and another at the back of the furnace, the tubes joining these drums to the steam-drum are adjacent to the furnace. It is well known that, owing to direct radiation, by far the greater part of the heat utilised in a boiler is transferred in the tubes nearest the fire. From the above it will be realised that as so large a proportion of the heating surface is adjacent to the fire, the total heating surface of the Yarrow boiler for the same output and efficiency can be considerably less than is usual.

Fig. 1 indicates a Yarrow type of S.F. 3 boiler, in which the gases traverse the outer nest of tubes three times, the area of the gas passage being designed for the required velocity of gases and to provide for a good transfer of heat by convection. This type is adopted wherever possible when high efficiency is required, as it takes full advantage of the transfer of heat both by radiation and convection.

Yarrow boilers are now made in various types, the principal eight of which, with their designations, are shown in fig. 2 (p. 1219).

Another characteristic of the Yarrow boiler is the very rapid circulation through the tubes owing to their being straight and nearly vertical.

An outstanding feature of this boiler is the complete absence of any form of horizontal baffle or obstruction in the path of the gases where dirt could collect. This feature can be readily seen from fig. 1, which shows the cross section of a Yarrow boiler with superheater as arranged when the fine gases are discharged overhead at the back of the boiler.

The Yarrow type of superheater consists of one or more mild steel drums into which inverted U-tubes are expanded. Suitable internal divisions are fitted in the drum so that the required number of passes through the U-tubes can be obtained. This form of superheater is self-draining.

In order to clean the interior of the boiler it is only necessary to remove the manhole door from each drum end (each is secured by two bolts). A man has ample room in the saturated steam drum for operating the tools for cleaning the inside of the tubes. For cleaning the outside of the tubes where this cannot be conveniently done from the furnace, access doors are provided in the side and end casings.

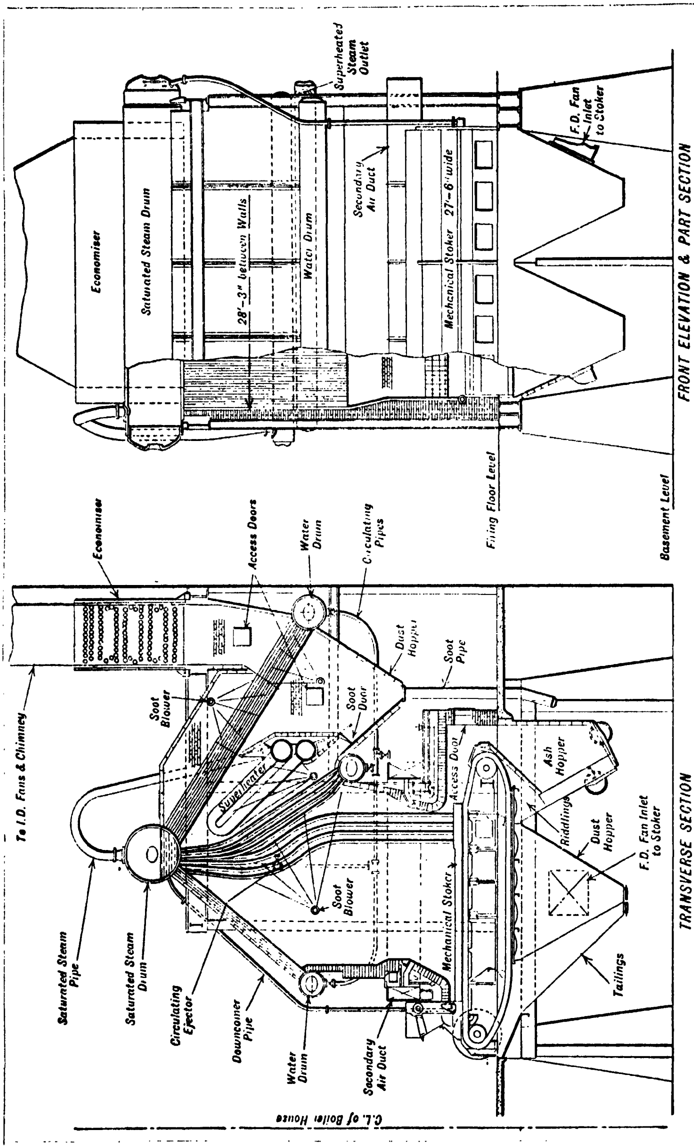


FIG. 1.—The Yarrow Land Boiler.

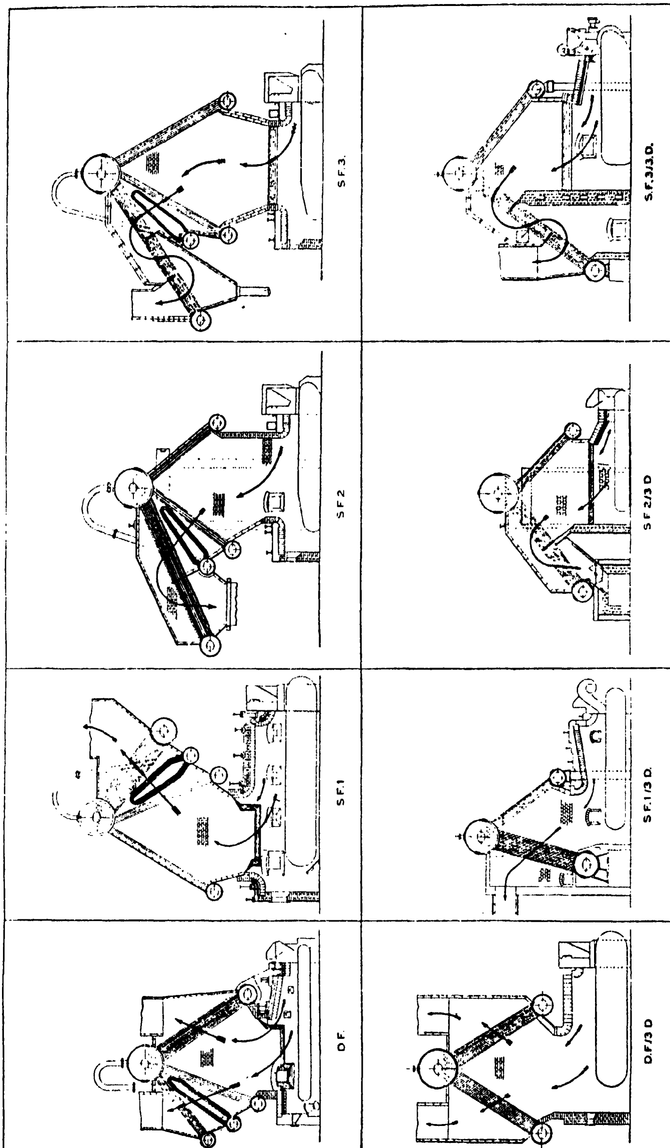


FIG. 2.—Various types of Yarrow Boilers. (Arrows indicate Gas Path.)

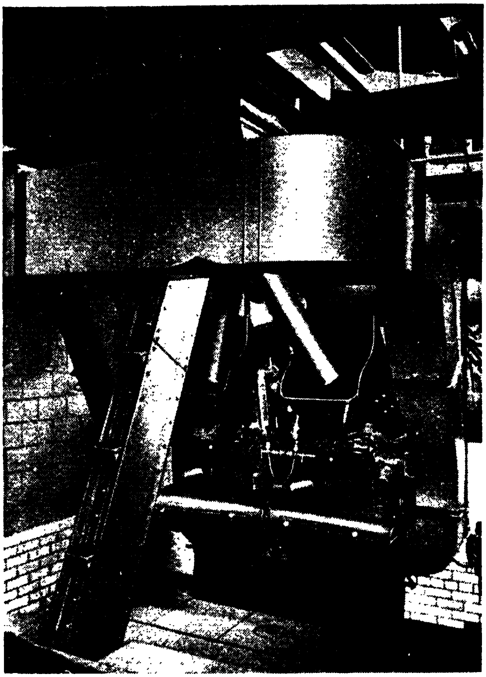
Owing to the great height of the furnace and to the large area between the tubes in the furnace rows, or, in other words, the low velocity of gases passing out of the furnace, the Yarrow boiler is found to remain comparatively clean after long periods of steaming. This, coupled with the fact that there are no places on which dirt can lodge, causes the efficiency of the boiler to be maintained.

For electric power stations where peak loads have to be provided for, the Yarrow boiler is particularly suitable, as it cannot be damaged by too quickly increasing the rate of evaporation, and it can be heavily forced without fear of damage. It is often fitted with feed-water economisers or airheaters, or with both.

Stokers.

'Bennis' Air-Draught Stoker and Self-Cleaning Furnace.

(Bennis Combustion Ltd., Little Hulton, Bolton.)



'Bennis' Air-Draught Stoker working in conjunction with 'Bennis' Automatic Elevator.

Front Plate.—The fuel feed and furnace actuating mechanisms, together with their associated driving units, are assembled on and secured to a baseplate designed to resist the warping effects of heat transmitted from the furnaces. The plate is secured to the boiler front, near to and enveloping the ends of the flues, and is extended on one side or the other, as conditions necessitate, to accommodate the gearbox units which are driven by either lineshaft-belt, motor-vee belt, or direct motor drive through flexible coupling.

The furnace driving or camshaft is connected to its gear unit by a semi-flexible large diameter coupling of substantial design, which excludes springs and rubber bushes. This shaft is fitted with a series of chilled cast-iron cams, upon whose rotation the reciprocating furnace bar motion depends. The camshaft bearings are of ample proportions, and are made integral with the side pedestals which support the front end of the furnace, together with the front deadplates and apron sheets. Altogether, these assemble into a rigid box-like formation.

Trough Bars.—The grate itself is composed of a number of interlocking grid bars manufactured from a special heat-resisting alloy. They are set in a series of air troughs, the air spaces being arranged and graduated in a manner conducive to perfect combustion. Since each trough

provides the air for a longitudinal strip of the grate, a thin or bare patch of fuel in one portion of the grate cannot affect the air distribution to the fuel over any of the other troughs.

The self-cleaning action is effected by moving all the furnace sections forward simultaneously through a short distance, then withdrawing alternate sections at a time, resulting in the bed of fuel being carried bodily away from the front end, while during the outward movement the fuel is held stationary by mass contact with those sections not in motion.

Combustion air pressure is developed by a forced draught fan producing a high velocity air flow. The profile of the duct is so shaped as to divide the air stream and pass it in correct predetermined proportions to the fuel bed. Riddlings falling into the troughed air ducts are propelled to the rear of the section concerned, and are expelled through traps into the ash chamber by a combined air pressure and mechanical means.

Grid Bars.—These are arranged to interlock securely and easily with each other into their respective carrying troughs, starting at the rear of the furnace and terminating at the more accessible front end where the front deadplate provides the final 'lock' to the system.

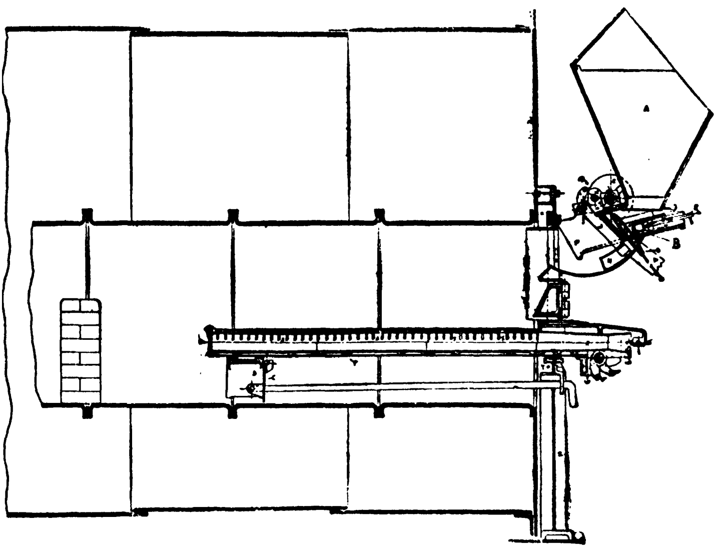
Ample space is available between the underside of the furnace components and the flue bottom for removing clinker from the ash chamber. This process does not interfere with furnace conditions, nor occasion any measurable loss in thermal efficiency.

Clinker Removal.—Clinker is removed through a chain-operated, semi-circular door by means of a specially designed shovel.

Operation.—The coal, which may be fed to the hopper by hand or mechanical means, passes to the feed-box, where the required quantity is delivered to the shovel, which then throws the fuel and spreads it over the fire in zones covering the entire grate. The fuel is slowly carried by the Self-cleaning Air-Draught Furnace to the rear end of the furnace bars; clinker and ashes are deposited in a chamber between the ends of the bars and the bridge, to be withdrawn as outlined above.

CROSTHWAITE MECHANICAL STOKER.

(*Crosthwaite Furnaces and Scriven Machine Tools Ltd., York Street Ironworks, Leeds* 9.)



Crosthwaite Mechanical Stoker and Self-cleaning Furnace.

The cheapest and smallest fuels of almost all kinds can be burned with high efficiencies with this sprinkler type of mechanical stoker. The illustration shows the application of the type to Lancashire and Cornish boilers. It works with equal efficiency on Water Tube and Sectional

Heating boilers. On the latter it is one of the very few plants that will handle all fuels from coke breeze and anthracite to high volatile coals. It is manufactured by Crosthwaite Furnaces and Scriven Machine Tools, Ltd., Leeds.

Beneath the hopper (A) there is a feeder slide (B), which has a forward and backward motion actuated by a cam. The stroke of the feeder can be regulated easily to suit the rate of feed required even whilst the stoker is running. The feeder pushes the fuel in small quantities over the edge of the feed-box (F) to the firing nose, dividing it during this process. The fuel is then thrown on to the grate by the swinging shovel (H). This shovel is carried on a central arm, which swings in a passage directly through the centre of the feed-box, and is almost noiseless in action.

The swinging shovel is actuated by a tension spring and a hardened roller working on a three-point cam. A special device permits instant release of the shovel the moment this reaches the limit of its backward stroke. This greatly reduces the spring tension, and therefore the power required to drive the stoker.

The throw of the shovel can be adjusted by means of the spring tension, and at its minimum the stoker, in conjunction with the self-cleaning furnace, becomes practically a coking type stoker.

Crosthwaite Self-cleaning Furnace.

This furnace is of the reciprocating type. The grate is divided longitudinally into close-fitting sections. Each section is a separate and complete unit carried in a mild steel channel. A set of cams push forward all the sections in unison, moving the fuel slowly towards the back of the grate. Independent cams return each section successively to its original position.

The forced draught is created by fan-driven air or by steam jets, each section having one jet.

The self-cleaning furnace is suitable for either hand firing or automatic stoking. The Crosthwaite Stoker and Self-cleaning Furnace are driven by separate gear-boxes, so that the speed of the stoker may be altered independently of the furnace.

This grate is suitable for all fuels except those of the pure anthracite type. It is especially designed to burn, with the minimum of smoke, low grade fuels from slurry upwards on boilers working under heavy loads. It gives complete combustion and the boiler load can be held fully during the periodical removal of ash.

Crosthwaite Fixed Forced Draught Furnace.

This furnace will burn efficiently the lowest grades of fuel. It maintains constant and even heat, and the upkeep costs are claimed to be very low. The firebars will last practically the lifetime of the boiler. Forced draught is created either by steam jets or by fan air. Steam consumption of the jets on the Crosthwaite furnace is not more than 2½ to 3 per cent. of the steam raised. The initial cost of steam jets is low. The initial cost of fan air is somewhat higher, but the running cost is less than that of steam jets if electric power is available at a reasonable cost. Fan air gives a quicker steaming start, a higher boiler duty and it is silent. In the Crosthwaite fan-driven air furnace, steam jets are fitted as a standby.

Sawdust, wheat chaff, bagasse, coal dust, slurry and indeed any combustible refuse can be burnt with the highest efficiencies in the Crosthwaite forced draught furnace.

Type 'L' Travelling Grate Stoker for Water-Tube Boilers.

(International Combustion Ltd., Nineteen Woburn Place, London, W.C. 1.)

This machine in its standard form is capable of burning practically any type of coal or coke, and in a modified form, wood refuse, Bagasse or similar fuels. It can be made in sizes up to 850 sq. ft. of active grate area and for steam ratings up to 400,000 lb. per hour with one stoker.

The design of the moving grate is an outstanding feature of this stoker. Referring to fig. 1 it will be seen that the grate bars are disposed transversely on the grate and lie upon each other like roof tiles. Each bar is pivoted at its ends and overbalances as it passes around the rear idler of the stoker, thus dislodging any adherent clinker or riddlings. This relative movement of adjacent links produced by pivoting neighbouring bars on different centres is claimed to be an exclusive feature, and accounts largely for the success which has attended this machine since it was first introduced in 1926.

The firebars are strung upon endless steel chains occurring at intervals across the grate and spaced correctly by tie rods of special design. The chains are equipped with rollers which take the drive from the driving sprockets and also carry the grate during its journey through the furnace.

The grate is self-cleaning, the air spaces between the bars remaining clean throughout the life of the bar. Any bar can be renewed at any position in the grate without disturbing other bars. If a bar breaks and falls out, the resulting space is filled immediately by the preceding bar.

The main tension chains are completely protected from the heat of the furnace and fuel bed. The grate as a whole is flexible, and therefore unlikely to suffer severe damage in the event of an obstruction causing the chains to mount the teeth of the driving sprockets.

The second outstanding feature of the 'L' stoker is the method of air distribution. Instead of having large, deep, air compartments connected to air supply ducts at the sides, the air is delivered to a large undivided reservoir beneath the grate and distributed by adjustable gates arranged in frames having shallow hopper-shaped depressions, immediately beneath the grate. There are several gates in each transverse row, comprising one combustion zone, all the gates in each row being operated simultaneously. This construction gives the maximum possible volume of air supply chamber beneath the distributing gates, thus reducing the chances of varying air pressure caused by velocity head and eddies.

Combustion is under perfect control, enabling widely varying fuels to be burned efficiently. The whole of the main frame of the stoker is cooled by the air for combustion, and the upper transverse members are completely protected from radiant heat by the air distribution gate frames which are substantial iron castings.

In the foregoing the 'L' stoker has been dealt with in its most widely-used form, viz. as a forced draught stoker fed with air from an external fan. It can be applied in other forms, including the self-contained type in which the forced draught fans and driving mechanism are arranged on the stoker itself and the natural draught machine, which is a simplified version of the forced draught stoker.

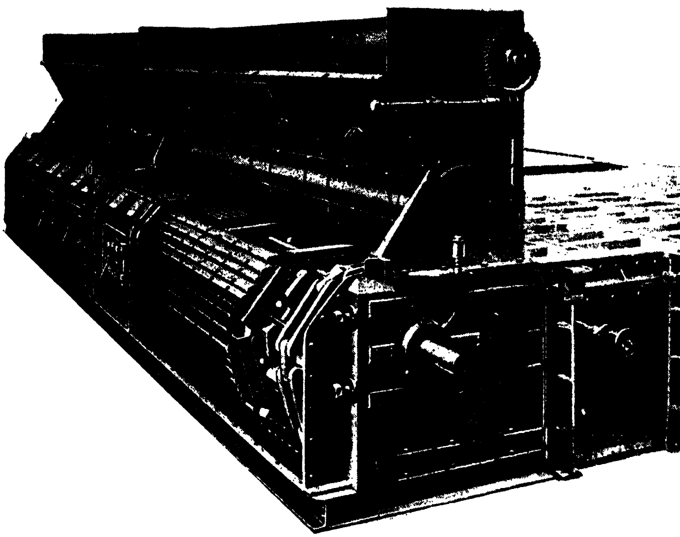


FIG. 1.—Part view of the Type 'L' Stoker, showing sections of the front removed to show the interior grate construction.

Various types of driving mechanism can be supplied to meet different cases. The most common type is the self-contained gear box giving either four or eight speeds and driven by an electric motor. In cases where automatic boiler control is to be employed, a special gear can be supplied suitable for use in conjunction with a variable speed motor. Where standby driving facilities are essential, they can be arranged by installing line shafting either above or below the firing floor, connected to the alternative driving units through the medium of clutches or fast and loose pulleys, and driving the stoker gears either through chains or vertical shafts.

The Riley Spreader Stoker.

(International Combustion Ltd., Nineteen Woburn Place, London, W.C. 1.)

The Riley Spreader stoker can be fitted to any type of water tube boiler and combines many of the advantages of pulverised fuel firing with those of a mechanical stoker, the fines being burned in suspension and the heavier particles distributed over the grate. The fuel is thrown or sprinkled over the entire grate surface by means of revolving plates forming the coal feeding units, one or more of which are mounted on the front wall of the furnace above the grate level. The fresh coal falls on top of the fuel bed and ignition is instantaneous. Forced draught is introduced under the grate and this chills the ash which is always on the underside of the fuel bed tending to reduce the formation of clinker and acting as a heat insulator to protect the grate from the high temperature of the furnace.

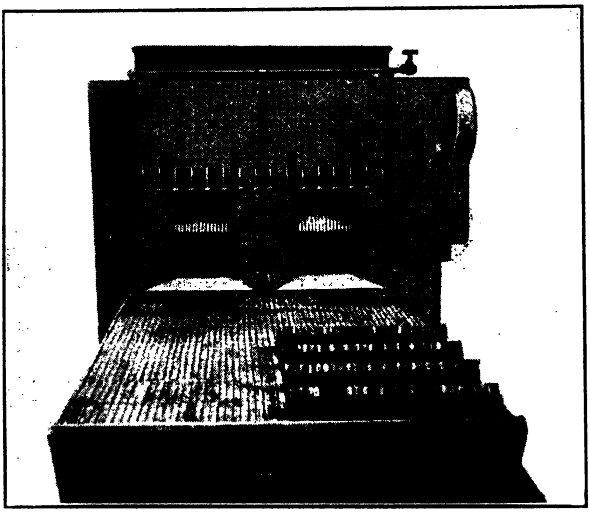


FIG. 2.—Riley Spreader Stoker.

Ashing is carried out by rocking the grate (which is divided longitudinally for this purpose), or in the travelling grate type by moving the grate to give continuous ashing either at the front or rear as may be convenient. This moving grate should not be confused with the better known travelling grate stoker which feeds the coal; on the spreader stoker the speed of the grate is regulated by the ash content and is consequently only a fraction of the rate required for coal feeding. As the fly ash and grit carry over is considerable a collecting and re-firing equipment is a normal feature of a spreader stoker installation; re-firing is carried out by means of a high pressure fan.

The spreader stoker will take any kind of rough slack coal, and thus permits of a wide choice of fuels. Coals high in ash content can be burned economically on the travelling grate to obtain the benefits of continuous ashing. The method of burning ensures a high rate of combustion and great flexibility in picking up and dropping boiler load without pressure variation. Automatic control can readily be applied, and as the power absorbed is low with either type of grate the stoker makes an attractive proposition for boiler evaporation up to say 200,000 lbs./hr.

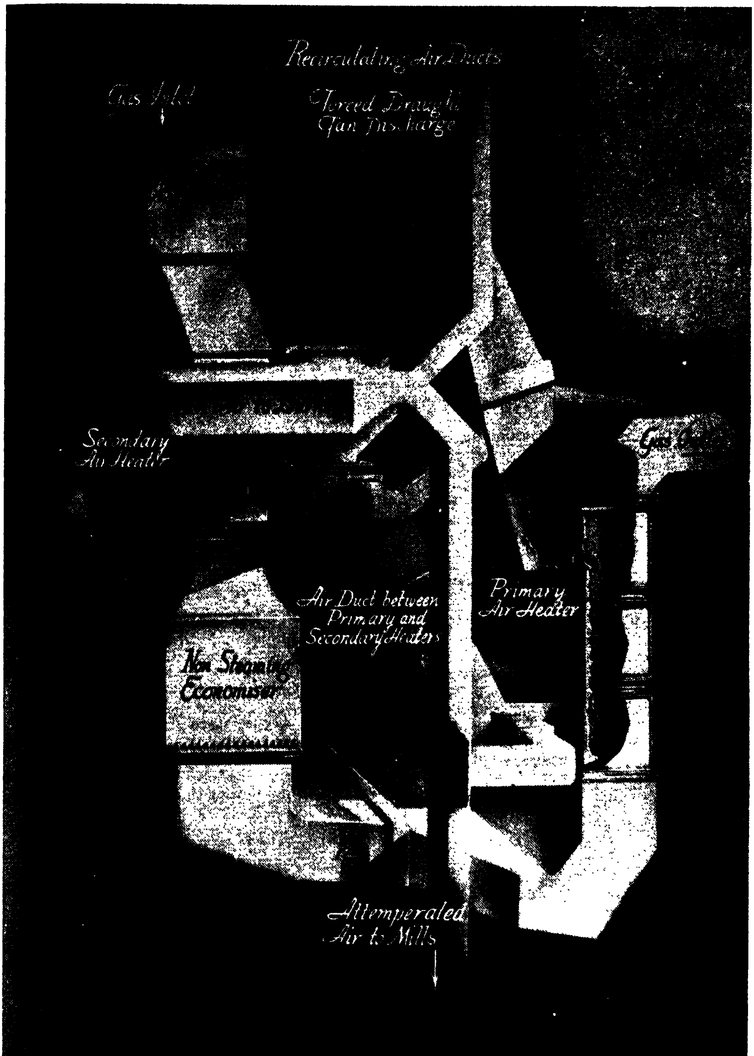


FIG. 1.—'Usco' Plate Type Air Heater.

THE 'USCO' PLATE TYPE AIR HEATER.

(International Combustion Ltd., Nineteen Woburn Place, London, W.C. 1.)

The 'Usco' plate type airheater is a high efficiency heat recovery unit providing preheated combustion air and by virtue of a special arrangement is peculiarly suited for pulverised fuel firing, where a high degree of preheat is required.

In construction, the air heater consists of a steel casing in which is contained a number of plate elements, each element comprising two plates electrically welded together to form an air- and gas-tight envelope. The edge of each envelope is pressed in such a manner that they present a perfectly streamline face to the passage of air or gas, with the result that no undue loss of energy occurs. Each envelope and the plates of the envelopes themselves are kept their uniform distance apart by means of special streamlined distance pieces, which do in effect help the passage of air through the heater.

The envelopes or elements are welded together in batches of ten, each batch being provided with a lug to permit of easy erection at site. These groups are then assembled in the casing and bolted, the casing is welded and the whole construction rendered perfectly air- and gas-tight. The 'Usco' air heater thus divides the flue gases to be cooled and the combustion air to be heated into alternate parallel streams which flow in opposite directions. The air flows through the elements or envelopes and the gases through the spaces between the adjacent elements. It will be noticed that the gas flow takes a straight path, whereas the air flow changes in direction both at inlet and outlet. The air and gases are made to flow in counter current directions in order to maintain a maximum and more or less constant temperature difference.

Plate temperature is an important factor in the operation of any type of plate heater, since the temperature must be sufficiently high to avoid condensation of the moisture present in the flue gases which causes choking of the gas passages and ultimately, by combination of the moisture with the oxides of sulphur contained in the products of combustion, corrosion of the heating surface by sulphuric acid attack. In the 'Usco' heater arrangements are made for a suitable proportion of the air to be recirculated so that the element metal temperature is maintained at the optimum figure under normal conditions of operation.

Plate heaters are usually disposed to suit vertical gas flow, the air being introduced at the sides of heater in a variety of arrangements. As applied to water tube boilers the heater is carried on an extension of the boiler structure or on the fan floor structure if the heater is above the boiler unit. For stoker fired boilers taking combustion air up to 300°F. a single heater is usual in series with the economiser if such be fitted. For pulverised fuel firing where air temperatures in excess of 500°F. are usual, it is necessary to arrange two heaters in series with provision for attemperated air supply to the mill. A typical layout is shown in fig. 1; the secondary air heater is located between two sections of economiser so that the gas flow is, in sequence, through steaming economiser, secondary air heater, non-steaming economiser, primary air heater to gas outlet. The air circuit is from the forced draught fan discharge through the primary air heater; thence by interconnecting ducting to the secondary heater with outlet to burners. Attemperated air to mills is taken from both primary and secondary heater outlets; there is an air byepass on the primary heater, while the recirculating ducts enable the F.D. fans to draw a percentage of heated air from the interconnecting duct for recirculation through the primary heater. Control of recirculation, attemperated air, etc., is by means of remote controlled dampers. Such an arrangement provides in an efficient way for the requirements of the pulverised fuel equipment and is, despite the apparent complication, simple in operation and maintenance.

Soot blowers are not now fitted to 'Usco' air heaters, as they merely increase the liability to corrosion by the introduction of additional moisture without presenting any advantages in maintaining cleanliness. In fact recent investigation into the source of bonded deposits has shown that soot does in many cases aggravate such deposits in that the dust cakes. Ample and well disposed cleaning doors for use during periods of overhaul have been found much more useful.

THE 'ROBOT' AUTOMATIC STOKER.

(Riley Stoker Co. Ltd., Nineteen Woburn Place, London, W.C. 1.)

The Riley 'Robot' Automatic Stoker (fig. 1) is of the underfeed type, designed to give smokeless combustion when burning bituminous and semi-bituminous coals. It is arranged for fully automatic control by thermostats or pressurestats, and other forms of control, such as time switches, can also be used. The stoker is of British design and construction throughout, being manufactured at the Derby Works of the Company's associates, International Combustion Limited.

The stoker consists generally of a coal hopper, worm conveyor, stoker body and firepot, variable speed gearbox, forced draught fan, electric driving motor and automatic controls. It can also be arranged to feed direct from the coal store to the boiler, in which case the coal hopper is not supplied.

The stoker body is of heavy cast iron construction and is made in two parts for easy assembly. The air duct and air chamber are formed within these castings, so that no brickwork is required for the construction of the air chamber. The firepot castings are in segments which fit into a grooved holder casting, and thus any segment can be replaced without the necessity of renewing the whole firepot.

On machines above the No. 1 size, a surrounding grate of firebars is fitted where possible. This grate assists in the final combustion of the fuel and enables the fine ashes to be riddled through the fire bars into the ash pit below. A feature of the surrounding grate is that the fire can be maintained by hand stoking in the event of failure of the electric supply.

The driving gear is enclosed in a dust-tight cast iron case and is designed for silent running. A small pump ensures positive lubrication to all moving parts. A safety clutch is provided, which prevents damage to the gearing in the event of tramp iron, or other hard substances, being in the coal.

Feed variation is effected by varying the number of ratchet teeth engaged by the driving pawl. When automatic controls are fitted, a small pulling motor is mounted behind the gearbox and this automatically puts the stoker on to the required feed, at the same time adjusting an air damper to suit. Thus the amount of air supply is automatically maintained in correct proportion

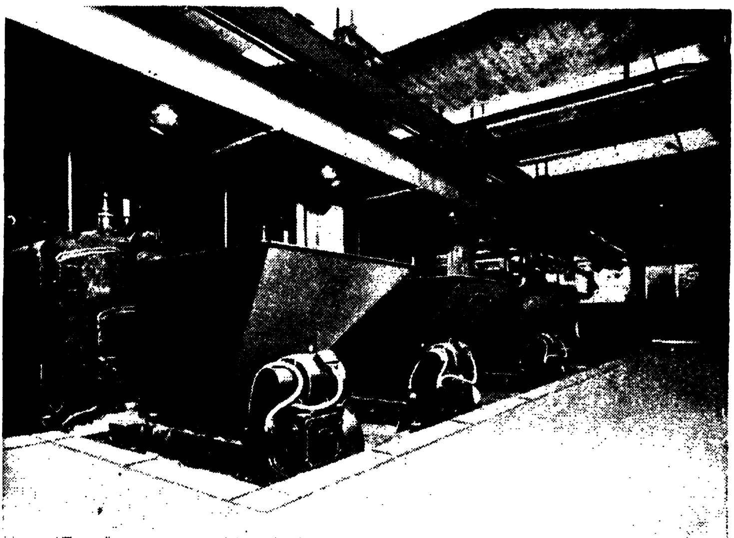


FIG. 1.—' Robot ' Automatic Stoker.

to the amount of fuel fed to the furnace. The pulling motor is influenced by a main controlling thermostat, or pressurestat, which is fitted to the boiler and a second thermostat is provided to act as a limiting device, should the temperature continue to rise when the stoker is on low feed. A special control box is provided for the electric driving motor, complete with master switch, fuses, overloads, relay and terminals suitably wired for the automatic controls.

The Riley ' Robot ' Stoker is suitable for firing all types of sectional, vertical and locomotive boilers and is also used extensively for firing core stoves, forge and annealing furnaces, brick kilns, dryers, air heaters, oil heaters, etc. It is made in several sizes to burn from 12 to 1,000 lb. of coal per hour.

THE CLASS ' B ' STOKER FOR FLUE TYPE BOILERS.

(*Riley Stoker Co., Ltd., Nineteen Woburn Place, London, W.C.* 1.)

The Class ' B ' Stoker is similar in principle to the ' Robot ' Automatic Stoker described above, and it has been specially designed for boilers of the Lancashire, Cornish and Economic types. The stoker has single centre retort with terraced grates on either side, and the air chamber is formed by a curved plate, to suit the circular flues of these types of boilers. Combustion space is limited in these circular flues but smokeless combustion is achieved due to the underfeed method of firing.

The Stoker consists of a coal hopper, worm conveyor, retort and grate, air box, air chamber and furnace front fitted with fire doors. It is a forced draught stoker, and is supplied either with a self-contained dust-tight gearbox, forced draught fan and electric driving motor, or with a ratchet

gear which can be driven from overhead shafting. With the latter arrangement, the air is supplied from a separate forced draught fan through underground or overhead ducting.

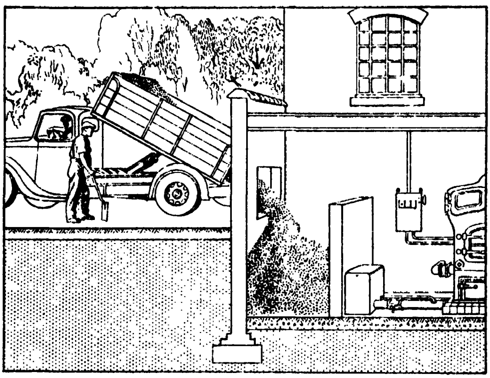
The Class ' B ' Self-contained Stoker can be auomatically controlled by thermostats or pressurestats. The controls are similar to those described above for the ' Robot ' Automatic Stoker.

The 'Iron Fireman' Stoker.

(Ashwell & Nesbit, Ltd., Barkby Road, Leicester.)

The disadvantages of using coal as a fuel in relatively small plants have been successfully overcome during recent years by automatic stokers. A prominent example is the ' Iron Fireman,' which is manufactured by Ashwell & Nesbit, Ltd.

This stoker is specially designed to ensure complete combustion, and this feature, in conjunction with the automatic controls supplied, effects a considerable saving in fuel costs; as much as up to 50 per cent. has, it is claimed, been saved over hand-firing, and up to 75 per cent. over oil-firing. In addition there is a large reduction in labour charges as only one hour or less per day of 24 hours is required for attention.



The bunker-to-boiler type stoker illustrated is the latest design to eliminate handling of coal. The coal is delivered to the bunker by the coal merchant and is not handled by the boiler attendant at all. In this type of machine the drive unit is located between the coal store and the boiler and the length from the face of the coal store to the front of the boiler could be any increment from 5 ft. to 20 ft.

This machine is made in many sizes from 200,000 B.T.U's. per hour to 10,000,000 B.T.U's. per hour.

The machine burns coal of the ' pea ' or ' bean ' size, which can be obtained at a much lower cost than either coke or normal steam coals. The standard bituminous machine is made in 14 sizes from 100,000 to 10,000,000 B.Th.U.'s per hour.

The correct volume of air required for combustion is forced through venturi air ports under the fire by a multi-vane fan. On the suction eye of this fan is fitted an iris type damper by means of which the volume of air delivered can be very closely regulated. The worm is driven through a multi-speed oil bath gear-box by a small electric motor. This small motor also drives the fan.

The coal is conveyed from the hopper by means of a worm to a retort which is fixed inside the boiler or furnace. The top layer of coal is incandescent, and as the green coal is fed underneath the fire, the gases which are released at a comparatively low temperature must pass through the incandescent layer of fuel and are thus completely burned, ensuring perfect combustion and absence of smoke. The ash in the coal is allowed to fuse into clinker and is removed once or twice per day.

The transmission to gear-box and fan is by means of ' V ' ropes, and as the motor is mounted on an insulated base plate, the whole unit is quiet in operation.

To safeguard the mechanism various safeguards are fitted, one being a safety shear pin in the event of tramp iron being jammed in the worm. The obstruction is forced to jam at a special place where it can be easily removed and the shear pin replaced in a few moments. A thermo trip device is also located in the motor, so that in the event of the motor overheating from any cause it is stopped.

The stoker is of pressed steel construction where possible and is cellulose sprayed, the finish giving it a very pleasant appearance.

Each stoker is provided with a full set of automatic electrical controls which cut out the motor on pre-determined temperatures or pressures being reached, and start the motor again when more heat is required.

The stoker can be applied to most industrial processes which call for heat, and the makers report that it has been applied successfully to cast-iron sectional heating boilers, domestic hot water service boilers, vertical steam boilers, underfired multi-tubular boilers, core and mould stoves for foundry work, oil heating furnaces, metallurgical furnaces, bakers' ovens, greenhouse boilers, tea dryers, hot air drying stoves, brick and tile kilns.

Superheaters.

'MELESCO' SUPERHEATERS FOR WATER TUBE BOILERS.

(*The Superheater Company, Limited, London and Manchester.*)

This superheater (fig. 1, p. 1230) is suitable for any type of water tube boiler. The elements are of the multiple loop type fabricated by means of the Superheater Company's machine forging process.

For intermediate pressures and temperatures up to 850° F. the element connections to the headers are of the ball and cone metal to metal type and individual elements can therefore be detached without damage to the tubing, joint, or adjacent elements.

For high pressure and temperatures in excess of 850° F. a short section of tubing is welded to the headers during construction and individual elements welded to these short sections during the erection of the superheater on site.

No header hand-hole fittings are required with either of these 'MeLeSco' joints.

'MeLeSco' superheaters for water tube boilers are usually either of the pendant type in which the tubes are arranged in a substantially vertical manner, or the self-draining type having horizontal elements. Each arrangement possesses certain advantages and the type it is preferable to employ depends upon a number of factors, foremost among which is continuity of operation of the plant.

With high temperature superheaters every precaution must be taken when raising pressure and placing the boiler on line to prevent overheating of the superheater tubes, and in the case of intermittent service the self-draining arrangement is of advantage from this operating point of view.

The pendant superheater has the advantage that element tube distortion is minimised and the whole superheater can be carried either from the superheater headers or from external steel-work in a simple and robust manner, whereas the horizontal elements of a self-draining superheater must of necessity be supported by slings which are exposed to high temperature gases and subjected to considerable loading. The use of heat-resisting alloy steel for this purpose is therefore unavoidable, but in 'MeLeSco' superheaters of this type the amount required is reduced to a minimum by employment of steam and water cooled supports of special design.

'MELESCO' INTER-CONTROLLED SUPERHEATERS.

(*The Superheater Company, Limited, London and Manchester.*)

The 'MeLeSco' Inter-Controlled Superheater (fig. 2, p. 1231) consists of primary and secondary superheater sections having a non-contact desuperheater connected in series between them. The desuperheater comprises a number of U-shaped steam tubes submerged in water, fed from the boiler drum, the water evaporated in the desuperheater being discharged to the boiler drum. Regulation of steam temperature is affected by varying the proportion of the total steam which is passed through the desuperheater on its way from the primary to the secondary superheaters The final steam temperature, changes in which determine the action of the control valves, is measured at the outlet of the secondary superheater by a thermostat, suitable means being provided to transmit the direction and extent of these changes to mechanism which in turn control. the power medium employed for operating the valves.

The desuperheater transforms superheat into latent heat and, because it provides a means by which additional heat can be absorbed by the superheater, there is an improvement in overall efficiency which more than offsets any loss due to radiation from the desuperheater and piping connection. Such a system makes it possible to maintain constant steam temperature at the turbine stop valve regardless of variation in boiler loading and other operating conditions and affords positive protection against damage due to excessive temperature.

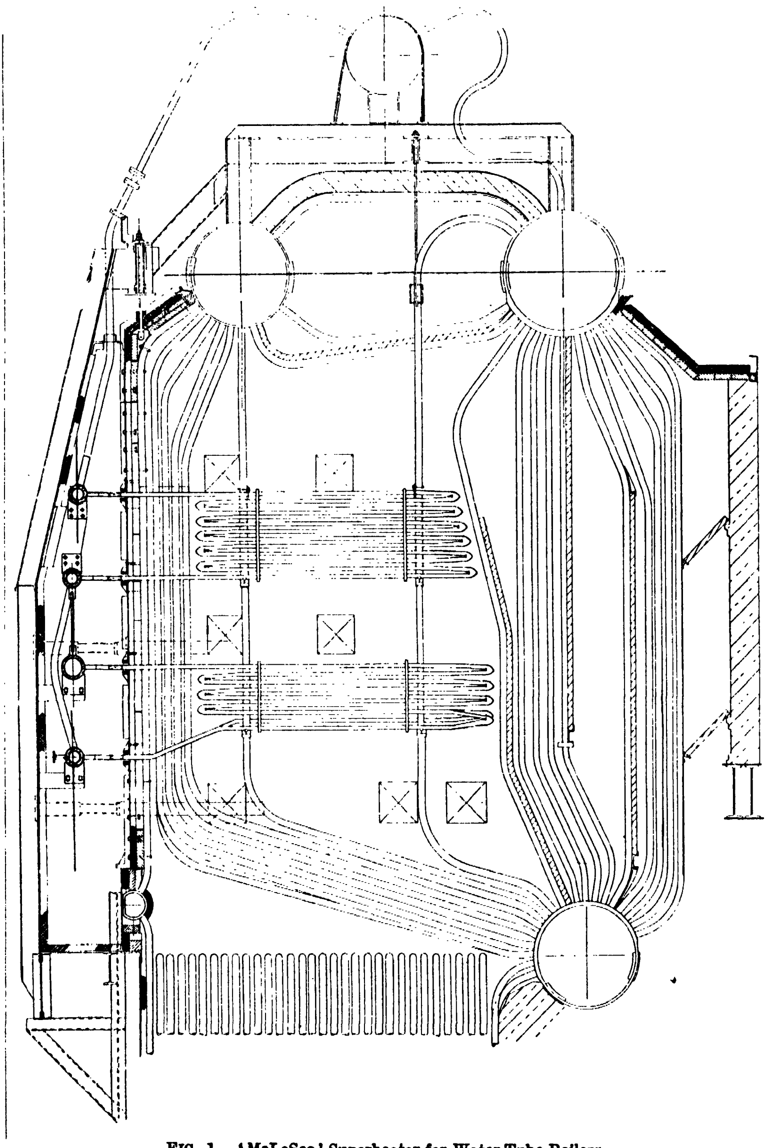


FIG. 1.—'MeLeSco' Superheater for Water Tube Boilers.

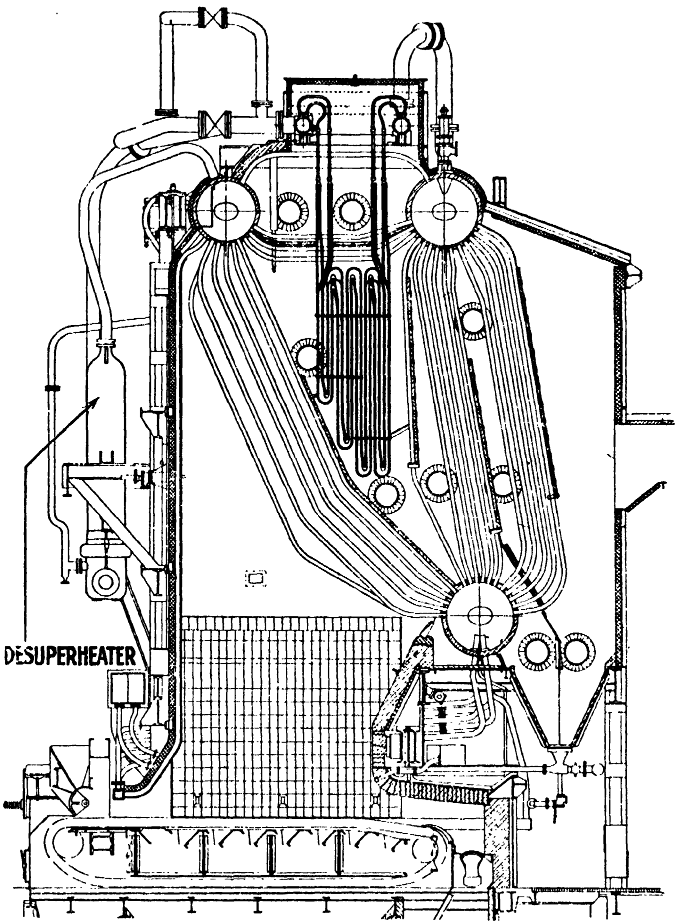


FIG. 2.—Arrangement of ' MeLeSco ' Inter-Controlled Superheater, showing Desuperheater.

Howden-Ljungstrom Air Preheater.

(*James Howden & Co., Ltd.*, 195 *Scotland Street, Glasgow, C.* 5.)

Application.—Air preheaters are installed just before the chimney in fuel combustion apparatus to recover from the flue gases any possible remaining heat before escape to atmosphere and by heating the combustion air to return this heat to the system.

As the temperature of the flue gases leaving an economiser must be 40 to 50° F. higher than the entering feed water temperature, some surplus heat is usually available in the combustion gases

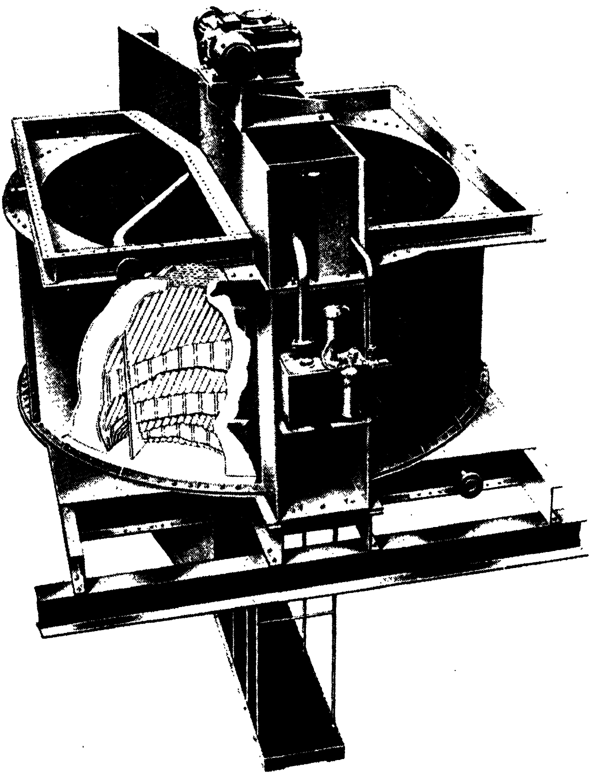


Fig. 1.

at the economiser outlet. The air preheater recovers some of this heat, thus improving the overall efficiency of the unit. Also, the use of preheated air improves combustion conditions; in fact, some low calorific value, low volatile, very high moisture or ash content fuels cannot efficiently be burned without it. With preheated air these inferior fuels are more readily ignited, leading to higher CO_2 content in the flue gases, reduced flue gas loss and thus increased boiler efficiency.

A reduction of 30 to 33° F. in flue gas temperature in the heater corresponds approximately to 1 per cent. saving in boiler efficiency. Normal gas temperature reductions in the air heater might amount to about 160° F. on stoker fired boilers and up to 450° F. with pulverised coal fired boilers, so the justification for the installation of air preheaters is evident.

Large electricity generating stations are among the largest users of preheated air.

The factor limiting the heater performance on stoker fired boilers is the maximum air temperature consistent with the avoidance of excessive renewals of stoker links.

Description.—The essential features of the Howden-Ljungstrom air preheater are the rotor containing the heating surface which forms the regenerative mass, and the casing so divided and arranged to direct flue gases axially through one side of the rotor while air for combustion passes through the other side as the rotor slowly revolves.

The rotor is an open-ended drum divided radially and concentrically into sections wherein the heating elements are carried. The heating elements are mild steel plates arranged on stoker fired boilers in packs to facilitate removal and replacement. These plates are shown in the accompanying illustrations, figs. 1 and 2. They are alternately undulated and notched, with the

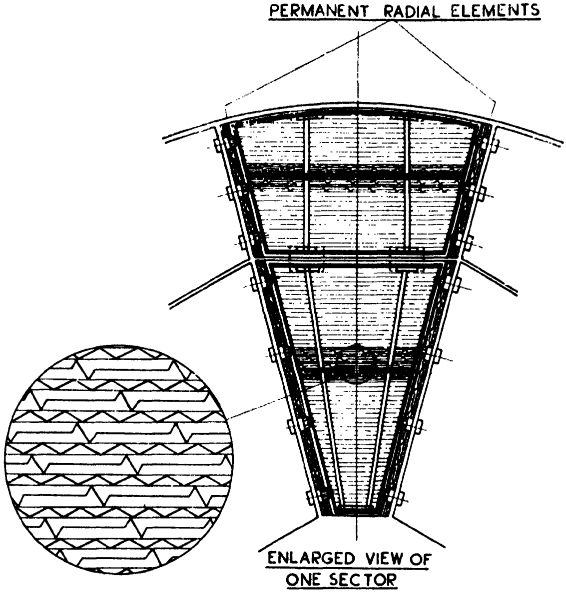


FIG. 2.

undulations running at 30° to the axis of the rotor and spaced by the notched plates which are of wider pitch and deeper than the undulations. Double turbulence of the gas flow is obtained by the sinuous passage between the undulated and notched plates in one direction and by the angle of the undulations to the direction of flow throwing the gases against the notches in the other direction.

The variation in temperature of the heating elements at any one point between the hot side and the cold side of the preheater during a complete revolution of the rotor is only a few degrees.

Howden-Ljungstrom air preheaters are fitted with a central gear drive incorporated in the main bearing housing and lubricated through an oil circulating pump with which is incorporated a filtering system, and, in special cases, an oil cooler. Both drive and bearings are designed to give at least twenty years' continuous service, and normally no replacement should be necessary during the life of the boiler.

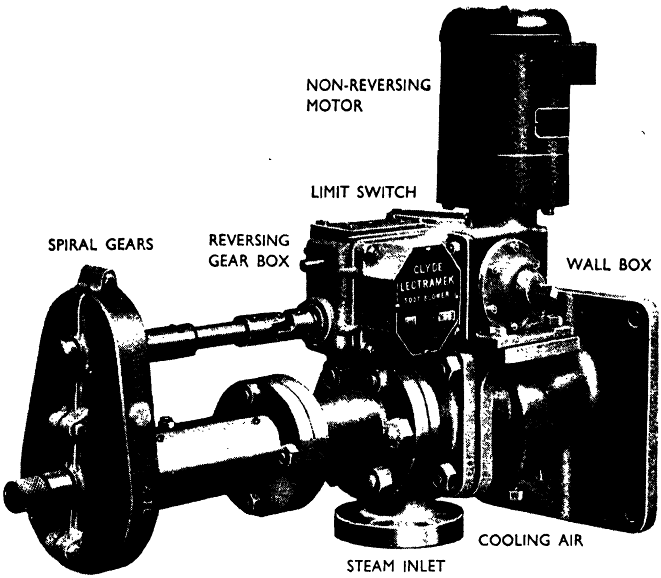
The rotor revolves slowly, ¾ to 3 r.p.m. being the usual limits of speed, and the power required to drive it is low, seldom exceeding 1 h.p. even for larger sizes. The driving motor, however, is always rated higher to provide the greater torque required at starting.

Soot-Blowers.

Clyde Soot-Blowers—Water Tube Boilers.

(*Clyde Blowers, Ltd., Livingstone Street, Clydebank.*)

The Clyde Blower, Single Nozzle Type, for water tube boilers, is built to deal effectively with the removal of slag and other deposits formed upon the furnace tube surfaces during steaming, and is designed for easy manipulation, durability in service, and accessibility in periodical examination. It is attached as a permanent fitting to the boiler walls, the nozzle being introduced into the furnace through a wallbox. A connection to the boiler provides steam, which is discharged in a powerful jet under full control of the operator. In order to spread the jet over the tube surfaces the nozzle is rocked to and fro. This is effected automatically, the motion being transmitted to the nozzle through mechanism to which the operating wheel is connected. When the blowing is completed, the nozzle is retracted and protected from overheating by a stream of cooling air introduced into the wallbox. The operating mechanism is designed for direct or remote control and an indicator is provided in close proximity to the operating hand-wheel.



Electrical operation of the Clyde Soot Blower is obtained by the application of a Lectramek Unit which is mounted on the wallbox and coupled direct to the blower mechanism. A control panel is provided for fixing at firing floor level for starting up the operating unit by push button.

The Operating Gear consists of a three-phase totally enclosed asbestos wound motor, squirrel cage non-reversing type, direct coupled to speed reducing gear driving bevel wheels mounted on a splined shaft, together with a reversing clutch automatically controlled by timing gear. The motor does not reverse but runs continuously in one direction, reversing movement being obtained by mechanical means, and is transmitted to the blower gearbox at reduced speed where it is converted to reciprocating motion.

A Limit Switch situated in the operating unit gearbox and connected to the control panel regulates the stopping and starting and the number of strokes made by the blower coming into action immediately the push button is pressed, and stopping automatically when the cycle is completed. The electo-mechanical operating gear described is applied as a self-contained unit to the blower and takes the place of the usual handwheel as fitted for manual operation.

The Control Gear consists of a dust proof steel cabinet containing all electrical contactors and fittings with an outside keyboard provided with push buttons to suit the number of blowers, each

button being protected by a cover plate with interlocking mechanism to ensure that only one blower is operated at one and the same time. The actual serial number of the blower in operation is recorded and appears through an illuminated window. Sequence control is available and is fitted in the cubicle when desired, whereby each blower functions automatically in correct rotation after being set in motion by a single start push button.

The necessary safety devices are incorporated in the cubicle together with transformer for alarm bell which comes into action in case of emergency.

Inspection Facilities.—Access to the mechanical portion of the Clyde Soot Blower is obtained as before by removing steam chest cover without breaking steam pipe joints, and it should be noted, without disconnecting electric supply wires attached to motor and limit switch. This facility is of great importance to the maintenance staff as it permits nozzle inspection or renewal to be made while the boiler is under steam. Another advantage is that the cover which contains all the working parts of the blower can be brought to the floor level where it is easily dismantled, adjustments made and replaced on the boiler in a short time.

The design of electrical operating unit has been standardised for application to retractable single nozzle blowers located on boilers in a vertical or horizontal position and to multi-jet and dual nozzle blowers as applied to economisers and air preheaters.

Clyde Soot-Blowers—Scotch Marine Boilers.

(Clyde Blowers, Ltd., Livingstone Street, Clydebank.)

Clyde blowers for application to multi-tubular boilers of the Scotch marine type are supplied for single and double-ended boilers, and are capable of operation from back or front ends. This necessitates three distinct types, classified as follows: S.M. Direct for single-ended boilers (back operated). D.F.O. Direct for double-ended boilers (front operated). R.F.M. Return flow for double-ended or single-ended boilers (front operated). Each of the above designs incorporates the usual 'Clyde' features, including self-contained steam shut off, obtained by the rotation of the blower operating wheel or handle, while both axial and oscillatory movements are also transmitted to the blower nozzle by the same wheel. A master stop valve on each boiler controls all the blowers on that boiler, and remains open whilst blowers are in use but steam is admitted to each blower nozzle in turn only when its operating wheel is used.

Standard blowers are fitted with hand-wheels or handles, for direct operation from platforms, but operating gear whereby complete control is obtained from the floorplates, is available, similar to that employed on Clyde blowers for water tube boilers. This facility enables the process of tube cleaning to be carried out without ascending ladders, or the use of platforms, and ensures that the work is effected expeditiously and with regularity. This EX type blower is applied for cleaning tubular and turbulent flow air heaters. Clyde blowers for marine type boilers and air heaters are supplied to Board of Trade requirements, suitable for superheated or saturated steam.

Clyde Soot-Blowers—Economic Boilers.

(Clyde Blowers, Ltd., Livingstone Street, Clydebank.)

The Clyde Soot Blower provides a simple method of maintaining the boiler tubes and heating surfaces free from soot without opening smoke-box doors. The time required for tube cleaning is reduced to a few minutes each day.

Another advantage is that in the process of tube cleaning with blowers, soot does not spread everywhere. The surroundings and atmosphere remain clean.

Boilers fitted with automatic stokers cannot be readily cleaned by any other method. The cost of tube cleaning by soot blowers is negligible as this is carried out by the attendant during his daily work.

Boilers can be run continuously for long periods without laying off as the tubes and heating surfaces are kept thoroughly clean by the regular use of the soot blowers. Coal consumption is thereby greatly reduced.

For a single pass economic boiler two blowers are fitted to the back of the combustion chamber, and for a double pass boiler it is necessary to fit one additional blower and in some cases two blowers at the smoke-box end. The equipment is simple to operate, and the boiler tubes can be kept consistently clean without laying off for hand cleaning, whereby considerable savings are effected both in labour and fuel.

Furnace Linings.

BIGELOW SUSPENDED WALLS FOR FURNACES.

(*Liptak Furnace Arches, Ltd.*, 68 *Victoria Street, London, S.W.* 1.)

Liptak Furnace Arches, Ltd., manufactures a complete range of unit supported fabricated walls for all types of furnace construction, ranging from very thin light weight air-cooled or insulated walls giving an insulating refractory cover of approximately 3 ins., as shown in fig. 1, to heavy duty type walls of double suspended construction either air-cooled or insulated, giving 11 ins. of refractory cover, as shown in fig. 2.

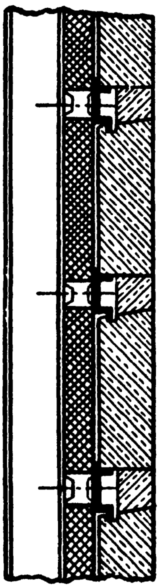


FIG. 1.

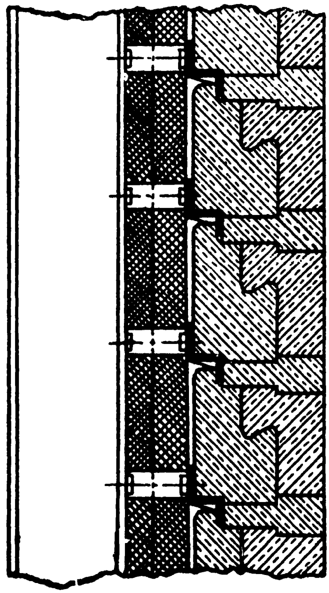


FIG. 2.

Features of these constructions exemplified in all designs within the range are that each block is independently supported and can therefore be replaced independently when necessary, and that a very much longer life is obtained from the refractory or insulating refractory material owing to the elimination of cumulative loading.

A full range of unit suspended double and single arches for boiler and industrial furnaces is also available.

Electrostatic Precipitation.

STURTEVANT ELECTROSTATIC PRECIPITATORS.

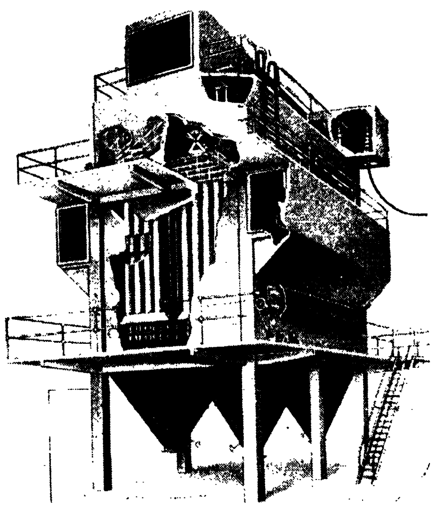
(*Sturtevant Engineering Co., Ltd., Southern House, Cannon Street, London, E.C.* 4.)

All industrial gases, as well as the atmosphere, carry solid or liquid particles which may be valuable commercially, or harmful to life and property, or of such a nature as to reduce the efficiency of some chemical process if allowed to remain in a chain of apparatus.

One of the most efficient means of removing these particles is the process of electrostatic precipitation. In this process, high voltage electricity is used to ionise the gas and its solid or liquid burden, thereby effecting the removal of the dispersoids.

The Sturtevant precipitator consists essentially of two sets of electrodes, one set charged to a high potential and termed the 'discharge electrodes,' and the other, which is earthed, called the 'receiving electrodes.' The discharge electrode system is made up of wires of suitable cross-section, and the receiving electrode system usually consists of a number of vertical tubes arranged in parallel. Each tube is equipped with a centrally suspended discharge wire and when these wires are charged to a sufficiently high potential a corona or brush discharge takes place, thus ionising the gas and dust and driving the solid or liquid particles to the receiving electrodes.

Continuous or intermittent motor-driven rapping gear is provided for relieving the two electrode systems of deposited dust which then falls into hoppers located on the underside of the precipitator casing.



The high tension equipment used by the Sturtevant Engineering Co., Ltd., is standard apparatus and gives constant and reliable operation over very long periods. The plants are so interlocked and protected that there is no danger to operators and every installation is equipped with apparatus for the prevention of interference with wireless reception.

These precipitators are used for the removal of a great variety of materials from industrial gases and the atmosphere. Powdered fuel grits, cement dust, atmospheric impurities, blast furnace dust, tar fog, pyrites dust, acid mist, crushed stone dust, lamp black, phosphate and various metallic oxides, such as those of tin, molybdenum, lead, zinc, vanadium and tungsten, as well as metallic tin, lead, zinc, nickel, gold and silver, etc., are amongst the solids and liquids collected by electrostatic plants.

In many cases large savings can be effected by the installation of an electrostatic precipitator, and even small plants, it is claimed, show a recovery of over £500 per week when used for the removal of certain metallic oxides from the flue gases in metallurgical works.

Considerable saving is also realised in acid works, which burn pyrites or blende, by the use of a precipitator located between the burners and the Glover Tower. An electrostatic plant at this point prevents the carry-over of burner dust and results in clean flues and towers, as well as clean, clear acid.

The majority of central power stations burning pulverised fuel use precipitators to eliminate the discharge of grits into the atmosphere and it is generally considered that this method is the most efficient and reliable one for doing away with the nuisance of grit emission from large stations. Most cement works are likewise equipped with electrostatic plants to clean the kiln gases before they reach the stack.

Feed Water Treatment.

THE WEIR 'OPTIMUM' DE-AERATOR.

(G. & J. Weir, Ltd., Cathcart, Glasgow, S. 4.)

Corrosion in boilers and feed systems can best be avoided by the use of de-aerated feed water The water in an open feed system absorbs gases from the air, and becomes corrosive; the feed can be rendered non-corrosive by the use of the Weir 'Optimum' De-Aerator.

In this de-aerator the feed water is sprayed through nozzles, meeting steam from a chamber below in which the water is collected and subjected to violent ebullition by incoming steam. The rapid boiling of the water, combined with the heating action of the vapour on the spray at reduced pressure, causes the air in the water to be liberated; the air is exhausted by means of an air ejector which maintains a vacuum in the de-aerator and assists in the de-aerating action.

All the heat in the operating steam is conserved in the feed water, and the de-aerator may be conveniently included in the feed system. The amount of oxygen in the water can be reduced to entirely negligible quantities with ease. The plant gives excellent de-aeration at low as well as at high temperatures, and may be designed to operate under either pressure or vacuum conditions.

WEIR EVAPORATORS.

(G. & J. Weir, Ltd., Cathcart, Glasgow, S. 4.)

High and low pressure evaporators (single and multiple effect) are built by G. & J. Weir Ltd., Glasgow, for both industrial and marine duty. A typical triple effect plant, comprising vertical evaporator shells with scroll type heating coil elements, is shown in the illustration (fig. 1).

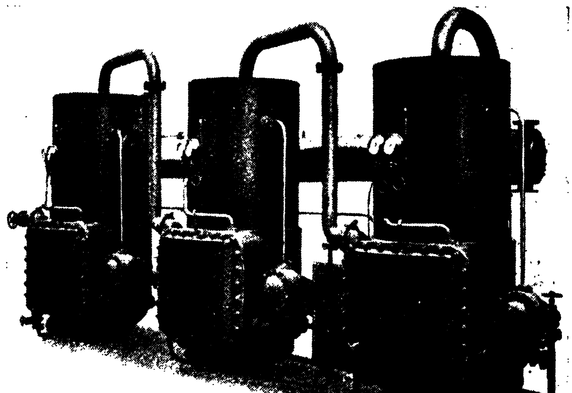


FIG. 1.—Weir Triple Effect Evaporating and Distilling Plant.

The coils are of heavy gauge, solid drawn copper, and each element may be withdrawn separately for cleaning. The water level in each shell is maintained by means of a float-operated automatic feed regulator. Special provision is made to secure freedom from priming. For marine work a brine ejector is fitted.

Fig. 2 shows a Weir horizontal low pressure evaporator, the elements consisting of straight tubes held between tube plates, large doors being fitted to allow for easy cleaning. The large water area for the release of vapour is a factor which retards any tendency to priming, and a large steam dome with baffle plates is provided.

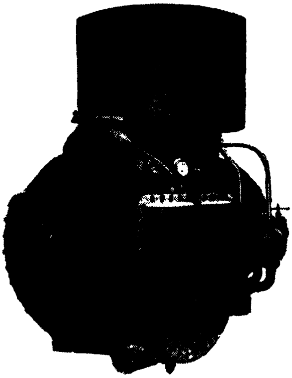


FIG. 2.—Weir Low Pressure Evaporator.

Evaporators should be installed in a power plant so as to form part of the system, the heat in the vapour produced being absorbed by suitable means so as to avoid loss. They provide an ideal means for protecting boilers, etc., from impure water, and amply repay the cost of installation. For the supply of drinking water in arid districts, plants can be operated either in conjunction with a power plant, or independently from a special boiler.

The number of stages in a multiple-effect evaporating plant depends on the steam conditions and the degree of overall economy desired, the amount of heating steam required diminishing as the number of effects increases. Two or three effects are the most usually employed in marine work, but quadruple and sextuple effect plants are frequently installed for the supply of fresh water in the tropics or in industrial works. The relative production of single and multiple-effect plants (*i.e.* the weight of distilled water produced per pound of primary steam supplied) is approximately as follows: Single effect, 0·8 to 1. Double effect, 1·5 to 1. Triple effect, 2·6 to 1. Sextuple effect, 4 to 1.

Corrosion.

BOILERINE.

(*Boilerine Ltd., Manufacturing Chemists,* 897 *Old Kent Road, London, S.E.* 15. *By appointment Manufacturing Chemists to H.M. King George VI.*)

This is a viscous fluid, consisting chiefly of a carbon compound which has no action upon iron, steel, or other metals, but which has the property of rendering all water impurities innocuous and incapable of forming scale, while it also works its way through the interstices of the scale, the rate of this percolation depending on the porosity and degree of compactness of the same, and when the preparation reaches the underlying plate of the boiler it there diffuses and insinuates itself behind the incrustation, causing it to crumble and flake, so that when the boiler is opened it can readily be removed. The preparation as a preventive of scale differs from the old style of boiler composition, inasmuch as it absorbs the gaseous carbonic anhydride from the water, forming an intermediate salt which throws the scale-forming salts out of solution, suspending them in the water in a non-crystallisable and powdery state. Incrustation is thus prevented. Steam generated from sea water containing 'Boilerine' is free from ammonia. Boilerine is not only a preventive, but also a disuniter of incrustation in boilers; it is absolutely non-volatile, and does not affect the purity of the steam produced from water treated with it; it does not discolour the water in the boiler, and it will not cause priming. It is guaranteed free from arsenic, acids, and poisonous substances, and is a neutraliser of grease and acids.

The boilers should be blown down and treated with Boilerine *daily*, as by this system greater efficiency is obtained.

BOILERINE RADIATOR CLEANSER.

If the cooling system of a motor vehicle does not receive periodical attention lime deposits form on the walls of the cylinder jackets and radiator tubes. Corrosion may also take place. There follows clogging up of the cooling system with, as a sequel, poor running, high oil consumption and increased wear.

Boilerine Radiator Cleanser has been introduced by Boilerine Ltd. to prevent these occurrences and is obtainable direct. It takes the lime into solution, is suitable for any climate, and keeps without deterioration.

ORGANIC 'ROSSELINE.'

This preparation is the ideal cleaner for domestic hot water boilers and pipes to remove the fur which causes a difficulty in readily obtaining hot water. The entire process usually takes about nine hours, the quantity requisite for this being *three* gallons. This is also a product of Boilerine Ltd.

BOILER MAINTENANCE.

(*British Paints Limited, Portland Road, Newcastle-upon-Tyne.*)

Six definite claims are made for Dampneys 'Apexior' for the internal treatment of boilers :—

(1) That corrosion from whatever cause arising will be prevented and if already rife can be checked in one application.

(2) That scale will be considerably reduced, entirely altered in character, rendered non-adherent and easily removed.

(3) That heat transmission will be appreciably increased, as certified by the National Physical Laboratory.

(4) That the process is simplicity itself.

(5) That the cost is very low, only comes once per annum at the most, and the treatment more than pays for itself.

(6) That 'Apexior' contains no chemical, does not function by means of chemical action, is non-poisonous, and is used regularly in boilers raising steam for direct contact cooking.

'Apexior' Compound easily applied once a year with a brush provides a thin but impenetrable barrier between the boiler metal and the feed water. It has been tested by the National Physical Laboratory, which found its powers of adherence unimpaired at a temperature of over 1,000° F.

Mechanical equipment is now available for the speedy and efficient coating of tubes, which can now be coated as easily as flat surfaces.

Engineers do not require to conduct periodical tests with their feed-water, or to arrange for periodical dosing of their feed tanks, as is the case with fluid remedies ; there is one operation, one cost, and the saving in labour cleaning and in renewals more than counterbalances the small expenditure. A Lancashire Boiler 30′ × 9′ can be treated for a few pounds for the initial treatment, *i.e.* two coats, and the cost is halved during subsequent years when only one coat is necessary.

More than one firm of boiler manufacturers automatically 'apexiorise' their boilers before despatching to clients in order to provide a corrosion-proof interior ; others actively recommend 'Apexior,' and its merits are well known to the majority of the principal Boiler Insurance Companies.

'Apexior' works very well in connection with water-softening plants, many of which do not deal effectively with dissolved oxygen or free carbon dioxide, two very fruitful causes of corrosion, and the whole process is so simple and reliable as to be a revelation to all who have not had the experience.

'Apexior' can also be used with marked effect in evaporators, the scale being quite non-adherent can be easily removed by a blow from a piece of hard wood, and the cleaning is carried out in a fraction of the time usually occupied.

'Apexior' has been used in boilers, economisers, condensers, calorifiers and steam turbines for thirty years, has been adopted as a standard treatment for all types of steam plant by a great number of the leading steam users—on land and sea—throughout the world, and is at present specified for the biggest boilers in the course of construction in this country.

'FOLIAC' BOILER GRAPHITE.

(*Graphite Products, Ltd., 52 Battersea Church Road, S.W. 11.*)

The use of this material enables scale in a boiler to be easily and safely removed, a wire brush being used for the purpose. The graphite does not attack the metal of the boiler, nor, it is claimed, does it cause the metal to be attacked ; it is not affected by any acid in the water or by the heat generated in the boiler. Particles of the graphite work through the minute fissures existing in the old scale and gradually penetrate between the scale and the metal. The scale thus loosened may be rapped off or removed with regular cleaning tools with little trouble. If the scale is very hard and thick it may require as long as three or four months for graphite to loosen it, but once removed the scale can never again adhere firmly to the metal as long as the graphite treatment is continued. The graphite also becomes thoroughly intermixed with new scale as it forms, rendering it soft and friable.

The graphite is introduced in the form of fine powder with the boiler feed, about half a pound being used per day for a 250-h.p. boiler.

Boiler Fittings.

Water Gauges and Cocks.

(Richard Klinger Ltd., Klingerit Works, Sidcup, Kent.)

One of the difficulties with which engineers in charge of steam plants have to contend is that of overhauling and repacking efficiently the various asbestos-packed cocks fitted as water gauge cocks, salinometer cocks, pressure gauge cocks, etc.

A type of cock known as the Klinger 'Sleeve-Packed' Cock is extremely simple to repack and, at the same time, is capable of withstanding pressures and temperatures far in excess of those possible with the old-fashioned type of packed cock. The packing takes the form of a renewable

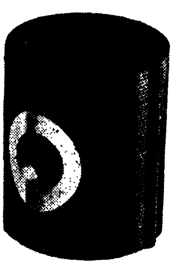


FIG. 1.

Renewable Asbestos Packing Sleeve for Klinger "Sleeve-Packed" Cock.

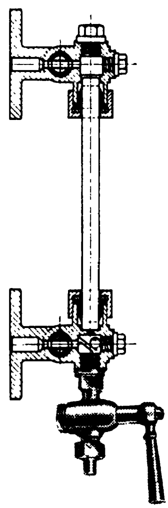


FIG. 2.

Klinger Water Level Gauge fitted with "Sleeve-Packed" Cocks.

sleeve of compressed asbestos (fig. 1), which is pushed into the body as a single unit and is compressed against the parallel plug by means of the bottom tightening nut. A ridge on one side of the sleeve fits into a groove in the body and makes it impossible for the ports to get out of line, while two stainless eyelets embodied in the sleeve around the ports protect the packing from the cutting action of the passing fluid.

Fig. 2 shows a standard pattern water gauge fitted with cocks of this type.

There is, however, a definite limit to the pressures at which it is possible to use a simple glass tube in water level indicators, and the growing use of higher and higher pressures in steam generating plant has necessitated the use of stronger types of gauges.

Fig. 3 shows the latest type of forged steel reflex water gauge made by Richard Klinger Ltd. for pressures up to 500 lb. per sq. in.

In this gauge the glass tube is replaced by a tough glass plate mounted in a forged steel body. The inside of this glass is corrugated with a number of V-shaped grooves. When steam or air is in contact with the back of the glass the grooves have a prismatic effect and light striking the glass is reflected to the eye of the observer. When, however, water rises and fills up the grooves light passes right through and is absorbed by the back of the gauge. Thus the steam space appears

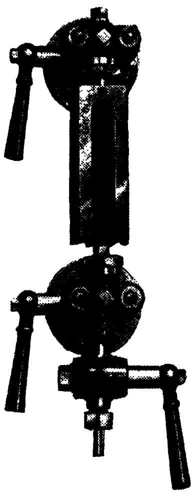


FIG. 3.
Klinger Reflex Level Gauge—
Type 'K.'

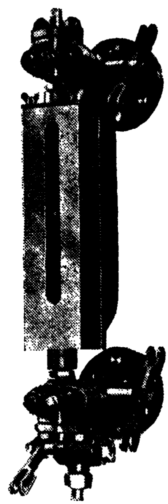


FIG. 4.
Klinger Double-Plate
Level Gauge.

bright and silvery while the water appears unmistakably black, and it is impossible for confusion to arise as to the level of water in the boiler even if it is allowed to pass out of sight in the glass.

Another difficulty encountered in modern high pressure steam plants is that of finding a glass suitable for water gauges which is capable of withstanding the corrosive action of the steam and water at such high pressures and temperatures. Contrary to what might be expected, the purer the feed water the greater is its solvent action on glass. In order to protect the glass it is, therefore, becoming usual practice to insert a thin strip of good quality mica against the face of the glass plate to keep it from contact with the steam and water.

Fig. 4 shows the latest type of mica-protected double plate gauge made by Richard Klinger Ltd. for steam pressures up to 1,200 lb. per sq. in. Two thick plates of specially toughened plate glass are bolted to either side of a steel body with the water space between them, each glass having a thin piece of mica plate against its inner face. These gauges are generally used with an illuminating device which shows up the water level as a bright spot of light. Klinger 'Sleeve-Packed' cocks are incorporated throughout, thus making a simple and reliable fitting for the highest pressures.

Regulators.

Copes Feed Water Regulator.

(Copes Regulators, Ltd., 9 Southampton Place, High Holborn, London, W.C.1.)

The Copes Feed Water Regulator consists of a thermostat, the actuating element, and a balanced valve controlling flow of the boiler feed water. A rigid strut connects the thermostat

TENSION TYPE.

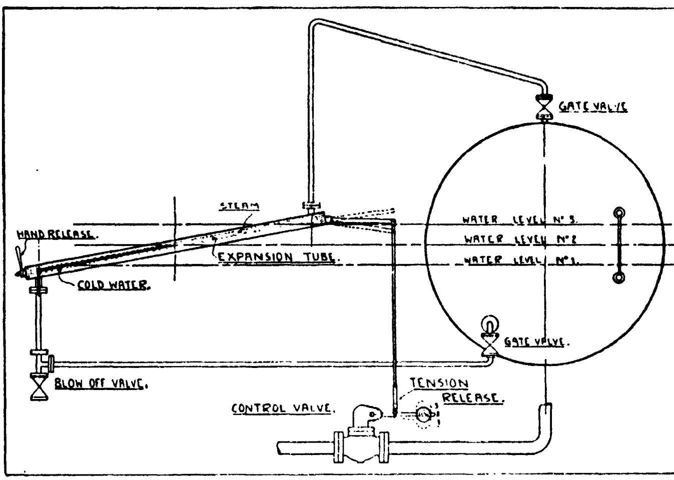


Fig. 1.

and valve. The thermostat consists of an expansion tube securely mounted between two steel channel pieces. It is so designed that the expansion tube is always in tension. The tube is welded to the flanged head and heel pieces.

The upper end of the expansion tube (see fig. 1) is connected to the steam space, and the lower end to the water space of the boiler. The same water level exists in the expansion tube as in the boiler; the upper end of the tube being filled with steam and the lower end with water. Movement of the water in either direction causes the expansion tube to act as a thermostatic relay, which actuates the feed water control valve.

The control valve is fully balanced, rugged and practically frictionless, the usual sliding stem moving in a stuffing-box being replaced by a horizontal rotating shaft.

Copes Flowmatic Regulator.

The equipment used in the Copes Flowmatic Regulator consists of the Copes patent Tension Type Thermostat, which instantaneously responds to water level changes; the Copes G.A. Regulating Valve with adjustable valve area and pressure compensation; and the Copes Flowmatic Controller.

The Flowmatic Controller is responsive to changes in steam flow, and participates in the movement of the feed regulating valve, with the action of the tension type thermostat. The

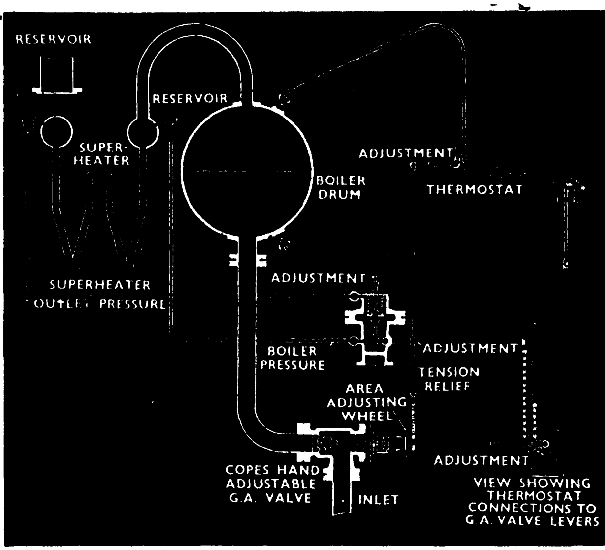


FIG. 2.—The Copes Flowmatic Controller.

rate of water flow to the boiler is thus correctly related to the steaming rate of the boiler in a manner which is impossible when water level control is used by itself.

Where automatic combustion control is used, the Copes Flowmatic Regulation is desirable.

THE COPES PUMP GOVERNOR.

The Copes Pump Governor is an automatically controlled balanced valve intended to be fixed in the steam supply main to the feed pump. The valve is operated by means of levers actuated by the difference of pressure between the boiler pressure and the feed-line pressure. The main steam header pressure (C) is carried to the inside of the sylphon bellows. The main water-header pressure (B) is carried to the outside of the bellows. The position of the weight on the lever arm determines how much the water pressure will be above the steam pressure. The amount of water that will pass through the control valve of the feed water regulator depends on the pressure drop across the valves as well as upon the area of the feed valve opening.

The governor maintains a fixed excess pressure between steam header and water header by varying the pump speed to correspond with the steam load on the boilers. As the steam pressure varies the water pressure should vary to correspond.

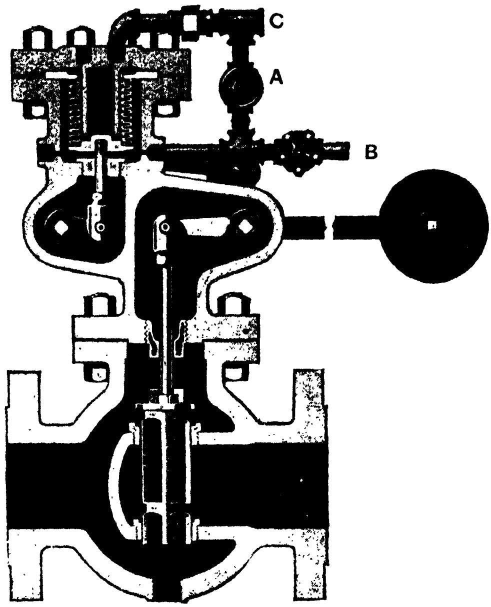


FIG. 3.—Copes Pump Governor.

CROSBY FEED WATER REGULATORS.

(*Crosby Valve & Engineering Co., Ltd., Crosby Works, Ealing Road, Wembley.*)

The action of this Regulator is in no way dependent upon the rising and falling of a float and there are very few parts which are liable to give trouble and so cause the apparatus to work in an erratic manner. This is an improved pattern Regulator and is furnished with an inclined power producer 'J,' shown in fig. 1, which will give continuous feed and is much simpler than the makers' original type of power producer.

The Regulator is very simple in design and there are no complicated moving parts to get out of order. In starting up the Regulator, it is only necessary to remove the air valve 'B' on the power producer and fill with water, it being so designed internally that there are no risks of air locks, etc. Once filled, the regulator will operate indefinitely without further attention.

The water level of the boiler is governed by the amount of water in the power producer. Thus it is possible to lower the water level of the boiler at any time whilst under load, or raise the control level. To lower the level it is only necessary to slack off the air valve nut 'B' and allow sufficient water to escape from the closed system to bring the boiler water to the desired point.

With this inclined Pattern Feed Water Regulator, as outlined above, the water level in the boiler is so maintained that a very reasonable line is given on the water flow meter chart, which is a great advantage where accurate records are kept. This is accomplished without even the use of an Excess Pressure or Differential Regulator on the inlet side of the boiler feed regulating

valve. If the excess pressure regulator 'S' is installed in conjunction with this type of feed regulator, and the makers strongly recommend that it should be to obtain the utmost efficiency and satisfactory control, then a uniform line can be maintained.

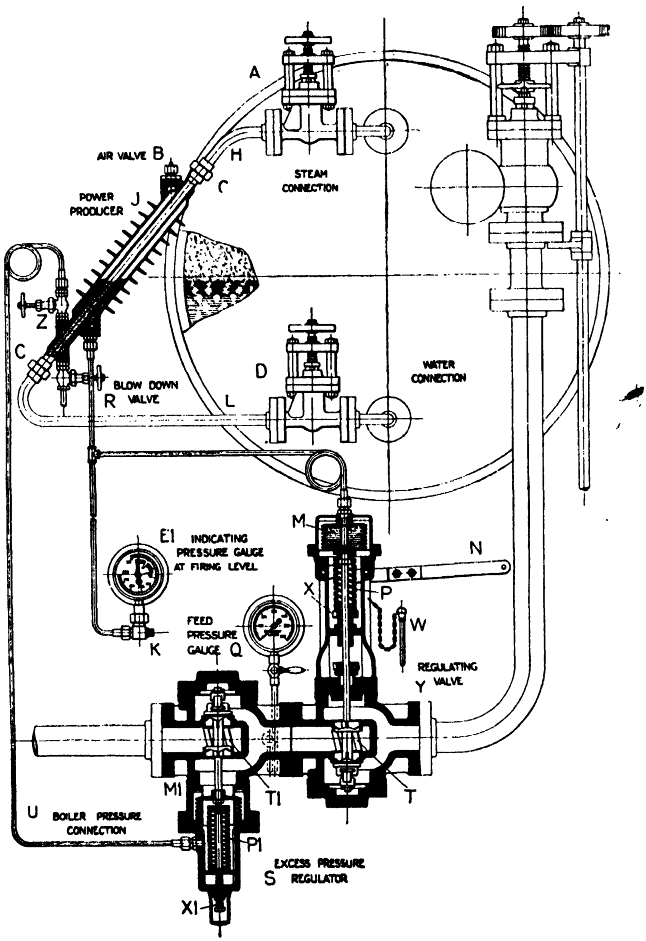


FIG. 1.

The regulating valve 'Y' is of a special double-seated semi-piston balanced type, having gradual opening ports giving accurate regulation of the water flow. This regulating valve can be equipped with a four-ply Sylphon seamless metallic bellows 'M,' as shown in the illustration, or a reinforced rubber diaphragm.

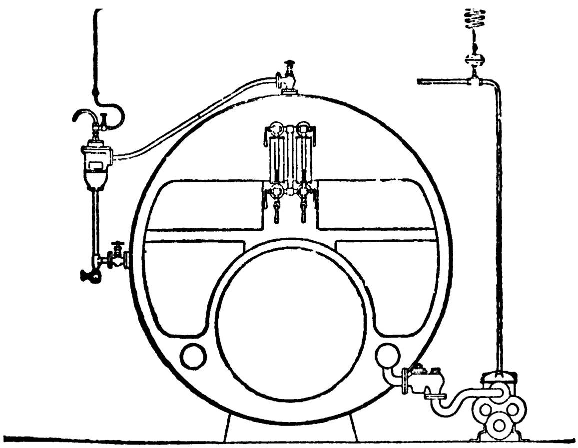
A feature of the Crosby Regulator is the indicating pressure gauge 'E1' fitted at the firing level, which gives a visible indication of the position of the regulating valve, whether 'shut' or 'open.' With the Crosby system of feed water regulation, the regulating valve can be in any position in respect to the power producer.

THE THERMOFEED REGULATOR.

(*Ronald Trist & Co., Ltd., Bath Road, Slough.*)

The 'Thermofeed' Regulator is designed to maintain an almost constant water level in a steam boiler. The variation is approximately ½ in. in Lancashire type boilers, and slightly more in vertical, economic and watertube boilers.

The 'Thermofeed' regulator consists of a float chamber containing a float-operated mechanism, which operates a double-seated valve. This valve controls the passage of steam to a 'Thermofeed' control valve or 'Thermofeed' automatic regulating valve, which is fitted either in the steam supply pipe leading to the boiler feed pump (as shown in the illustration), or in the feed line to the boiler, depending upon the lay-out of the installation.



A feature of the 'Thermofeed' Regulator is that the valves fitted in the feed line or the steam line to the pump are in the fully open position until they are closed by the operation of the 'Thermofeed' regulator.

Upon the 'Thermofeed' regulator being put out of commission these valves automatically resume the fully open position, permitting the feed supply to the boiler to be hand controlled.

From the illustration it will be seen that the float chamber is mounted externally to the boiler, and that the steam and water stop valves are provided to permit the examination of the equipment even with the boiler under steam.

THE WEIR 'ROBOT' BOILER FEED REGULATOR.

(*G. & J. Weir, Ltd., Cathcart, Glasgow, S.4.*)

This regulator is designed to maintain a steady, continuous feed at all rates of evaporation. It is very sensitive to small changes in water level, and contains no springs, diaphragms, or thermostats. The action is not affected by excessive pressure differences across the regulator, and as the single mitre valve gives a positive shut-off, it can be operated on boilers which are banked, hand regulation being unnecessary.

The action of this regulator (fig. 1) is as follows: When the float J falls, due to a fall in the boiler water level, the needle valve H rises, cuts off the flow of water through the opening

K into the chamber D, and the pressure in the chamber D falls, due to the leakage of water from the chamber through the clearance between the piston B and the cylinder C. Immediately the water pressure in the chamber D falls below the pressure for equilibrium, the valve A lifts and allows feed water to pass into the boiler. The valve A can rise only the amount that the needle H has been raised, because immediately the opening K is uncovered the pressure on top of the piston B rises above the pressure necessary to maintain the valve in equilibrium, and a closing movement of the valve A takes place until the pressure in the chamber D reaches the equilibrium value.

When the water level in the boiler rises, needle H is lowered and opening K is uncovered, thereby allowing pressure water to pass into chamber D and valve A falls to the same extent as the needle.

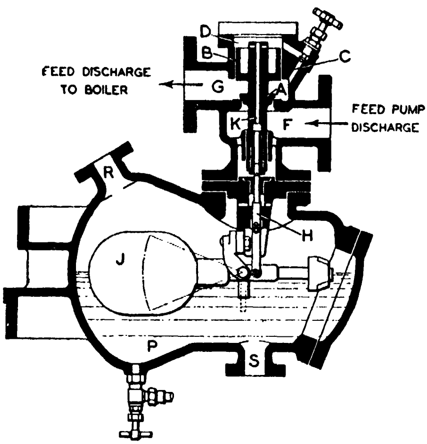


FIG. 1.—Weir 'Robot' Boiler-Feed Regulator.

It will be seen that the valve A and the piston B are constrained hydraulically to move in the same direction and to the same extent as the needle H, which is operated by the float; and for any given position of the needle H the valve A is maintained in equilibrium in a corresponding position.

The working water level in the boiler varies from a top level, when no steam is being generated in the boiler, to a bottom level when the boiler is operating at its maximum rate of evaporation, and the feed regulator maintains a steady water level between the top and bottom water levels for any given rate of evaporation between no load and the maximum load.

THE WEIR 'MUMFORD' FEED WATER REGULATOR.

This is another type of float-controlled automatic boiler feed regulator, also operated by the discharge pressure of the feed pump as in the 'Robot.' In the Weir 'Mumford' type, however, the float gear with needle valve is separate from the special check valve, connection between the two being made by means of a small copper pipe. The check valve is provided with an extension spindle with a piston, controlled by hydraulic pressure. The float rises or falls with the boiler water level, opening or closing the needle valve, and thus controlling the flow of water past the balance piston of the valve; when the needle valve closes, pressure is built up beneath the piston, and the valve opens, while when the needle valve opens, the pressure is released and the valve shuts.

FEED WATER HEATERS.

(G. & J. Weir, Ltd., Cathcart, Glasgow, S. 4.)

The location of feed heaters in the system determines the type of heater which can be used. Heaters placed on the suction side of the feed pump may be of the direct-contact type, but on the discharge side of the feed pump they must be of the surface type, and being subjected to the full discharge pressure of the pump must be constructed accordingly. In the Weir 'Multiflow' feed heaters the heating surface consists of solid drawn copper tubes of heavy gauge, securely expanded into cast iron headers of substantial section. The entire element of tubes is free to expand, being held at one end only, and the construction avoids the leakage troubles inseparable from the use of ferrules and packing exposed to the variations of temperature in apparatus of this class. Vertical and horizontal heaters for all duties are manufactured.

De-Superheaters and Safety Valves.

THE CROSBY DE-SUPERHEATER.

(Crosby Valve & Engineering Co., Ltd., Crosby Works, Ealing Road, Wembley.)

This arrangement is designed to reduce automatically the pressure and temperature of high superheated steam by spraying the steam with water.

The outfit can be used in conjunction with a steam pressure reducing valve where this is necessary, or as a straightforward de-superheater without pressure reduction.

The use of a de-superheater enables steam which has been generated in modern high temperature plants to be used in connection with the older prime movers, process plants, auxiliary engines, etc., designed for low temperature steam.

For reducing the pressure the 'Crosby Patented Relay Operated Single Seated Balanced Valve' is used. It is fitted between the high pressure steam main and de-superheater and will deal successfully with any quantity of steam from the minimum to the maximum without hunting. The device has a cast steel body with special heat resisting nickel alloy valve and seat and is furnished with a balancing piston and gradual opening 'V' ports. The valve is operated by a hydraulic relay through chain led over suitable guide pulleys and connected to the level of the balanced valve. The relay consists of a hydraulic cylinder mounted upon a diaphragm chamber which is in communication with the low pressure steam main. The load on the diaphragm is balanced by means of a weighted steelyard which is connected by a bridle to a small 'pilot valve' and any movement of the diaphragm, due to a slight variation above or below the pre-determined pressure in the controlled or low pressure main causes this pilot valve to admit water to the top or bottom of the piston as the case may be. The movement of the piston operates the balanced valve in the steam main controlling the reduced pressure within very close limits irrespective of the volume of steam passing or variations in the high pressure main. The low pressure steam now passes by means of a connecting pipe to the inlet of the de-superheater chamber, this piping and the de-superheater being protected from an excess pressure by Crosby Pop Safety Valves with forged steel nozzles and nickel alloy parts, the number depending upon the pressure, size of main, and degree of safety required.

For temperature reduction the de-superheater chamber consists of a mild steel vessel with electrically welded seams, generally provided with a single pressure loaded, self-cleaning and self-adjusting spray nozzle so designed that sufficient pressure is always maintained at the nozzle to provide an efficient and fine spray. To regulate the admission of spray water to this spray multiple diaphragm control valves are provided, set so that they come into operation in sequence and thus give a means of accurately controlling the quantity of water admitted to the spray. The operation of these valves is in turn controlled by means of a bi-metallic type thermostat fitted on the outlet side of the de-superheater through a suitable bend or tee piece in the steam main. This thermostat regulates the admission of water through a small nozzle thus varying the pressure in the connecting pipe between the thermostat and the diaphragm control valves, causing these to open or close and thus varying the quantity of water sprayed into the steam. The water supply for the spray must be at an excess pressure of about 40 lbs. over and above the low pressure de-superheated steam. This may be taken from the feed main through a strainer and reducing valve or in large plants a separate supply is sometimes provided. The water for operating the hydraulic relay of the reducing valve and the thermostat should preferably be taken from an overhead tank so that the valves will work under a steady static pressure. A relief valve should be provided between the hydraulic reducing valve and the diaphragm control valves to protect these against any excess pressure and a back pressure valve is fitted to prevent steam passing into the hydraulic main should the water supply be cut off for any reason.

NOZZLE TYPE SAFETY VALVES.

(Crosby Valve & Engineering Co., Ltd., Crosby Works, Ealing Road, Wembley.)

These valves, which are fully protected by various patents, are manufactured in many designs to suit a wide range of operating conditions. The valves have exceptionally large discharge capacities far exceeding those of an ordinary bevel seated type of valve.

The discharge capacities realised are guaranteed and are achieved through a patented secondary lift which gives a full lift to the valve. These valves give their capacities with an accumulation not exceeding 3 per cent. and the blowdown, which is adjustable, can be varied between 2 per cent. and 4 per cent., both figures comparing favourably with the Board of Trade allowance of 10 per cent. The valves open and close sharply without any simmering or 'warn.'

For high pressure service these valves incorporate a full length nozzle as illustrated in fig. 1, which is manufactured from a special alloy steel highly resistant to erosion and corrosion. This nozzle is correctly shaped to give maximum flow and is so designed that the high pressure only comes in contact with the inside of this forged steel nozzle, thus the body is protected from excessive strains or distortion. The disc is also manufactured from a special forged high nickel chrome steel.

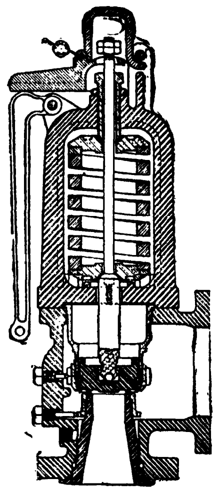


FIG. 1.

Crosby Nozzle Type Safety and Relief Valves can be furnished to suit pressures up to 2,000 lb. per sq. in. and temperatures of 1,000° F.

These valves are being used extensively for high pressure boiler plants and in oil refineries, etc. For oil service the valves can be supplied with totally enclosed spring bonnets, packed easing levers, etc., being perfectly tight on the discharge side.

Various modifications in the design of the easing lever, spring casing, etc., are available to suit individual requirements. Where necessary all the internal parts can be supplied of stainless steel to resist corrosion and for use on oils and oil vapours which would attack other metals.

THE KLINGER VALVE.

(*Richard Klinger Ltd., Klingerit Works, Sidcup, Kent.*)

The Klinger Piston Valve, fig. 1 (p. 1251), being seatless, eliminates all the troubles inherent with the ordinary seating valve. There is no seat to skim up, no valve to grind in and no gland to pack. The valve consists of a simple parallel piston sliding inside two resilient "Klingerit" compressed asbestos packing rings (marked A and A1 in illustration), which are separated by a ported lantern bush and held in place by the cover. Tightness of the valve is maintained by the pressure of the packing rings against the smooth surface of the piston.

When after prolonged use wear takes place it is corrected by tightening the cover nuts, thus compressing the rings further against the piston. When finally further compression is impossible the rings can be easily and quickly renewed at a very low cost, thus making the valve as good as new again.

An indicator on the handwheel shows clearly the position of the valve at any time and allows accurate control of the flow through the valve.

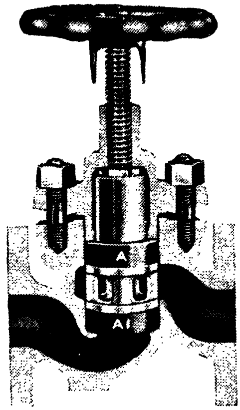


FIG. 1.—Section through Gunmetal flanged straight through "Klingerflow" Piston Valve.

These valves, it is claimed, show up to particular advantage when used under very strenuous conditions, such as continuous operation on throttling, as the parts on which their tightness depends are not directly subject to the scoring action of the passing fluid.

Klinger valves are made of various materials for use with saturated or superheated steam, hot oils and spirits, hydraulics, gases and chemicals, and are made in all sizes up to 10-in. bore. Special valves can be supplied for pressures as high as 1,500 lb. per sq. in.

'Newman-Milliken' Valves.

(Newman, Hender & Co., Ltd., Woodchester, Glos.)

The 'Newman-Milliken' glandless lubricated plug valve is of simple construction and possesses very few component parts. This valve is claimed to be free from the inherent defects of the taper plug cock and to have advantages over gate and globe valves. Primarily intended for low-temperature services not in excess of 365° F., it is suitable for handling a wide variety of products, including petroleum derivatives, acids, gases, water, compressed air and so forth at pressures varying from vacuum to 3,000 lb. per sq. in. It is made in sizes from ½ in. to 12 in., with screwed or flanged connections.

Referring to the accompanying sectional view (fig. 1, p. 1252), it can be seen that the design embodies a parallel plug without any form of packing, which has a shoulder on the upper portion, butting up against a corresponding shoulder machined in the head of the valve body. The plug is forced tightly against the body head by the pressure exerted by the spring within the bottom cover. Sealing is effected by means of a suitable plastic compound or lubricant, which is stated to ensure an entire absence of sticking. Different lubricants are employed for different services, but each of these lubricants is quite insoluble and is stated to have no deleterious effect on the liquid or gas passing through the valve. A reserve of lubricant is contained in the space below the lubricant screw and it is fed through a ball check valve, upper horizontal lubricant duct, vertical ducts, and a lower duct, thus lubricating the plug shoulder and working faces. The lubricant is maintained by this screw-feed arrangement at a pressure equal to, or greater than, that of the liquid or gas controlled by the valve. Should the valve show any tendency to leak it is only necessary to give a turn or two to the screw to stop it so doing.

When the valve is closed the pipe line pressure is prevented from escaping into the outlet port, for the pressure keeps the plug tightly in contact with the outlet side of the body, making a seal with the lubricant throughout its length. In addition, under these circumstances, the pressure moves the plug away from the inlet side through a distance of about half a thousandth of an inch, so that the pressure acts beneath the plug. As a consequence, the higher the line pressure the tighter the valve will hold, both in the port and at the head. It will be seen that the plug support spring only helps to keep the plug firmly against the body head and consequently to keep the

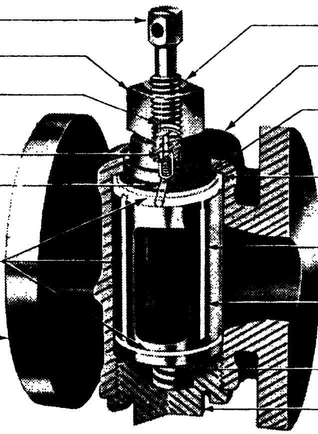


FIG. 1.

valve tight at this point. For high-pressure services the force exerted by the spring is small compared with the force produced by the line pressure, but on low pressure or vacuum services, of course, the opposite is the case.

A feature which is stated to be unique is the provision of a visual check on full lubrication, for when the valve is fully lubricated excess lubricant is forced out between the shoulder of the plug and body. Hence lubricant cannot be forced into the pipe line by excessive application of the lubricant screw. A further advantage of the design is that, since a parallel plug is employed, the plug cannot leave its seating (as happens in the case of a taper plug) when it is lifted to ease the turning movement; therefore, no space is created between the plug and the body for the inclusion of foreign matter, with the possible danger of subsequent scoring taking place.

This type of valve is manufactured in a number of patterns, with rectangular or round ports. Both rectangular and fully round ported valves provide a full pipe area opening, and the last-named are particularly suitable for all abrasive services, or where solids are present in the pipe line. Illustrated in fig. 2 is one of the standard cast iron valves, suitable for working pressures to 200 lb. per sq. in. and for any service in which cast-iron or steel will not be attacked by the liquid or gas. It is made with the body, plug, and bottom cover in cast iron of a fine close-grained structure. The lubricant set screw, ball, and ball seat are of mild steel and the plug support and ball check springs are of oil-tempered steel. These valves are also made in cast steel, gun-metal, acid-resisting bronze, Ni-resist, Aterite (for weak sulphuric acid), and various other metals. Geared operation can be fitted to all valves when so desired.

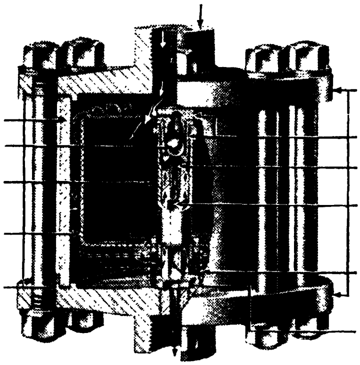
FIG. 2.

Newman-Carter Steam Traps.

FLOAT TYPE.

(Newman, Hender & Co., Ltd., Woodchester, Glos.)

HEAVY ENDS WITH FULL DEPTH TAPER THREADS FOR PIPE.



DESCRIPTION OF TRAP.

The body of the standard 'Newman-Carter' trap consists of a steel tube 'A' with massive cast-iron covers 'B' and 'C' at each end held together by an ample number of steel studs and nuts. The joint rings between these parts are very thin and made from graphited asbestos of high quality. For working pressures above 200 lb. per sq. in. the flanges are made either from steel castings or fabricated from steel plates.

All the working parts are attached to the bottom flange 'B' so that when cleaning or repairs are necessary the body tube and bottom flange can be removed and access gained to these parts without removing the complete trap from the pipe line.

The working parts of the trap consist of a brass hexagonal centre pillar or guide 'D' which is screwed into the bottom cover 'B,' the joint being made by means of a corrugated nickel alloy ring. The seating 'E' is of Monel metal which is pressed into a tapered hole of standard size in the top of the centre pillar 'D.' A brass float tube 'F' is attached to the brass float 'G' by means of nuts of the same material. The float 'G' is balanced by a stainless steel spring 'L' and so can be made of heavy gauge material, giving a great advantage over other steam traps where the float has to be made from material of a very thin section to function correctly.

The valve cage 'H' is attached to the upper part of the float tube 'F' into which it is screwed and locked in position by the lock-nut 'J.' The valve consists of a non-corrosive steel ball 'K' which is quite free to move about in the cage 'H,' this cage only being used to place the ball on the seat 'E' when closing and to lift the ball clear of the seat 'E' when opening to discharge. It should be particularly noted that it is the pressure only that keeps the ball valve 'K' closed and when in this position it is independent of any movement of the float. Furthermore it is only the very small weight of the ball which contacts with the seat and this contact is not accentuated by the weight of either float or levers as is the case in other traps of the Bucket or Inverted Float type. For this reason repairs to valve or seat in the 'Newman-Carter' trap are rarely, if ever, required.

OPERATION OF TRAP.

Steam and condensate enter the trap at the top which must be filled with water on installation. The condensate keeps the balanced float full and overflows into the trap body. The water gradually rising in the body lifts the float which in turn opens the ball valve and discharges the condensate at the bottom of the trap. As the level of the water in the trap body falls, the float drops and allows the valve to close before all the water in the body has been blown out.

The 'Newman-Carter' trap cannot blow steam unless the ball valve is held off its seat by dirt or other foreign matter. It is almost impossible for this to occur, as the ball is never lifted from its seat for more than a quarter of the diameter of the discharge orifice, so that anything that can get under the ball can pass through the orifice.

When the amount of condensate is high, the trap discharges continuously, as the ball valve is kept a certain distance off its seat according to the rate of condensation, but when the volume of condensate is low, the trap works intermittently. The ball valve closes tightly on its seat when there is no condensation and is held in position by the steam pressure.

ADVANTAGES OF TRAP.

1. The trap is simple and very robust in construction whilst relatively large discharge capacities are obtained without the complication of levers.
2. The ball valve is absolutely free and independent of any float movement. It is continually changing its position and finds a fresh seating surface every time the trap shuts off.
3. Wear on the valve seat is reduced to a minimum as this seating has only to support the free ball. The ball is so light that the contacting action which takes place does no damage to the seat or itself even after extended periods.
4. It works equally well on steam or compressed air lines.
5. Will function perfectly when a number of traps are connected in parallel.
6. There is only one type of 'Newman-Carter' trap and this is suitable for all service conditions.
7. It is of light weight.
8. As the inlet of the trap is at the top, it is unnecessary to fit air vent cocks.

'THE BRADFORD' STEAM TRAP.

(*United States Metallic Packing Co., Ltd., Soho Works, Allerton Road, Bradford.*)

'The Bradford' Steam Trap embodies a new principle in this class of device. It is a float trap, but with an important difference. The float does not open or close the discharge valve. It simply releases or engages a weighted lever, which is so arranged that the outlet is instantaneously opened or instantaneously closed. There is no middle position. This effectually prevents the wire-drawing of the discharge, which cuts the valve and seat and so destroys the efficiency of those traps where the opening and closing take place gradually.

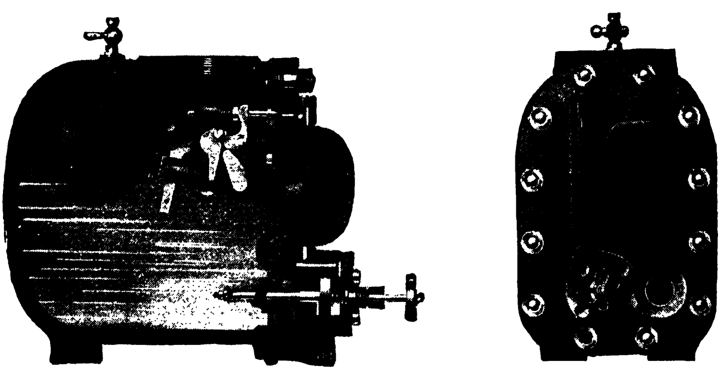


FIG. 1.—Piston Operated Trap.

The method of operation is as follows. The water accumulates in the collecting chamber and gradually raises the float until the latter reaches the desired point, when it trips the weighted lever, which immediately opens the discharge to its fullest extent. The water is then forced out under full pressure until it reaches the bottom level, when the lever is again tripped and the discharge is at once closed. The outlet is always submerged, so that there can be no escape of steam.

The apparatus is made in two classes, viz. the piston-operated trap and the direct-operated trap. Fig. 1 represents the piston-operated trap, in which the discharge valve is operated by the steam acting on a piston working in a cylinder. By this method the maximum diameter of outlet opening is obtained without reference to the steam pressure. This is a most important feature, as in consequence of the unusually large outlet the trap will deal equally well with flooding and priming as with ordinary condensation. In the direct-operated trap the action is simpler, the discharge being opened and closed directly from the trip lever, but the area of the outlet is smaller.

The traps will work at any pressure from 650 lbs. downwards, and are made in several sizes. The piston-operated traps will discharge from 1,000 to 6,000 gallons per hour when working at 100 lbs. pressure. In the direct-operated traps the discharge capacities range from 200 to 1,200 gallons per hour at 100 lbs. pressure. Both traps will force the discharge water 1 foot high for every pound of steam pressure.

An *eliminator*, which combines the above trap with a convenient form of separator, is also prepared, the manufacturers being The United States Metallic Packing Co., Ltd., of Soho Works, Allerton Road, Bradford.

Valve Refacers.

THE SIMPLEX PATENT VALVE RESEATING AND FACING MACHINE.

(*W. Crockatt & Sons, Ltd.*, 62 *Darnley Street, Glasgow, S.* 1.)

(1).

This tool, fig. 1, has been designed to avoid the necessity of using the studs in the valve chest as a means of steadying. At the same time it is simpler and stronger than the machines which are secured and centred by means of a chuck operated by a scroll. A very important feature is that the seat being faced is *visible* during the facing operation.

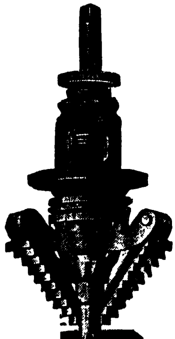


FIG. 1.

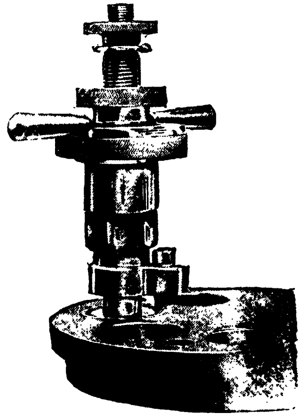


FIG. 2.

The tool is fixed in the valve chest (which may be either plain or screwed) by means of three stepped arms, pivoted at their upper ends. These arms are expanded by a cone which is depressed downwards by means of the hand nut. The action of expanding these arms both centres the tool in the valve and fixes it securely in position. This is ensured by the manufacture of the tool, the arms being turned up truly central with the spindle, after assembling. The cutters used are made of flat steel, and one end is arranged for flat and the other for bevelled seats. The simplicity of this type of cutter is found to counterbalance any slight advantage the milled type

of cutter may have. Also a very much smaller number of cutters can be employed to deal with a large range of valves, as each cutter is adjustable. In order to prevent chattering of the tool, the feed is against the action of a strong spring, which keeps an upward pressure on the tool, so that feeding only takes place when the cutter is forced down by the action of the nut. This prevents the cutter following any irregularities in the face of the seat.

A simple attachment is provided when desired for trueing up the valves themselves. The valves are centred in this part of the tool and faced up by a movable tool, which can deal with either flat or bevelled valves. The arms and lower part of the body of the tool are made of cast steel, so that the machine will stand hard wear. Ball-bearings are fitted throughout to ensure smooth working and to avoid jarring.

The machine is made in five standard sizes to deal with valves from ¾–12 ins. diameter, but has been supplied and can be easily adapted to do valves even larger.

(2)

In order to meet the demand for a tool which will face up the seats of the group valves in Weir's and similar feed pumps, the Simplex No. 3 machine has been introduced. This tool, fig. 2, is secured in position by the expansion of three segments, which are turned concentric with the central spindle, and which are expanded hard against the inside of the bore of the valve seat by the action of the tapered end of the spindle. It can easily be adapted to flat or bevelled seats by simply changing the cutter used.

If it is desired to face up the valves in the bottom seat of the pump without drawing the complete seat, a special tool with an extended body can be supplied.

Besides being suitable for the type of valve mentioned, the tool is equally adaptable to dead-weight safety valves, or, indeed, to any design of valve in which the inside of the bore is a plain cylindrical surface. Ball-bearings are fitted to ensure smooth and easy working.

Tube Cleaners.

THE CROCKATT SIMPLEX TURBINE TUBE CLEANER.

(W. Crockatt & Sons, Ltd., 62 Darnley Street, Glasgow, S. 1.)

This appliance (fig. 3) is made in sizes from 1¾ ins. upwards, and is driven by a turbine suitable for water at about 100 lbs. pressure. The cleaning head is detachable, so that special cutters or

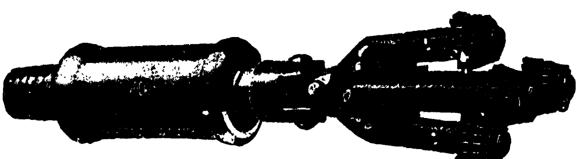


FIG. 3.

brushes may be attached to suit different conditions, and the universal joint enables the too to be used in curved tubes, such as in Stirling boilers. It is suitable also for Babcock tubes, economiser tubes, and any other kind of pipes having hard scale.

Salinometers.

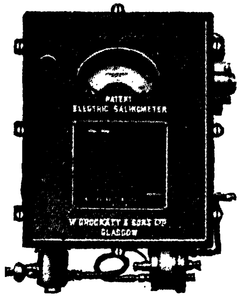
THE CROCKATT SIMPLEX PATENT ELECTRIC SALINOMETER.

(W. Crockatt & Sons, Ltd., 62 Darnley Street, Glasgow, S. 1.)

This is a direct reading instrument showing the amount of salt or other dissolved impurity in water, in grains per gallon, or other convenient measurement.

The illustration shows the standard Admiralty type, which is calibrated from 0 to 0·5 grains of chlorine per gallon and compensated for temperature variation of the water, between 60° and 120° F. This type has been fitted to cruisers, destroyers, etc., in the British and many foreign navies, as well as on liners, in power stations, etc.

Where a number of points are to be tested, the Multipoint pattern is fitted. In this, there is only one measuring instrument for any number of testing points, and it is mounted on a control board which also carries interlocked selector switches (one for each point), with corresponding



relays and warning lamps. When salt is present the warning lamp glows and the actual amount of salt present is noted by switching in the corresponding compensator box (which is very similar to the self-contained instrument illustrated, but has no instrument). The Multipoint type has the advantage that the readings of the various condensers, evaporators, etc., are obtainable on the one board, which can be mounted in a convenient central position. Also, where a large number of points have to be tested for water purity, it is cheaper than the corresponding number of self-contained instruments. A notable installation is one of 16 points on s.s. *Queen Mary*, and also her sister ship, *Queen Elizabeth*.

In addition to these instruments, which are used for testing the purity of the condensate, distilled water, etc., the same manufacturers make a Boiler Density Tester, which indicates the density of (or amount of dissolved matter in) the water in the boiler. This instrument is of great value as a protection especially for watertube boilers, where only impure feed is available. It is also an aid to economy, as it saves unnecessary blowing down of the boiler.

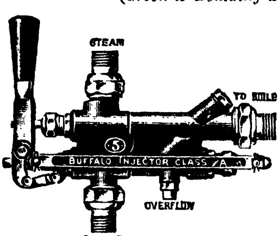
The latest modification is the Brine Density instrument which indicates the concentration of the brine in evaporators.

Special instruments have also been supplied for testing the concentration of sulphuric acid, during manufacture; testing for leakage of glycerine in Soap Works, and other industrial applications.

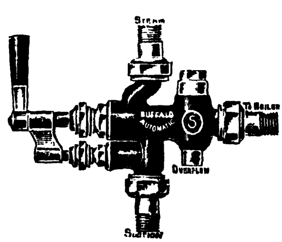
Injectors.

THE BUFFALO AND BISON INJECTORS.

(*Green & Boulding Ltd.*, 162*a* *Dalston Lane*, *London*, *E.* 8.)



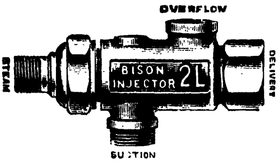
Class A.



Class B.

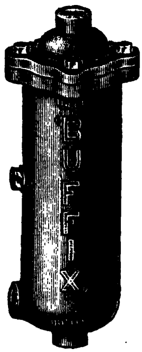
The Class A Buffalo Injector illustrated above is a double tube injector capable of taking either hot or cold feed water and is operated entirely by one handle. The lower tubes raise the water and supply it under pressure to the upper ones thereby enabling hot water (according to the steam pressure and other factors) to be taken. It has a positive shut-off overflow, thus eliminating dribbling, and the entrainment of air. The water supply adjustment is incorporated in the Injector.

The Class B Buffalo Injector is of the single tube type. It is a re-starting Injector and requires no vlave in the suction pipe for regulation.



The Bison Injector is particularly suitable for use with low steam pressures and may be fitted either submerged in the feed tank or with the feed water flowing to it.

Compressed Air or Steam Cleaners and Dryers.



THE "BUFFIX" SEPARATOR.

(Green & Boulding, Ltd., 162a Dalston Lane, London, E. 8.)

The Buffix Separator is a device for cleaning and drying compressed air or steam. The air or steam entering at the top has a centrifugal motion imparted to it, causing the water, oil and other impurities to be thrown out to the side of the container and so out through the drain. The clean, dry air or steam passes out at the bottom. It is a purely metallic type of cleaner and contains no felts, pads, etc., to become sodden; the pressure loss incurred is thus negligible but the degree of separation is extremely high. The Buffix Separator should be fitted as near to the point at which the air or steam is used as is possible so as to enable it to deal with condensation occurring during the passage through the pipe line.

Jointing Material.

(Richard Klinger Ltd., Klingerit Works, Sidcup, Kent.)

"Klingerit" Jointing was invented over sixty years ago and was the first compressed asbestos fibre jointing material to be made. Recent scientific developments have enabled its quality to be improved, and it is now a universal jointing material which can be used for superheated steam at the highest working pressures and temperatures, as well as for acids and other chemicals and all types of oils, spirits and other hydro-carbons.

It is made in thicknesses from ·008″ to ¼″ and can be supplied in sheets as large as 80″ × 240″ and is available also with graphited finish.

COMPRESSED ASBESTOS FIBRE JOINTING BONDED WITH SYNTHETIC RUBBER COMPOUND.

(Turner Brothers Asbestos Co., Ltd., Rochdale.)

Compressed asbestos fibre jointing made by Turner Brothers Asbestos Co., Ltd., asbestos manufacturers, of Rochdale, under their mark 'OAF,' is a sealing medium which for many years has held an important part in the functioning of steam plant and has helped in no small way to achieve the increased steam pressures prevailing in modern engineering practice.

Turner Brothers Asbestos Co., Ltd., make special qualities of 'OAF' incorporating ingredients resistant to acids and oils which fulfil a need for an effective seal and have already found a ready field in the chemical industry and in oil refinery work. Other specialised uses are covered by OAF' jointings made to Ministry of Aircraft Production, Admiralty, etc., specifications.

'OAF' Jointing is made from selected asbestos fibre and a special compound bonded together under pressure into a pliable and resilient sheet.

It is well known that there has been a serious curtailment in the supply of natural rubber and because of this it has been necessary to undertake the development of the use of synthetic rubber. It is, therefore, gratifying to be able to record that the Research Department of Turner Brothers Asbestos Co., Ltd., has successfully developed high grade 'OAF' jointings by the scientific blending of a percentage of synthetic rubber in the bonding compound.

'SEATRIST' RINGS.

(Ronald Trist & Co., Ltd., Bath Road, Slough.)

It is claimed that this automatic packing ring is correct in practice and theory. Fluid pressure reacts on a flexible tongue or lip : the higher the pressure the tighter the seal. Therefore maximum sealing efficiency with no following-up and a minimum of wear.

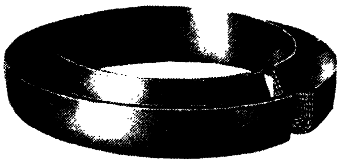


FIG. 1.

Fig. 1 shows the 'Seatrist' gland ring which is still fundamentally the same as when manufacture started more than 40 years ago. Note the tongue tapered to a wedge furnishing flexibility with strength and the reinforced body and heel on the outside, giving stability and robustness, permitting the gland assembly to stand up as a unit to the shocks and pressures inevitable to its functioning. One ring nests into the next behind from which it receives support without interference of its automatic operation. This nesting has another good result; rings are held snugly to the ram or rod when the pressure is off.

One of the useful features of the 'Seatrist' ring is that it can be fitted split without interfering with its automatic action. The rings can therefore be opened by careful bending to fit over the ram or rod without dismantling. When 'cut' rings, as they are called, are used, more than a single ring must be employed. The normal assembly shown in fig. 2 comprises three rings with support rings, soft headers for low and medium pressures, metal headers or a combination of metal and moulded support.

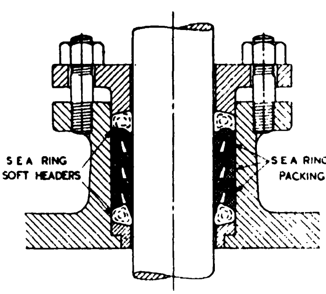


FIG. 2.

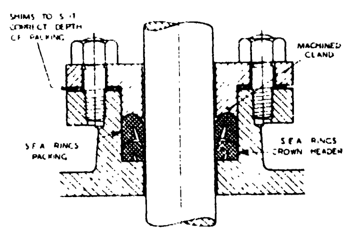


FIG. 3.

This last arrangement, termed a crown header, is very useful for shallow glands, with a single 'Seatrist' Ring which must be fitted uncut (fig. 3).

For use on pistons 'Seatrist' Rings have the flexible tongue outwards. For this reason they are called inverted 'Seatrist' Rings.

THE UNITED STATES METALLIC PACKING.

(*United States Metallic Packing Co., Ltd., Soho Works, Allerton Road, Bradford.*)

This packing is made in four distinct forms: 1, block type of packing; 2, cone or locomotive type of packing; 3, duplex type, consisting of combination of the block and cone types; 4, atmospheric-duplex type, specially designed for use on low-pressure condensing cylinders.

The United States Duplex Packing.

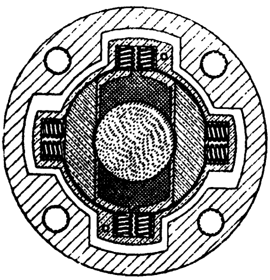


FIG. 1.

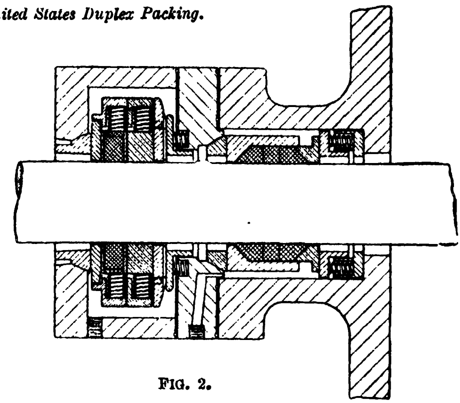


FIG. 2.

Figs. 1 and 2 show the duplex type of packing. The block portion consists of eight blocks which are held in strong rings, having pockets or horns holding springs. The packing blocks are put together in sections, four blocks to a section. Each section is, therefore, composed of two working blocks, Babbitt lined, and two guide blocks. The joints between the blocks in one section are at right angles to those in the other section, thus breaking joint. The blocks are regulated by springs, which merely keep the parts in place when steam is shut off, as when the steam is applied the steam pressure regulates and sets the packing. A ball-joint on one side and follower with springs on the other give the packing free play and keep it steam-tight, without regard to the vibration of the rod. It works equally well whether the rod vibrates or not, and, causing no friction, it does not wear the rod, but gives it a high polish. With efficient lubrication the blocks wear very slowly, many of the packings having given over twenty-four years' service without renewal of the blocks.

The cone portion of the duplex packing consists of three white metal or bronze rings placed in a floating cup, the interior of which is partly conical. This cup rests against ball and socket-ring, and the whole is kept in place by a follower with suitable springs, which allow the packing free play in all its movements. The whole is enclosed in a strong case, and bolted to the cylinder head with the usual stud bolts. The packing is steam setting, hence the friction is reduced at least 50 per cent. as compared with non-steam-setting packings. Drain valves are provided for removal of water, core sand, or other accumulations.

The block type of packing is used for pressures under 100 lb. per square inch. The duplex type is employed for pressures over 100 lb. Either packing can be arranged for working with steam superheated to 700° F.

Fig. 3 shows the cone or locomotive type of packing, which is practically similar to the cone packing described above. The white metal rings are the only parts which require renewal, and it is no uncommon thing for locomotives to run 80,000 to 100,000 miles without requiring adjustment of packing. All packings can be arranged to pass over enlarged ends.

Cone.

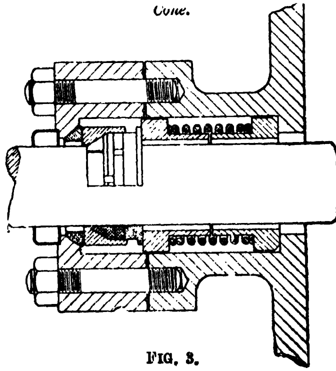


FIG. 3.

SECTION XXVII

PART V

Steam Condensers—Air Pumps—Circulating Screens—Cooling Towers.

THE WEIR 'REGENERATIVE' SURFACE CONDENSER.

(*G. & J. Weir, Ltd., Cathcart, Glasgow, S.4.*)

This condenser was introduced to eliminate the temperature loss which takes place between the temperature of the condensate and the vacuum temperature in normal type condensers, and also to give a condenser which delivers its condensate practically air or oxygen free. Consequently, although this condenser is suitable for any feed system, it is pre-eminently suitable for fitting to a closed circuit feed water system.

The design is the result of a series of researches carried out by G. & J. Weir, Ltd., and a typical example is shown in fig. 1.

It will be seen that a large lane is provided down the middle of the condenser, so that steam can penetrate right down to the bottom; this steam re-heats the condensate dropping from the lowest rows of tubes to practically steam temperature. Tests from actual plants show that the difference between the condensate temperature and the vacuum temperature does not exceed 2° F. This result is independent of the air leakage present. These tests also show that the oxygen contained in the condensate can reach as low as zero oxygen, 0·015 ml. per litre being quite a normal figure.

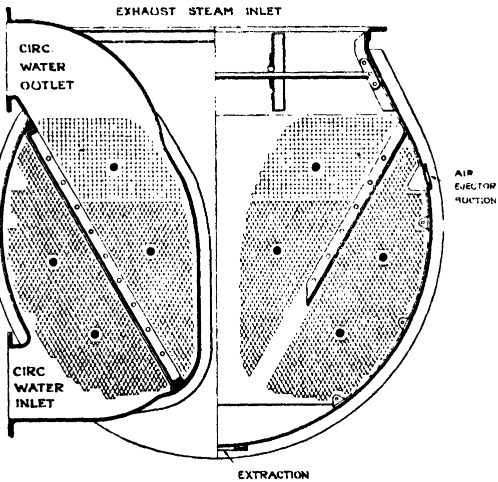


FIG. 1.

A hood plate is fitted on each side of the condenser, thus giving two reflex flows on the steam side; this arrangement permits a large surface to act as an air cooler if large air quantities are present, while if the latter are small the air cooling surface will be small. This action is automatic, the surface acting as an air cooler, automatically adjusting itself to the air quantity. This property results in the air being drawn off by the air pump at a minimum temperature for the conditions and ensures an enhanced condenser performance. The double arrangement shown lends itself to the twin water box design favoured by many present-day land engineers, the idea being that one-half can be shut down for cleaning or tube plugging while the other half is working, so that the main turbine need not be shut down.

The shape of the centre lane, together with the position and length of the hood plate, permits of a large entrance area being presented to the entering steam, with a gradually reducing area to the air suction, so that all the surface is equally effective and a minimum of pressure drop is attained.

THE WEIR ROTARY CONDENSATE EXTRACTION PUMP.

(G. & J. Weir, Ltd., Cathcart, Glasgow, S.4.)

The Weir Rotary Condensate Extraction Pump has been specially designed for use with closed feed systems, and is capable of operating under a particularly low suction head. It possesses stable characteristics, and is therefore entirely suitable for operating in parallel.

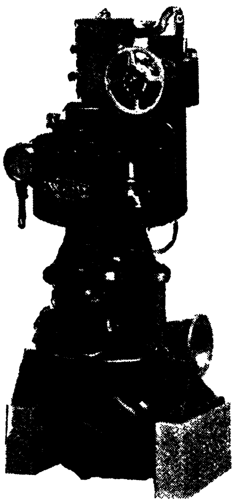


FIG. 2.

The vertical design of the pump permits the suction to be placed at the lowest point; this is important in ships having underhung condensers, where the head room available is small.

The Weir pump is of the vertical spindle, two-stage type, the lower impeller being the first stage and the upper impeller the second. The pump gland is arranged to be subjected to the full discharge pressure of the pump which effectively seals the gland and prevents any possibility of air entering the feed water.

The pump casing is divided down the centre line on a vertical plane, and suction and discharge branches are arranged on the rear casting. The pump may therefore be opened quickly and the shaft and impellers examined or removed without dismantling the remainder of the pump. The turbine is mounted on a flanged extension of the frame, and the drive is through a coupling which permits the power unit to be lifted clear for overhaul.

The turbine is of the impulse type, with one pressure and several velocity stages, and the unit runs at a relatively low speed.

A speed governor is provided, so adjusted as to maintain a reasonably constant speed between no load and full load. The usual emergency governor is also provided which closes the stop valve by means of trip gear when the speed of the pump reaches a predetermined limit. The turbine is fitted with roller bearings and ball thrust of substantial design.

The pump can also be arranged for turbine with geared drive or for electric motor drive as required. The impellers and armature are hung from an accessible thrust bearing at the top of the motor.

THE WEIR AIR EJECTOR.

(G. & J. Weir, Ltd., Cathcart, Glasgow, S.4.)

The three-stage ejector as developed by G. & J. Weir, Ltd., is illustrated in fig. 3 (p. 1263) and is used when the highest vacuum is required. On dry air tests these ejectors have attained vacua within 0·02 in. of the barometer; such extreme vacua, however, are seldom called for commercially, but this property enables them to reduce the partial air pressure in normal plants down to the very lowest limits. The saving in steam consumption of the three-stage type over the two-stage type is about 30 per cent. at 29 in. vacuum, and increases with higher vacuum. The three stages are generally similar in design, but are proportioned to suit the actual air, steam, and vapour volumes reigning at the respective stages. The steam nozzles are convergent, divergent, wherein the pressure energy of the steam is converted into kinetic energy; the steam issuing from the nozzles at high velocity (over 3,000 ft. per sec.) is delivered into a convergent, divergent diffuser nozzle; entrainment of the air and vapour takes place in the convergent portion of the diffuser, and the pressure rise takes place in the divergent portion; the combined mass of operating steam and entrained air and vapour pass to first intercooler, which is of the surface type and is arranged alongside and partly embraces the diffuser; the operating steam and vapour are condensed out in this cooler, while the air and its associated vapour pass on to the second stage, where the action is repeated, and so on to the third stage, the air being finally discharged to atmosphere.

The steam nozzles are of Monel metal to resist erosion, while the diffusers are of hard gun-metal.

The vertical design of intercoolers allows of the maximum air cooling and devaporising effect, so that the air is cooled and dried to a maximum for the conditions before it passes on to the next stage.

Between the discharge of the second stage and the inlet to the third stage a light plate non-return valve is fitted. This valve has the effect of stabilising the vacuum produced by the third

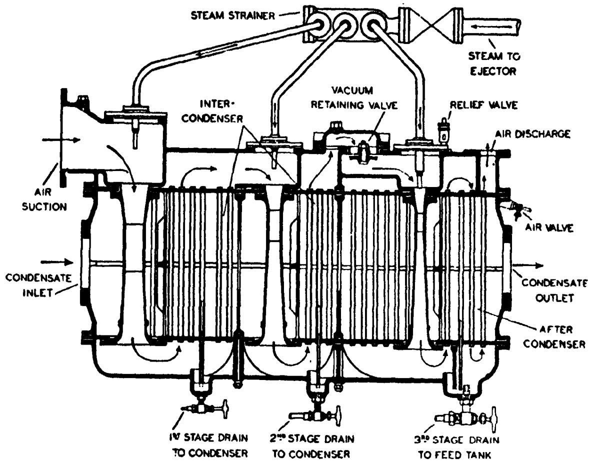


FIG. 3.

stage should the steam pressure fall below the designed figure. It also allows of the full vacuum being retained should the ejector steam valve require to be shut for any reason whatever; each intercooler is drained separately to the main condenser and the after-cooler to feed tank.

Circulating Water Screens.

(*F. W. Brackett & Co., Ltd., Colchester.*)

When water is taken from streams, rivers and tidal estuaries for cooling purposes to the condensers of power houses, it is necessary to remove from the water leaves, fish and other debris liable to choke the condenser tubes. For this purpose, mechanically operated self-cleaning screens are generally used, and the screen described here takes the form of two continuous chains carrying perforated panels through which the water flows. This screen band is revolved continuously so that the part which served to hold back the debris under water is brought up, automatically cleaned and returned to service.

The screening surface takes the form of a series of perforated and galvanised steel panels, which are attached to heavy type roller chains of either all steel, gunmetal or stainless steel construction to suit existing conditions. These chains are hung on sprocket wheels above ground level and dip down into a pit through which the water circulates, the chains and head-gear being housed in a fabricated steel framework sealed against the free passage of unscreened water. The water passes through the screen and is drawn off in a clean state, whilst the debris, which is caught on the outside of the screen, is elevated with the help of conveyor shelves to a point above ground level. The perforated panels above ground level are washed clean by a number of water jets, and the debris and wash water are diverted by a collecting hopper which discharges into a rubbish trough, which is itself led to a convenient point of outfall.

An important feature of this screen is that there are no working parts permanently under water.

The screening band is revolved by headgearing fixed in the frame above ground level, and is usually driven by an electric motor through a multispeed gearbox, so that the speed of revolution of the band may be related to the amount of debris in the water.

The type of screen just described is pre-eminently suitable for situations where the level of the water supply varies considerably, such as is the case with tidal waters, but for cases where the water level variation is inconsiderable the makers' 'cup' type screen may be preferable In this case the screening surface is arranged round the periphery of a large drum, the unscreened water flowing into the 'cup' and passing out through the mesh, otherwise the principle remains the same as for the 'band' screen.

'Cup' screens are now widely in use for screening sewage effluents and other factory wastes and also for waterworks, in which case fine mesh gauze is usually fitted. In the case of 'cup' screens, maintenance costs are claimed to be negligible, as there are practically no wearing parts.

PREMIER WATER COOLING TOWERS.

(The Premier Cooler & Engineering Co. Ltd., Shalford, near Guildford, Surrey.)

The Premier Cooler Co. manufactures all types of water cooling apparatus which depend upon the atmosphere for lowering temperatures by evaporation, by which means water may be cooled to within a few degrees of the prevailing wetbulb temperature, which is the theoretical limit of cooling by air.

These coolers are extensively used for re-cooling the water required by steam condensing plants, the jacket water of internal combustion engines, and many industrial purposes, and incidentally may be employed for removing gases from liquids and the evaporation of water from liquids containing solids which it is desired to concentrate as far as possible by this process.

The various types of water cooling apparatus in general use may be described as follows and the makers should be consulted as to the most suitable to meet any particular requirement.

Pond and Spray Cooling.—The simplest method of cooling is to discharge the warm water to an open pond of sufficient area to cool it to a low enough temperature by surface evaporation, which is very sluggish, making necessary a pond of large dimensions. The size of the pond may be greatly reduced by spraying the water from nozzles to obtain additional contact between water and air. These nozzles, which atomize the water, are carried on pipes placed a few feet above the surface of the water in the pond, but if an artificial pond has to be constructed the initial cost of this system may be greater than that of a cooling tower with its smaller tank. In high winds water loss from spray plants is heavy even when louvres are fitted round the edge of the pond, which interfere with the air flow on still days, and trouble may be experienced from clogging of the nozzles by foreign matter in the pond.

The cooling results from sprays cannot equal that of a tower occupying much less space and the pumping head may not differ materially in either system.

The Chimney Type Tower, which is almost invariably used for the larger schemes, is made in rectangular or circular form with a height ranging from 70 ft. upwards, designed to resist the full wind pressure experienced with a good factor of safety without the assistance of the internal filling or irrigation. These towers are generally of timber comstruction, but they can be supplied with steel, concrete, or brick chimneys. In the wooden construction all the bolts, fishplates, grappling irons, etc., are heavily galvanised.

The irrigation system is designed to give the largest area possible for the uprising air, which is brought into contact with the water or other liquid to be cooled or evaporated, in the finest possible form. The towers operate on the drop and film principle, which extensive experience has proved gives the best results. The distributing troughs and gutters are of the open top type, thus allowing the quick removal of solids and foreign matter which invariably accumulate, due to the continuous process of evaporation. The distributing tubes are of 'Venturi' type made of acid-resisting earthenware, as also are the splash plates. The water is thoroughly atomised before falling into the irrigation or lath system. The laths are fixed without the use of nails, being held in place in machined-notched bearers. The design provides for keeping the air in contact as long as possible, whilst allowing the maximum amount of air to be circulated through the plant.

The Open type construction, such as is usually employed for dealing with the jacket water for internal combustion engines, is constructed on somewhat similar lines, but the outside faces are louvred to prevent splashing out. Since this type of tower is not fitted with a chimney, the air circulation is much slower, and consequently the space occupied for a given duty is much more than with the natural draught Chimney type construction. In cases where the water volume is comparatively small, and it is desired to reduce the temperature to a very low degree, this type of cooler is especially suitable.

Forced draught.—Where the space available is limited, Forced Draught Towers can be supplied but they are uneconomical from the point of view of running cost, as compared with Natural Draught Plant as power must be provided to drive one or more fans, but it is the most flexible type as the air supply is positive and under control. Where space is very restricted or low re-cooled temperatures essential the forced draught tower is the only solution. It may be constructed of timber, steel, or concrete and for indoor installation 'Premier' steel cased 'Mechanical' coolers, or shell and tube heat exchangers are available.

Airblast coolers.—These are of the motor car radiator type fitted with gilled tubes through which the liquid to be cooled is circulated and over which air is forced by a fan, or fans. These can be used as air coolers by inversion.

The Premier Cooler Co. also manufactures air washers, dry cloth pocket and adhesic air filters and oil separators.

WATER AND OIL COOLERS.

(Visco Engineering Co., Ltd., Croydon.)

Water cooling plants utilising the principle of evaporative cooling are essential in many industries although the performance of such coolers is governed by the atmospheric wet bulb temperature which is the physical limit to which water can be cooled by this method. In practice, how-

ever, it is only commercially practicable to cool to within a few degrees of the wet bulb temperature.

The Visco Engineering Co., Ltd., of Croydon, has had many years experience of the design and construction of coolers of all types and sizes in many different parts of the world.

The most usual type of plant for dealing with large quantities of water as in power station work, is the natural draught chimney type cooling tower, in which the draught is caused by the temperature differential and the chimney effect of the cooler shell. In these towers, the outer shell or 'chimney' may be constructed of suitable timber braced and strengthened to resist any wind pressure likely to be brought to bear on it, or of ferro-concrete.

The lower part of the cooling tower contains the water distribution and cooling hurdles, and the upper part forms the chimney of sufficient height to create the necessary draught.

The hot water delivery pipe is connected to main distributing troughs from which the water flows into the distributing gutters placed at right angles. The gutters are so arranged that all receive an approximately equal quantity of water without swilling. The bottom of the gutters is fitted with glazed china nozzles through which the water falls in firm jets on the glazed china circular splash plates, which throw it in an upward and outward direction, forming an umbrella-shaped cone. The splash plates are so spaced that the whole area of cooling stack is covered equally by the falling spray.

The cooling hurdles and the water distribution are carried by a substantial framework independently of the tower itself. The posts of the framework rest on the concrete piers of the foundation. The cooling hurdles consist of substantial triangular laths cut diagonally out of square timber, and laid horizontally into notched bearers, which in turn, are carried by the posts of the framework. No nails are used for fixing the rails or bearers, as they would be liable to corrosion from the damp air.

For somewhat smaller installations an Open Type Natural Draught Cooler can be utilised to advantage.

To all intents and purposes the open type cooler consists of a chimney cooler without the chimney itself. This type of cooler is dependent on natural air currents, and is so designed that wind currents can pass straight through it. A properly designed cooler of this type will function satisfactorily with air currents as low as 3 miles per hour. Such towers are very inexpensive to maintain, and running costs are limited to the cost of pumping. They are of timber construction and may be mounted on a steel supporting structure or installed at ground level with a concrete sump. Open type cooling towers are provided externally with louvre boards which prevent loss of water due to windage.

Where cooling conditions are most stringent and space and site limitations preclude the installation of a natural draught cooler, forced draught is necessary. This is usually provided by fans either directly or indirectly driven by weather-proof electric motors. In cold weather, and when full load is not required, one or more of the fans can sometimes be stopped and the cooling obtained by natural draught with a corresponding saving in power.

Where space is plentiful, particularly where large ponds already exist, spray coolers can be used to advantage. These consist of a system of cast iron pipes projecting the water to be cooled through gun metal nozzles upwards to the atmosphere in finely divided form. When correctly dimensioned, they give efficient service, where the cooling conditions are not very stringent.

In cases in which it is necessary to cool small or medium quantities of water for Diesel engines, air compressors, degreasing plants, refrigerating plant condensors, etc., a mechanical cooler, such as the Visco 'Steelshell Forcedraft' cooling unit will be found very satisfactory. A cooling unit of this type is independent of natural air currents and can be installed indoors or outdoors. It is also so compact that it can often be placed in an odd corner which cannot be utilised for any other purpose. The installation of such a cooler will often save over 90 per cent. of water which would otherwise be wasted.

These steel shell coolers are sometimes of the induced draught type with the fans at the top. With this design, air inlets are at the base of the cooler shell on all four sides. Very good air distribution is thus obtained. Other advantages are that the cooler is perfectly symmetrical thus occupying less space, and the height of the water inlets is reduced with consequent saving in pumping costs.

Oil Coolers.—The Visco 'Sprayblast' Oil Cooler (Patent No. 531052) was designed to meet the needs of heat treatment plants for an efficient method of cooling oil for both batch quenching and continuous quenching.

The main principle of the design is the full use of the latent heat exchange caused by the evaporation of a small quantity of water in addition to the normal heat exchange between the air and the oil.

Cooling is effected by passing the oil through copper tubes on the outside of which cold water is sprayed, the cooling action of the spray being assisted by a continuous blast of air directed downwards over the cooling tubes.

As there is no contact between the oil and the cooling medium, the characteristics of the oil are not impaired by oxidation.

The amount of water lost by evaporation is relatively small, and amounts in practice to approximately 50 gallons per hour per ton of steel quenched per hour.

Coolers of this design are also being used successfully with oil cooled transformers and for other similar purposes.

SECTION XXVIII

Internal Combustion Engines.

THE BROTHERHOOD-RICARDO ENGINE.

(*Peter Brotherhood, Ltd., Peterborough.*)

The Brotherhood-Ricardo Engine (p. 1267) was first produced on a commercial basis in 1928 after many years' experimental work. It was the first high speed engine, using heavy oil as fuel, in which the inlet and exhaust valves of engines of the usual four-cycle type are replaced by a single sleeve surrounding the piston.

The operation of the sleeve is as follows :—

Air inlet and exhaust ports are formed on opposite sides of each cylinder and are opened and closed by corresponding ports in the sleeve. This sleeve is driven by a ball socket bearing on a rocker beam to which motion is imparted by a crank pin on a half speed shaft working on the centre bearing of the beam, the other end being coupled to a fulcrum link fixed to the crankcase.

This mechanism makes the sleeve move over a path approximately elliptic, with the minor axis vertical. In consequence, the change in velocity, and therefore the forces due to inertia are extremely small.

Another leading feature of the design is the system of fuel injection.

The rapid intermixing of the particles of the fuel with the air needed for quick and complete combustion is not, as in most engines, achieved by injecting the oil in a fine spray, but by relying on the turbulence of the air compressed in the cylinder. This state of violent turbulence is caused by tangential arrangement of the air inlet ports, and the acceleration of the initial swirl of the air as it is trapped in the combustion space. The fuel is injected where the speed of the air is greatest, *i.e.* at the side of the combustion space and complete mixing of air and fuel results.

In other respects such as forced lubrication, oil cooling, etc., the design follows modern high speed engine practice, except that the crankshaft is built up with case hardened and ground journals and floating bushes on the crankpin journals.

The engines are made with two standard sizes of cylinder, 44/50 B.H.P. at 750/900 r.p.m. and 63 B.H.P. at 800 r.p.m., so that combinations of from 4 to 8 cylinders cover all powers between 200 and 500 B.H.P.

The fuel consumption at full load ranges, according to the size and power of the engines from 0·36 to 0·42 lb. per B.H.P. hour.

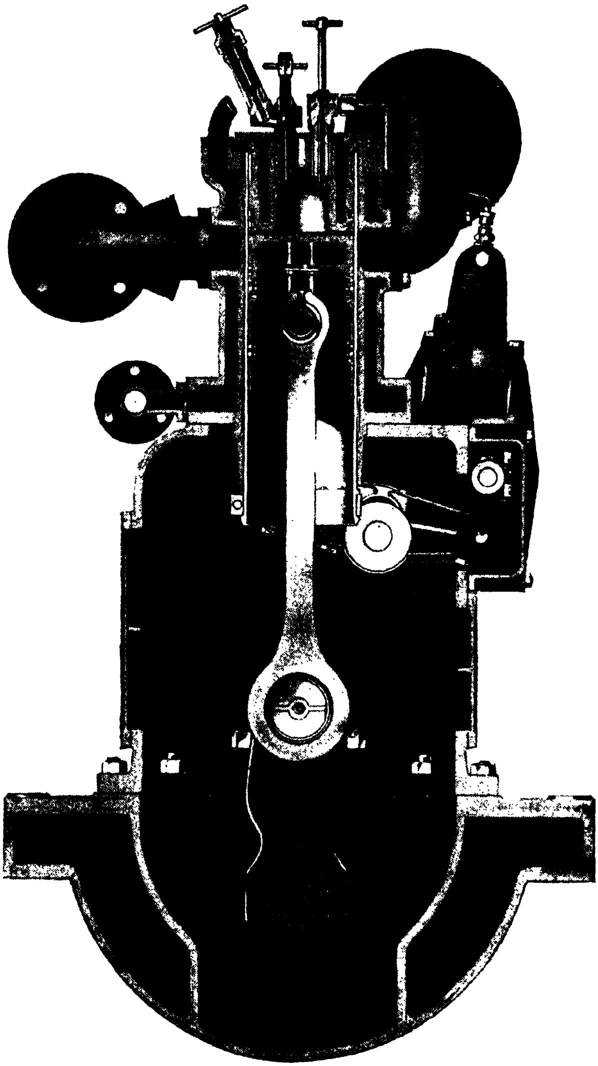
THE GLENIFFER MARINE DIESEL.

(*Gleniffer Engines, Ltd., Anniesland, Glasgow.*)

The Gleniffer heavy-oil engine is designed specifically for marine propulsion and is made in the D.C. series in units of 3, 4, 6, and 8 cylinders, developing 20 b.h.p. per cylinder continuously at 900 r.p.m. In the D.B. series it is made in units of 2, 3, 4 and 6 cylinders developing 12 b.h.p. per cylinder continuously at 1,000 r.p.m. As will be seen from the cross section of the D.C. engine (the D.B. being of somewhat similar design) the valves are placed horizontally facing each other in the cylinder head and are opened by the camshaft through levers and pull rods for the admission valves and direct through levers for the exhaust valves. The engine is of the solid injection type having the sprayer central over the top of the combustion chamber proper.

A projection on the top of the piston increases the turbulence at the top of the stroke by entering a restricted passage in the head. The makers guarantee a consumption of fuel oil not exceeding 0·44 lb. per b.h.p. per hour. Starting is effected by means of a compressed air motor which is mounted over the flywheel and engages through a self-disengaging pinion with a gear ring on the flywheel. In the case of the D.B. series the air motor is of the V-type and is carried in a cradle which can, if required, take a 6-in. electric starter. As soon as the engine is started the air motor automatically disengages.

The manufacturers claim that with their air motor up to thirty starts can be obtained from a 3½ cub. ft. air receiver without recharging, and this includes the first start from stone cold. Compressed air up to 300 lb. per sq. in. is obtained by employing one of the main engine cylinders as



Brotherhood-Ricardo Engine.

a pure air compressor, whilst its fuel pump is cut out of action. It is also claimed that these engines can be slowed down to well under 200 r.p.m. at which speed they will continue to operate in a most satisfactory manner, accepting ahead or astern gears without protest.

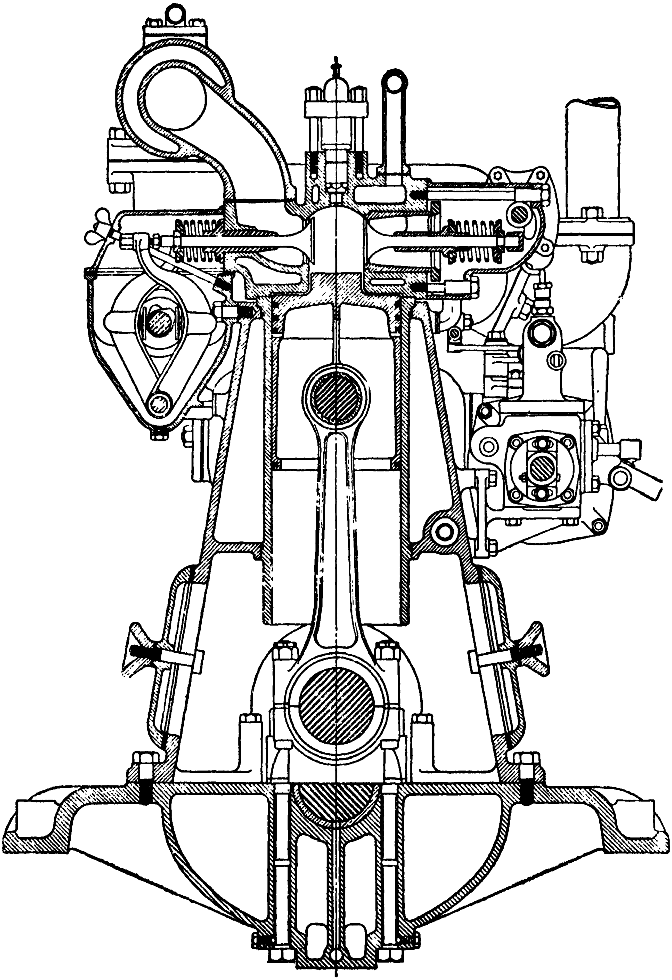


FIG. 1.—The Gleniffer Heavy-Oil Engine.

The lubrication system is of the force-fed type to the main and big end bearings and gudgeon pins, all the passages being drilled, with no internal pipes inside the crankcase. The force and scavenge pumps are of the valveless oscillating cylinder type and maintain a dry sump. They respectively suck from and deliver the oil to an oil reservoir fitted with a dip stick and large filler cap, and a small gauge shows the pressure to the bearings. The delivery pipe from the reservoir

passes through an annular passage in the water jacket, thereby ensuring sufficient cooling in the hottest of climates.

In the case of the D.B. engines the scavenge pump and the cooler are omitted, the pump being submerged in the sump.

Being designed wholly for marine propulsion all Gleniffer engines are equipped with full thermostatic control of the temperature of cylinder walls and heads. Without modifications of any kind they can be used in Arctic or tropical waters. Marine reverse gears, of the bevel epicyclic type, having a very large margin of safety and requiring no adjustment for several thousands of operating hours, are fitted to all models and specially developed marine governors are employed to ensure complete ease of operation and steady running under all conditions of speed, load and weather.

In addition to these engines the firm also manufacture a twelve-cylinder and a sixteen-cylinder V engine of 240 b.h.p. and 320 b.h.p. continuously at 900 r.p.m. These engines are a natural development of the D.C. series, and have many of the parts strictly interchangeable with this group of engines.

Engines of 16 cylinder 320 b.h.p. size have been built and supplied for patrol boat work, with superchargers driven from the engine units, and the output including the power absorbed for the superchargers is 420 b.h.p. at 1,000 r.p.m. The superchargers are so arranged that the engines can be run either as ordinary straight-forward units or by clutching in the superchargers to give the power stated above.

Bibby Couplings.

(*The Wellman Bibby Co., Ltd., Artillery House, Artillery Row, London, S.W.1.*)

As shown by the accompanying engraving (fig. 1), the Bibby coupling comprises two discs, each provided with teeth and coupled together by a continuous spring threaded between the teeth in a serpentine manner. The spaces between the teeth are flared, and the spring fits them

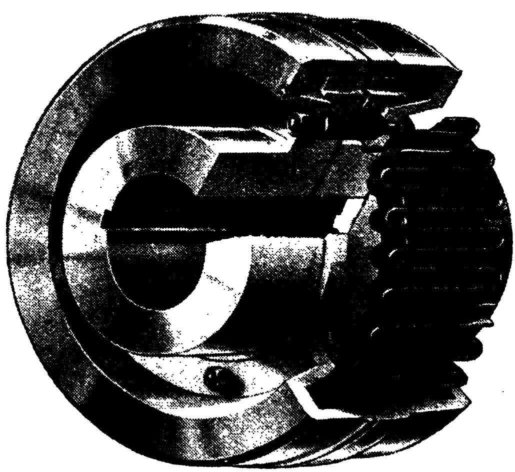
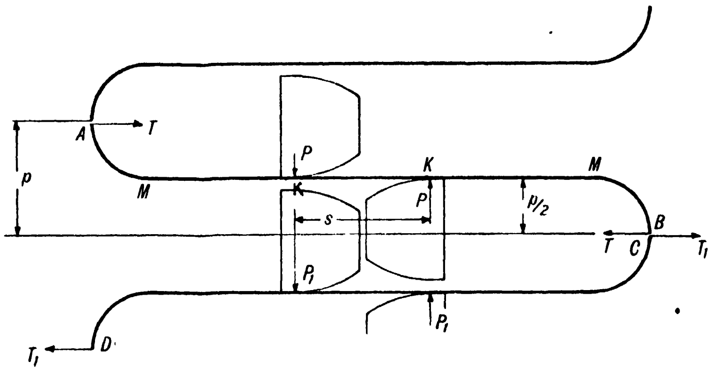


FIG. 1.

freely to allow of the desired movement. One of the discs is, of course, keyed to the driven and the other to the driving shaft.

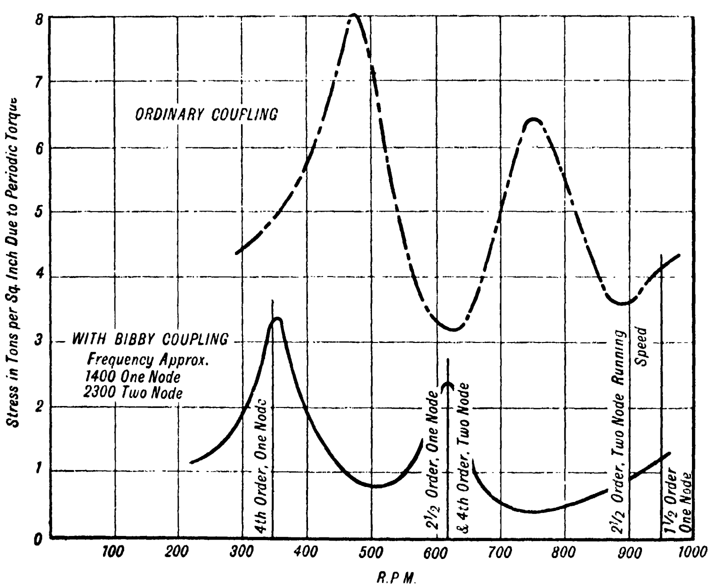
The Bibby coupling was the original coupling having flared teeth connected by plate springs. The separate plate system was discarded on account of its low resiliency efficiency, due to its acting as a double cantilever. The efficiency of any cantilever spring is only 33 per cent.

Now consider one element, AB in fig. 2 in the Bibby Continuous Grid Spring transmitting a load P from a driving to a driven tooth. The reactions from the teeth P and P form a left-hand



T
A
P
K
M
p
M
K
P
p/2
s
B
T C
T_1
P_1
P_1
T_1
D

FIG. 2.



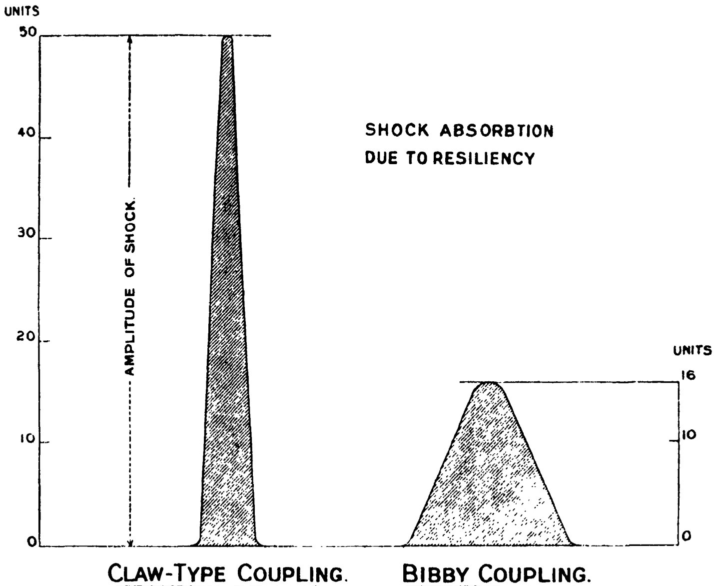
ORDINARY COUPLING
WITH BIBBY COUPLING
Frequency Approx.
1400 One Node
2300 Two Node
4th Order, One Node
2½ Order, One Node
& 4th Order, Two Node
2½ Order, Two Node
Running Speed
1½ Order One Node
Stress in Tons per Sq. Inch Due to Periodic Torque
R.P.M.
0
1
2
3
4
5
6
7
8
100
200
300
400
500
600
700
800
900
1000

FIG. 3.

couple, P × s, in the spring element. For equilibrium this must be balanced by a right-hand couple. The other external forces on the element are its connections at the middle of the bends, A and B, to the adjacent elements and consist of forces T and T forming a couple T × p, which is equal and opposite to P × s. The forces T on the element must act inward since the couple is right hand.

For the next element, CD, we have the same left-hand couple from the teeth loads, which are again balanced by a right-hand couple. For this element the bends are turned in the opposite directions, so that the forces T_1 and T_1 at C and D act outwards, giving the right-hand couple.

Where the two bends meet at BC there is an inward force T and an outward force T_1, which are equal and opposite and cancel each other. There are, therefore, no external forces on the spring except the useful normal ones at the teeth.



VIBROGRAPHS TAKEN BEFORE & AFTER FITTING BIBBY COUPLING.

UNDER EXACTLY SIMILAR DUTY.

FIG. 4.

Along the straight outside portions of the spring, KM for instance, the bending moment is Tp/2, and is constant. As the points of contact K and K between the spring and teeth approach each other, the length KM becomes longer, until, finally, almost the whole of the spring is under a constant bending moment and, therefore, has an efficiency of nearly 100 per cent. instead of the possible 33 per cent. with the cantilever types.

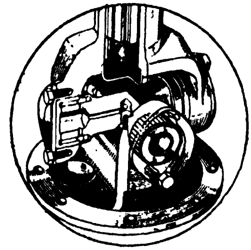
This property of high resiliency has a marked effect on every kind of drive. Fig. 3 shows a typical Diesel torque diagram, taken before and after fitting a Bibby coupling. The coupling stiffness is adjusted to take the criticals away from the running speed, and a special damping factor is introduced to detune the criticals. Fig. 4 shows the shock diagram taken from a 40,000 h.p. rolling mill.

Needle Bearing Propeller Shafts and Universal Joints.

(Hardy Spicer & Co., Ltd., Birch Road, Witton, Birmingham, 6.)

The original universal joint, or 'Hook's Coupling,' consisted of a cross and two yokes. Either yoke could be swung about the arms of the cross to which it was connected, and as the two arms lay at right angles it was possible to move the shafts in any direction.

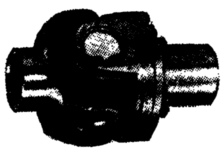
The latest Hardy Spicer needle bearing joint consists of two one-piece forged steel yokes and a single forged steel trunnion cross with the addition of four needle bearing assemblies. The needle bearings which fit into the yoke holes are self-contained, the needle rollers being assembled in a hardened steel retaining cup. In light duty types the needle bearing assembly is a light press fit, held in place by a lock spring, while in the heavy duty models the assembly is inserted in the yoke holes over the trunnion cross and is held by a retaining plate and cap screws.



Sectional Drawing showing Needle Bearings.

Hardy Spicer Needle Bearing Universal Joints are manufactured in two different types. The Standard, operating at angles up to 22°, is available in all sizes covering the entire field of light and heavy duty work. The Wide Angle operates at maximum angles from 30° to 35° and is designed for heavy duty service.

Standard Joints.



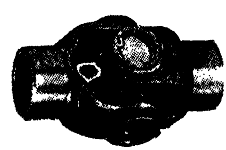
Standard Joint.

The simplicity of construction in accurately and securely retaining the needle bearings in proper position is shown in the above illustration.

The rigidity of the integral yoke lugs, which is not affected by bolt fastenings of any kind, reduces to a minimum objectionable torsional deflection, and the shaft assembly as a whole is right in weight and practically frictionless in operation; the tunnion cross on these models is provided with a large oil reservoir in each end.

Wide Angle Joints.

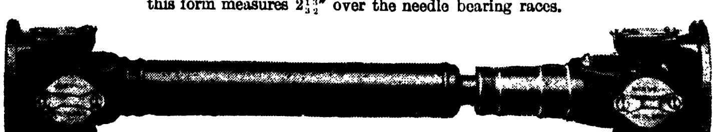
The wide angle universal joints have been designed to accommodate the extreme angular action required in many present-day specialised vehicles where dual axle drive (six-wheel construction) is used, and will give satisfactory and trouble-free operation at running angles from 30° to 35° each way. These wide angle joints are of the same construction and material as the standard needle bearing assembles with the exception that special yokes are incorporated in the design to provide the extreme angles required in an installation of this kind.



Wide Angle Joint.

Purposes.

Quite apart from automobile usage the Hardy Spicer universal joint has very wide and increasing engineering applications in the driving of machine tools, in accessory drives and power take-offs, in operating remote controls and generally in any situation where angular control or drive is needed. The smallest assembly available in this form measures 2$\frac{3}{32}$" over the needle bearing races.



Hardy Spicer Universal Joints and Propeller Shaft Assembly.

Water Coolers.

(*Heenan and Froude, Ltd., Worcester, England.*

The P type cooler manufactured by this firm has the advantage of being compact and occupying comparatively small space. The apparatus is self-contained, and has no moving parts except the fan which circulates the air, and the pump which circulates the water. The cooler is constructed entirely of metal.

The cooler consists of an outer casing, which is rectangular in shape, built up of thick mild steel plates. The base of this casing forms a tank into which the water drains after cooling, previous to re-circulation.

The water to be cooled, after passing through a strainer, enters at the top of the cooler and percolates through screens. In its fall it comes into intimate contact with a current of cold air which is forced upwardly through the casing by a high-efficiency axial flow fan, the discharge from which is introduced to the underside of the lowest screen.

The screens are formed of metal and provide a very large area for the interchange of heat; large access doors are provided through which the screens can be withdrawn for inspection.

The cooling effect is positive and continuous, the volume of air being a predetermined quantity which is not dependent on climatic conditions.

As the cooler makes use of the principle of evaporation it provides cooled water at a temperature which is frequently below that of the prevailing atmospheric dry bulb temperature; it is widely employed for re-cooling the jacket water of Diesel and other internal combustion engines, refrigerating condensers, air compressors, etc., and saves at least 95 per cent. of the water otherwise wasted. The power absorbed by the fan is usually only a very small proportion of the power of the engine with which the cooler is dealing, and it is often possible to supply the cooler with air drawn in from the engine room, thus assisting in the ventilation of the latter. The cooler can, in fact, be installed in the engine-room itself, thus simplifying the piping system and rendering the engine house almost independent of outside sources of water supply.

SECTION XXX

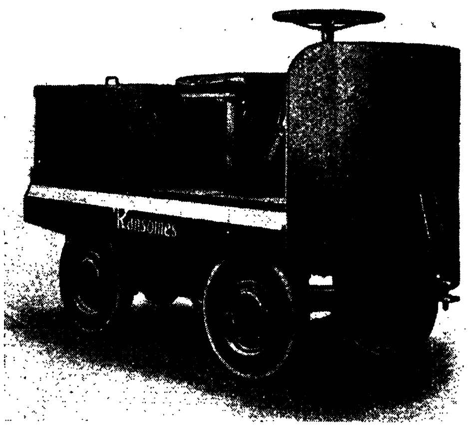
Automobiles.

Electric Industrial Trucks.

(*Ransomes, Sims & Jefferies, Ltd., Ipswich.*)

In present-day manufacturing conditions where efficiency in production is of the utmost importance, handling of goods and materials is a vital factor. In the majority of cases, the modern electric truck, by virtue of its low running and maintenance costs and length of life, constitutes the most economical goods-handling unit it is possible to employ.

It is no exaggeration to say that, as compared with hand trucking, an electric truck will in many cases treble the work performed at one-eighth the labour cost.



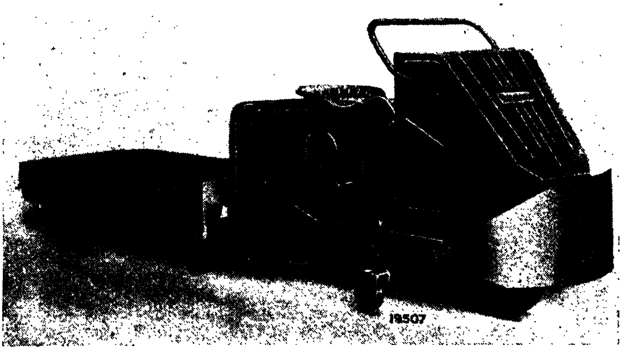
Electric Tractor, Type T.3 for hauling loads up to 6 tons.

Ransomes, Sims & Jefferies, Ltd., of Ipswich, were the first British makers of battery-driven electric trucks, and to-day they offer a wide variety of types. A model in great demand is the 2-ton truck with elevating platform. This truck may have either 2- or 4-wheel steering and carries its 2-ton load at a speed of 4–5 m.p.h. It is capable of ascending a 1 in 5 gradient. The drive is by means of double reduction spur and helical gear totally enclosed and running in oil, a 4-pinion differential is incorporated, and the shafts drive the wheels through universal ball joints. The drum type controller operated by foot-pedal gives four speeds forward and reverse and is interlocked so as to prevent simultaneous application of power and braking. The reverser is hand operated. The platform is elevated electrically by operating a lever conveniently placed for the driver's hand.

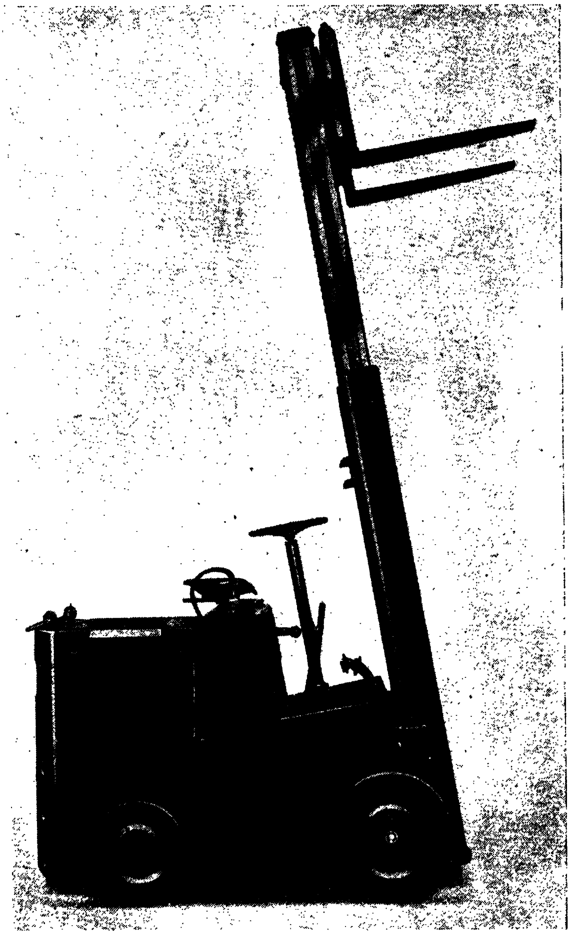
The 2-ton trucks can be supplied with high or low platforms and there is a similar model with a smaller platform particularly suitable for heavy loads and for operating in narrow gangways. The elevating trucks are, of course, intended for use with stillages which are loaded as required and left for the truck to collect when it comes along. The stillage is picked up in a few seconds and is deposited at its destination just as quickly. In some cases, however, it is preferred to place the load direct on the truck platform, and to meet such requirements similar trucks without the elevating gear can be supplied.

Ransomes' 1-ton capacity trucks are available in various models, with or without elevating gear, and with platforms high or low, wide or narrow, long or short. The driving unit is self-contained and mounted on a turntable, the drive being by double reduction roller chain from the motor to the driving wheel. Steering is effected by turning the whole of the driving unit by means of the control rail.

Electric Crane Truck—lifts 10 cwts.—transports 30 cwts.

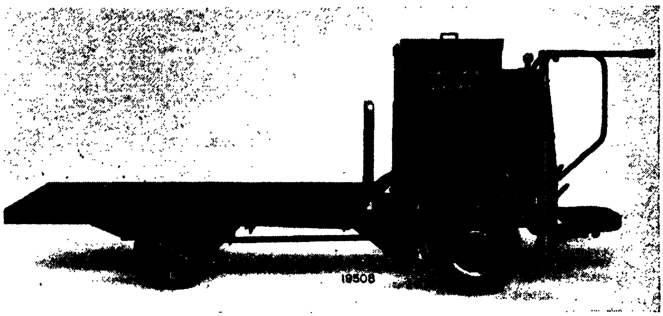


1-ton Electric Truck with low elevating platform.



' Fork-lift 20 ' Electric Fork Truck.

Ransomes' latest production is the Forklift 20, which is a battery-driven electric fork truck for loads up to 1 ton. The maximum height of lift is 10 ft. and the hoisting speed 22 ft. per minute. Special features are the compact design, small turning radius, and telescopic lifting gear which enables the truck to pass through doorways and girders, while still allowing for stacking between



2-ton Electric Truck with high elevating platform.

the girders. The mast tilts forward or backward, and the forks are adjustable laterally to suit different sizes of loads or pallets.

For heavier work there is the 'H' type truck which will deal with loads up to 4 tons, and in addition, the firm offers various other types including tipping trucks, crane trucks, and tiering trucks, among the latter being special models for handling loads of tin plate and other sheet metal. Other Ransomes battery-driven products include stackers, tractors, and run-about cranes.

Motor Lawn Mowers.

(Ransomes, Sims & Jefferies, Ltd., Ipswich.)

More than 100 years ago—in 1832—Ransomes made the first lawn mower (Budding's patent) while 70 years later they produced the first petrol-driven motor lawn mower. Today the range of Ransomes' motor lawn mowers covers every possible condition for which a power driven machine could be required.

The standard machines with air-cooled engines range from 14 ins. to 36 ins. in width, while for large flat areas, up to about 12 acres, there is the 40-in. combined motor mower and roller with water-cooled engine and with seat for the operator.

Exceptional care has been taken to ensure efficiency, simplicity, and economy in running. All engines conform to the most modern motor practice and are specially designed so that effective power is obtained at medium speed. Self-aligning ball bearings are fitted wherever advisable, and the maximum protection is given to transmission and driving mechanism. All controls in the air-cooled models are operated from the handles, while to facilitate turning the land rollers are made in sections. In the water-cooled machine the driver has complete control from the seat, and, moreover, can empty the grass-box without dismounting. In all sizes the cutting cylinder can be put out of gear and the machine used for rolling only.

A very interesting feature of the latest machines is the self-energising automatic switch, which makes these machines particularly easy to control.

To illustrate the saving in time which can be effected, and the low cost of running, it may be mentioned that the 30-in. model will cut up to an acre in an hour with a petrol consumption of less than 2 pints.

In addition to being first in the field with hand and petrol lawn mowers, Ransomes were the pioneers in this country of the electric lawn mower. In these machines an electric motor takes the current through a flexible cable from some convenient point, so that the mowers are particularly suitable for compact lawns where the supply is easily accessible. The motors can be supplied for alternating or direct current and are of Ransomes' own manufacture, specially designed for the purpose. A feature of these machines is that the absence of vibration ensures a particularly smooth even cut.

SECTION XXXII

PART II

Locomotives and Rolling Stock.

THE 'MELESCO' SUPERHEATER.

(The Superheater Company, Limited, London and Manchester.)

The headers or collectors of this apparatus are situated in the smoke-box and are all made of 'Melescoloy' a special semi-steel mixture, giving a tensile strength of not less than 28,000 lb. per sq. in.

The elements shown in fig. 1 are of double loop type, giving a four-fold steam path; they are made from solid cold-drawn steel tubing, and the three return bends on each element are formed by the 'MELESCO' Integral Machine Forged Return Bend Process, by which these bends are made integral with the tubing itself, and all electric and oxy-acetylene welding eliminated, thereby prolonging the life of the element and ensuring a clean, smooth finish both internally and externally.

Where elements are supplied having the ball joint attachment, fig. 2, the ball end is also made integral with the tube itself by upsetting the tube, thus avoiding any welding. It is afterwards turned and ground to gauge, having a diameter of 2¼ ins.

Where bolts are required for the joint, these are supplied in high tensile steel having a tensile strength of not less than 50 tons per sq. in. with a yield point of 35 tons per sq. in., so preventing the possibility of stretching when screwing up.

Other details, such as clamps and washers, are all made from drop forgings, which ensures sound material and maximum strength.

All elements are fitted with one or more steel bands and supports of robust construction, so arranged on the elements that the latter are properly positioned in the flue tube and at the same time reducing the obstruction to the flow of the gases to a minimum.

THROUGH BOLT HEADER, TYPE 'A.M.'

In this design of header, fig. 3, the casting is divided into transverse compartments or pockets communicating alternately with the saturated and superheated steam passages which are situated longitudinally at the back and front of the header respectively. Air passages which act as insulating spaces between these compartments are utilised for housing the element clamp bolts, the 'T' heads of which lie in a horizontal plane above the header, the nuts being below, or if desired the bolts may be used with their heads bearing upon the element clamp and the nuts on the top of the header. The ball joint attachment is universally fitted with this type of header, and is shown in fig. 3. As will be seen, any element can be easily withdrawn without removing the bolt.

'STIRLING' THROUGH BOLT HEADER, TYPE 'A.S.'

This header, fig. 4, has similar insulating passages between the compartments or pockets In this case, however, the saturated and superheated passages are arranged one above the other at the back of the header with communicating compartments or pockets at right angles. The slots or insulating passages, in addition to allowing for independent expansion of the compartment walls, are utilised for housing the clamp bolts with nuts either on top of the pockets or below the clamps as desired. It will be observed that to withdraw the element it is merely necessary to slacken the nuts, after which the bolt will slide out of the insulating space as the element is withdrawn, without having to remove the bolt from the clamp.

THE 'MELESCO' SUPERHEATER.

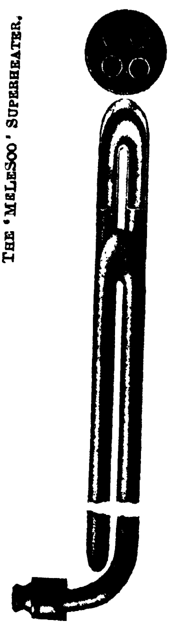


FIG. 1.—'MELESCO' Ball Joint Element.

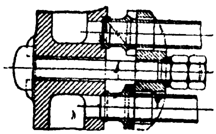


FIG. 2.—'MELESCO' Ball Joint.

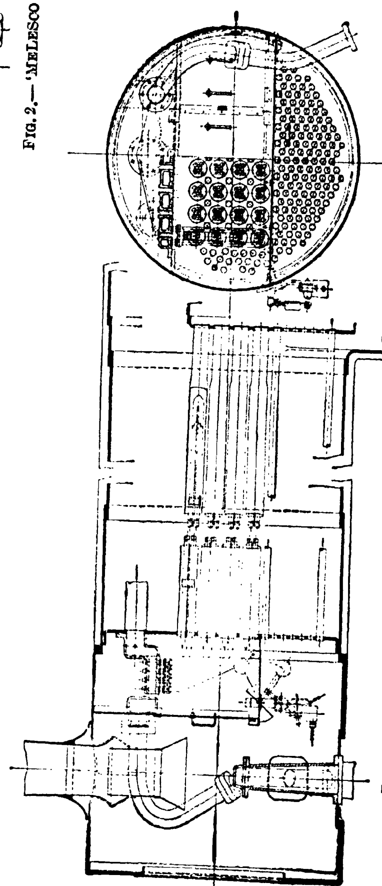


FIG. 3—'MELESCO' Through Bolt Herder, Type 'AM.,' with Ball Joint Attachment.

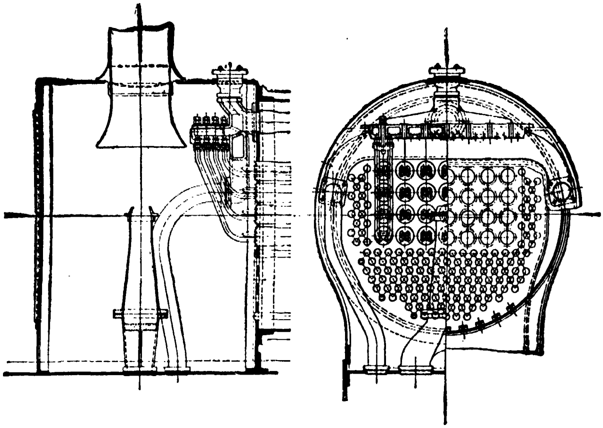


FIG. 4.—'Stirling' Through Bolt Header, Type 'A.S.'

MULTIPLE VALVE REGULATOR HEADER, TYPE 'AXV.'

With the advent of higher steam pressures and larger boilers, consequent on the increased size of locomotives, the combined superheater header with Multiple Valve Regulator as shown in fig. 5 has become a necessity.

This apparatus consists of Through Bolt Type Header, with which is cast integrally a chamber for housing the regulator valves and cam shaft.

The valves, which are of comparatively small diameter and therefore not liable to distortion, are operated by this cam shaft and closing affected by positive mechanical action of the cams.

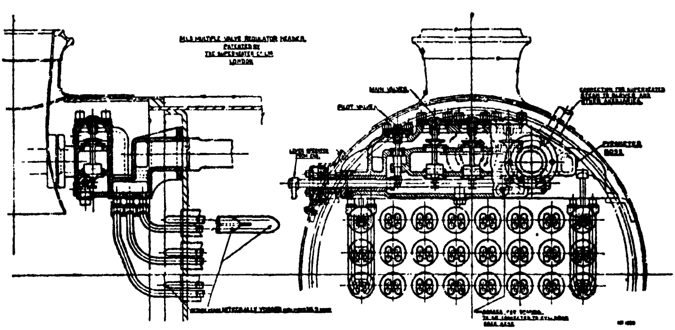


FIG. 5.—Multiple Valve Regulator Header, Type 'AXV.'

The valves are so nearly balanced that only a minimum amount of effort is needed to move them. This balancing is accomplished by providing a small pilot valve which opens first admitting steam to the balancing chamber preliminary to opening the main valves. The main valves are designed to open so that the passage of steam through the superheater and regulator is uniform providing a perfect graduation in the supply of steam to the locomotive cylinders.

In other respects the apparatus is similar to the usual designs and Ball Joint Elements are fitted. Large numbers of this type of header are now in service on railways throughout the world.

UNITED KINGDOM METALLIC PACKING.

(*Locomotive Packing cast-iron type.*)

(*The United Kingdom Self-Adjusting Anti-Friction Metallic Packing Syndicate, Ltd., Fazakerley, Liverpool,* 9.)

This type of packing has been specially designed to meet the requirements of consulting engineers and locomotive builders who are now favouring cast-iron blocks for packing to work against the present-day high degree of superheat.

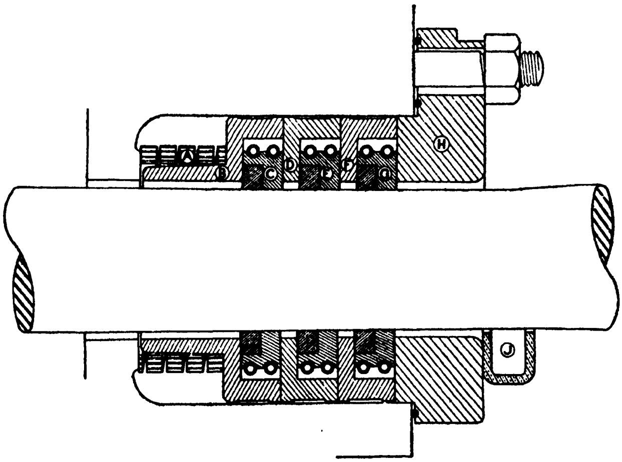


FIG. 1.

Its simplicity will be obvious from an examination of the illustration, fig. 1.

The covers B, D, F, containing the wearing blocks C, E, G, are held in position by a powerfu spring A, which adjusts the packing and tends to take up the wear on the walls of the covers (see fig. 1). When wear does take place, the covers themselves require only a slight adjustment by refacing, and can be replaced at a very low cost as against the entire replacement of the expensive housing made in halves, a prominent feature in the old German design of cast-iron packing.

The withdrawal of the packing is greatly simplified as, on the release of the gland H, the spring forces the packing from the stuffing box.

These packings are made from the highest quality of cast iron obtainable, and the best workmanship is used in their manufacture.

Lubrication arrangements can be provided as part of the gland, or separately.

One of the leading British Railways to which this packing has been supplied gave the official mileage covering a period of 5 years without renewals at 289,000. On dismantling the engine for general repairs, the internal covers B, D, F, and bearing rings C, E, G, were found to be in good condition. After regrinding the steam faces of the covers, together with the necessary adjustment to the cuts in the bearing rings, the complete packing was put back for further service and has since done well over 100,000 miles.

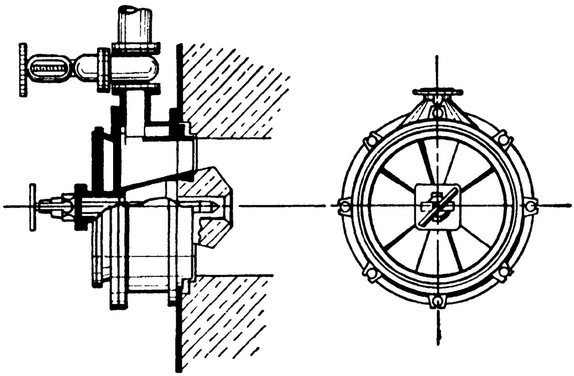
SECTION XXXIV

PART I

Heating.

THE 'GAKO' GAS BURNER.

(*Liptak Furnace Arches, Ltd.*, 68 *Victoria Street, London, S.W.* 1.)



The principle of the 'Gako' gas burner is the introduction of the gas and air in separate streams, to which is imparted a whirling motion. The resultant intensive mixing produces practically complete combustion. There are no moving parts, and the capacity range is wide without fear of back-firing. The gas consumption is claimed to be very low and the efficiency high.

The illustration shows a standard oil burner inserted through the centre for auxiliary firing.

SECTION XXXIV

PART II

Ventilation—Air Filtering.

VISCO AIR FILTER.

(*The Visco Engineering Co., Ltd., Stafford Road, Croydon.*)

The Visco Air Filter (devised in 1916) is used in conjunction with turbo alternators, pipe ventilated motors, and general ventilating plants. It works on the principle that the air volume is divided into a large number of thin streams which are passed through a layer of specially prepared filter bodies, the surfaces placed irregularly to each other, so that the direction of the air flow is changed suddenly and a great number of times. The filter bodies are thin copper-plated steel rings $\frac{7}{16}$ in. diameter and $\frac{7}{16}$ in. long, coated with a thin film of Viscinol, which is a specially prepared blend of highly viscous mineral oils. When the direction of the air flow is changed, the dust particles continue in the original direction and are thrown against and retained by the Viscinol-coated surfaces.

The filter is standardised in units 20 by 20 ins. square and 3 ins. deep, consisting of frame and cell, which latter is simply clipped on the frame. The capacity per unit is 600/700 c.f.m. Pressure loss is practically steady at 0·25 in. water gauge. Average consumption of oil, 1 gallon per unit per annum. Cleaning is done by rinsing in hot soda water on the average every eight weeks, depending on the original dust content and length of daily run. There are practically no other running costs.

The filter units are assembled in a convenient shape to make up any capacity, and are preferably built into the alternator foundations (thus taking up no valuable space) or as self-contained units in any position.

The *advantages* claimed are: No moving or wearing parts, all metal, fireproof, foolproof, small space.

The Visco Duplex Filter has two cells in series, the primary of standard construction and the secondary filled with smaller size filter rings. This type has an extremely high efficiency and is for ordinary purposes, coated with Viscinol, but for hospitals and food factories, with Stericidol, a stable copper-glycerine solution for not only filtering, but at the same time removing and destroying bacteria.

In the Visco Patent Self-Cleaning Air Filter sinuous metal plates are used as the filtering medium. These are contained in rectangular cells which are arranged in vertical columns. By suitable gearing the bottom cell is lowered with a swinging motion into a bath of oil in such a way that a vigorous rinsing action takes place. At the same time, another cell from the previous operation, in draining position behind the column, is moved upwards to the top of the column. Where the dust loading is exceptionally heavy, instead of using an oil bath, sprays can be fitted which subject the cell to pressure oil spraying from both top and side which removes all the accumulated dust.

The Visco 'O.E.' Filter with 'throw-away' filter element is designed for use when a filter of **the non-viscous type is required.**

Dust Collection.

'VISCO-BETH' DUST COLLECTORS.

(The Visco Engineering Co., Ltd., Stafford Road, Croydon.)

This fully automatic Dust Collector is used for collection of almost any kind of material in powder or fibrous form, such as flour, coal, cement, lime, tobacco, flue ashes, sugar, soap, milk sulphur, ores, enamels, chemicals, metallic dusts and metallic fumes, in connection with pulverising, drying, calcining, hydrating, conveying and other plants.

With the 'Visco-Beth,' which operates under suction, the air or gas is drawn by main fan through a number of cloth tubes, on which the dust entrained in the air is retained whilst the cleaned air (to exhauster) is discharged to atmosphere.

The tubes are made of wool, cotton or other material suitable for particular working conditions. They are arranged in compartments of 9, 12, 15 or more with, usually, a diameter of 6 ins. or 8 ins. and a length of from 7 ft. to 10 ft.

The unit comprises a number of compartments, each of which is, in rotation, isolated by the mechanical closing of independent air valve. During this period the tubes are automatically shaken a number of times and a current of clean air (scavenging) is permitted to flow through the filtering media in an opposite direction to that taken by the normal air stream. The scavenging air is, as a rule, controlled by separate fan, and pre-heated in cases where the dust-carrying air is highly charged with moisture, thus preventing condensation in the casing and wetting of cloth.

All valves and gearing are mounted on a top plate and operated by separate motor or from the shaft of the main or scavenging fan. Below the filter casing is a hopper for collecting the recovered material with conveying spiral to discharge the dust through a star type rotary valve.

This Dust Collector is entirely automatic in action, and the regular cleaning keeps the filtering efficiency constant at over 99 per cent.

Modified constructions are available for high vacuum installations, such as pneumatic conveying of grain or coal, and also industrial vacuum cleaners for boiler houses, flue ash removal, and a variety of other purposes.

HOWDEN CENTICELL DUST COLLECTOR.

(James Howden and Co., Ltd., 195 Scotland Street, Glasgow, C.5.)

Application.—The Howden Centicell Collector is a centrifugal type dust collector for high efficiency extraction of fine dusts which are air or gas-borne. The design is varied according to the type of dust to be extracted, and to the conditions of service. For example, applied to steam raising plant, small cells are used for the extraction of dust from the flue gases of pulverised coal-fired boilers, while larger cells are used in the case of stoker fired boilers. Performance and size of plant can be varied also, depending on the design of inlet nozzle ring. The collectors are made with the cells either in the vertical or in the inclined position, and overall dimensions can be varied to suit most layouts.

Description.—The Howden Centicell Collector, a sectional view of which is shown in fig. 1, consists of a large number of small high efficiency dust separating cells. All the cells in one collector are similar, and each is composed of three units: (*a*) the cell body; (*b*) the inlet nozzle ring; (*c*) the clean gas or outlet tube. The dust-laden gas enters the inlet nozzle ring of each cell, and in passing through the nozzle ring it is given a spiral whirl which causes the dust particles to move radially outwards to the inside wall of the cell body. The separated dust discharges from the body through the bottom of the cell, and the clean gas moves into the outlet tube and flows away. The gas inlet chamber of the collector is common to all the nozzle rings, and the gas outlet chamber is common to all the outlet tubes.

The dust outlets from the individual cells discharge into common dust hoppers. These hoppers are divided up to form separate compartments which are sealed off from one another in order to reduce gas flow in the hopper between adjacent cells. This inter-cell flow tends to cause entrainment of the very fine dust particles already separated, and is accordingly detrimental to high efficiency operation.

Inter-cell flow is further counteracted by the pressure equalising system which creates a slight depression in the hopper compartments, this slight depression being sufficient to overcome gas flow from one cell to another. The gas flow through the pressure equalising system is about 4 per cent. of the main gas flow. This system comprises a fan with a small dust collector situated before it, and ducting connecting the small collector to the various compartments in the main hopper. The small dust collector is of similar design to the main collector and has its own dust hopper, the discharge point of which is brought close to the main dust discharge point. The clean gas from the pressure equalising system is led back into the outlet duct from the main collector.

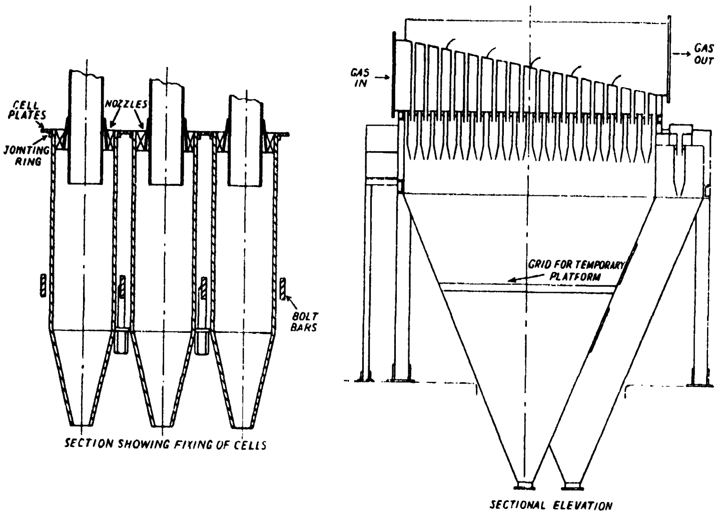


FIG. 1.

Interference between the gas whirls of individual cells, where these whirls still persist unprotected in the hopper compartments, is reduced to a minimum by a system of confluent whirls, thus ensuring high efficiency operation. With this system the gas whirls in adjacent cells are arranged in opposite directions of rotation so that, where the whirls persist unprotected, the gas motion in each whirl is in the same direction at adjacent points.

Performance.—The performance varies according to the size of the dust particles and the density of the dust presented to the collector. For example, when operating on flue dust from pulverised fuel-fired boilers efficiencies in excess of 90 per cent. are obtained, and on stoker flue dust efficiency lies between 95 per cent. and 98 per cent. Typical gradings for such dusts are as follows :—

Pulverised Fuel		Stoker	
Grade, microns.	Per cent. in Grade.	Grade, microns.	Per cent. in Grade.
Above 40	26	Above 421	22
40–30	14	421–296	12
30–20	14	296–211	9
20–15	10	211–152	8
15–10	14	152–104	10
10– 0	22	104– 76	14
		76– 63	4
		63– 43	3
		Below 43	18
	100		100

Movement of Air by Fans.

The 'Wing' Fan.

(F. W. Potter & Soar, Ltd., Phipp St., Great Eastern Street, London, E.C. 2.)

The shape, curvature and angle of the blades have been carefully determined to give the highest possible efficiency. Although it has been found that an angle of 27½° to 30° is the best for combined output and efficiency, the mounting at the hubs allows for the adjustment of the blades to any required angle in order to fulfil special duties.

The Wing fan is made in all sizes and can be arranged for belt drive or direct coupling to motor.

When considering the application of propeller fans it is important to specify the direction of the air current in relation to the direction of running. The illustrations, termed Set A, B, C and D, show clearly the four cases to be considered.

Economy of Power.—Where conditions permit of the choice between a propeller type and a centrifugal type of fan, the following rules are useful :—

(*a*) For open ventilation use propeller fans.

(*b*) Make main air duct same diameter as fan, or larger, avoiding acute or sharp angle bends

(*c*) When small ducts and sharp bends are unavoidable use centrifugal fans.

(*d*) Use large fans at as low a speed as will overcome whatever resistance is in the way of free passage of air.

EXAMPLE.—A 10 ft. Wing fan at 90 revs. per min. delivers 71,000 cu. ft. of air per min., absorbing 1·2 h.p.

A 5 ft. Wing fan at 720 revs. per min. delivers 71,000 cu. ft. of air per min., absorbing 19·2 h.p. Volume for volume, therefore, the smaller fan uses 16 times the power absorbed by the larger fan.

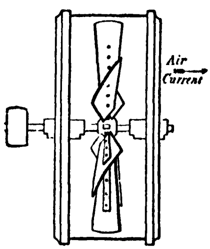
(*e*) Use two fans of the same diameter at low speed instead of one at double the speed.

EXAMPLE.—A 48 in. Wing fan at 600 revs. per min. delivers 30,200 cu. ft. of air per min., absorbing 3·6 h.p.

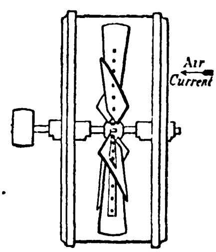
Two 48in. Wing fans at 300 revs. per min. deliver 30,200 cu. ft. of air per min., absorbing ·9 h.p. together.

Volume for volume, therefore, one fan uses four times the power absorbed by two fans.

SET A.

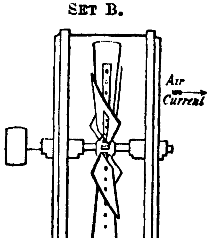


SET C.

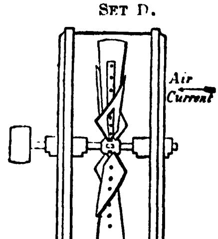


Viewing these fans from the pulley side, they both revolve in the same direction as the hands of a clock move, but the air currents are in opposite directions, as indicated by arrows.

SET B.



SET D.



Viewing these fans from the pulley side, they both revolve in a direction contrary to the motion of the hands of a clock, but the air currents are in opposite directions, as indicated by arrows.

Approximate Capacities of 'Wing' Fans (Smaller Sizes).

Diam. of Fan Blades Ins.	Slow Speed.			For ordinary Ventilation.			For Factories and Engine Rooms.			Fast Speeds.		
	Revs. per Min.	Net H.P.	Cubic Ft. per Min.	Revs. per Min.	Net H.P.	Cubic Ft. per Min.	Revs. per Min.	Net H.P.	Cubic Ft. per Min.	Revs. per Min.	Net H.P.	Cubic Ft. per Min.
14	680	·012	850	1,090	·05	1,360	1,540	·13	1,930	2,000	·29	2,500
18	530	·02	1,410	850	·08	2,250	1,200	·22	3,170	1,600	·51	4,250
24	398	·034	2,510	640	·14	4,050	900	·40	5,680	1,200	·92	7,550
30	318	·052	3,900	510	·21	6,260	720	·63	8,850	960	1·50	11,800
36	265	·074	5,620	425	·30	9,000	600	·87	12,700	800	2·05	17,000
42	228	·10	7,680	364	·43	12,250	515	1·20	17,400	685	2·75	23,100
48	200	·14	10,100	318	·55	16,100	450	1·55	22,600	600	3·60	30,200
54	177	·17	12,700	283	·68	20,800	400	1·95	28,700	535	4·70	38,300
60	145	·19	14,300	255	·85	25,000	360	2·4	35,400	480	6·00	47,200

SECTION XXXV

Refrigerating Machinery.

WATER COOLERS.

(Heenan and Froude, Limited, Worcester, England.)

The P type cooler manufactured by this firm and described separately on p. 1273, is particularly useful for employment with refrigerating machinery on account of its capacity for providing very cool water even in hot weather. Owing to its operating on the principle of evaporation, its capacity is controlled much more by the prevailing atmospheric wet bulb temperature than the dry bulb temperature, and consequently by suitably proportioning it to its duty, the water can be cooled down to a temperature closely approaching the wet bulb temperature.

Thus in hot weather, in both temperate and tropical climates, cooled water can be provided considerably below the dry bulb temperature of the surrounding atmosphere ; and as the efficiency and power absorption of refrigerating machines depend to some considerable extent upon their condenser temperatures, this is a most useful feature of the cooler. The apparatus is constructed entirely of metal and is self-contained, and the space taken up is very small in relation to its duty.

The refrigerating plant is rendered almost independent of town's water, and the cooler effects a large saving in water costs.

SECTION XXXVI

Compressors.

High Pressure Compressors.

(Peter Brotherhood, Ltd., Peterborough.)

Over sixty years ago the first Brotherhood high pressure compressors were made for supplying the air for the propulsion of the newly invented Whitehead torpedo, the pressure being 750 lb. per sq. in. This was the start of the development of the extensive range of high pressure compressors which has continued through the intervening years, so that there are, it is claimed, practically no conditions of pressure and capacity for which there is not a Brotherhood compressor.

Standard types cover all pressure up to 1,500 atmospheres and capacities up to the equivalent of 4,000 B.H.P. In one chemical works alone there are installed twelve high pressure compressors working on a system requiring a pressure of 3,500 lb. per sq. in. and driven through gearing by Brotherhood steam turbines developing between 1,500 and 3,500 B.H.P.

With the exception of the special type of compressors for oxygen, all machines are fitted with a special type of packing for the pistons which is claimed to ensure absolute tightness over long periods of continuous running.

A few typical machines are illustrated. Fig. 1 shows a two-stage double acting compressor specially designed for auxiliary service in motor ships, the pressure being between 350 and 450 lb. per sq. in. Fig. 2 shows a three-stage single crank compressor of capacity 400 cub. ft. of free air per minute at 1,000 lb. per sq. in. Compressors of this type are largely used for compression and liquefaction of CO_2, coal gas storage, auxiliary service in motor ships with air blast injection, and many other purposes. Two crank three-stage air compressors of more recent design for delivery pressures up to 600 lb. per sq. in. are available for supplying high pressure air for starting Diesel engines.

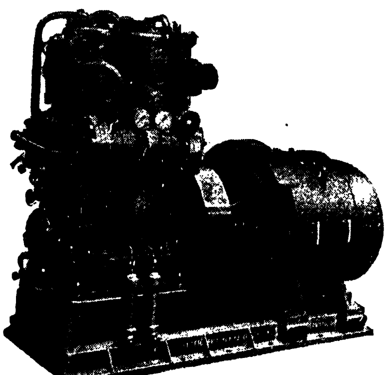


Fig. 1.
Two-stage Double Acting Compressor.

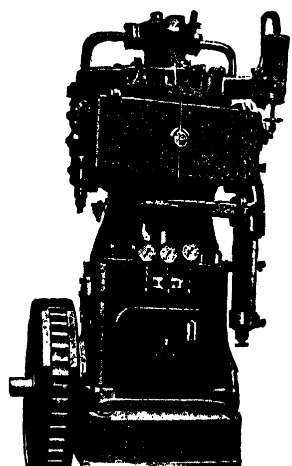


Fig. 2.
Three-stage Single Crank Compressor.

In fig. 3 is illustrated a five-stage three-crank gas compressor having an output of 400 cub. ft. of free air per minute at 5,000 lb. per sq. in. The largest high pressure compressors made in Great Britain are of generally similar arrangement.

Fig. 4 shows a four-stage air compressor for torpedo service in a submarine, the working pressure being 3,500 lb. per sq. in.

The Brotherhood compressors are also made for low pressures in one or two stages for all capacities up to 6,000 cub. ft. of free air per minute.

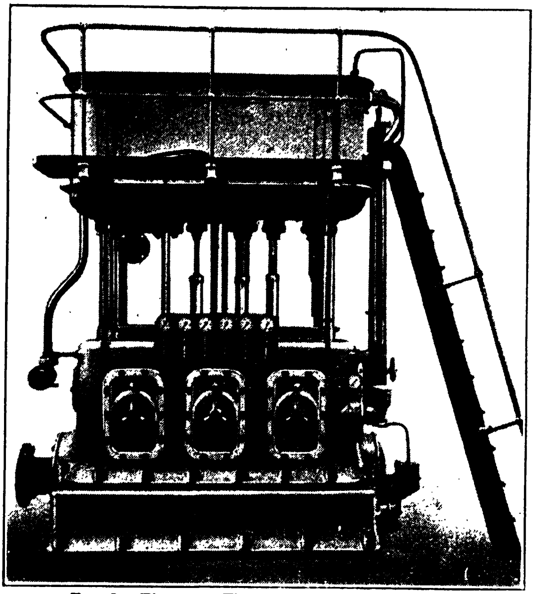


FIG. 3.—Five-stage Three-crank Gas Compressor.

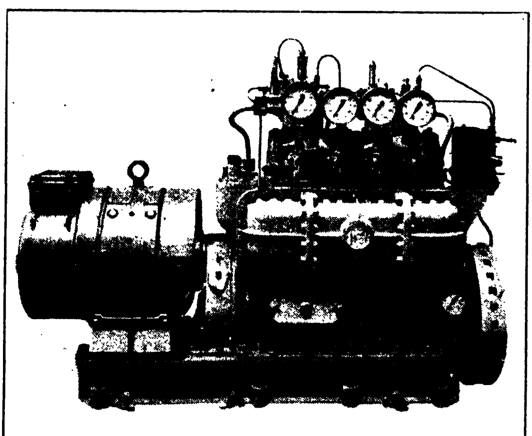


FIG. 4.—Four-stage Air Compressor.

High-Pressure Air Compressors.

(*G. & J. Weir, Ltd., Cathcart, Glasgow.*)

Fig. 1 shows a typical Weir four-stage, twin-crank, motor-driven air compressor for a delivery pressure of 4,500 lb. per sq. in. In this machine the first and second stages are formed by differential type pistons in two cylinders. The first stage of compression takes place in the tops of both cylinders, the air then passing through the first-stage intercooler and back to the second-stage which is formed in the annular space in both cylinders below the pistons. The third stage to which the air passes after leaving the second-stage intercooler is at the top of one of the differential piston rods and the fourth stage at the top of the other. There is of course an intercooler between these two stages. From the fourth stage the air passes to the pressure main

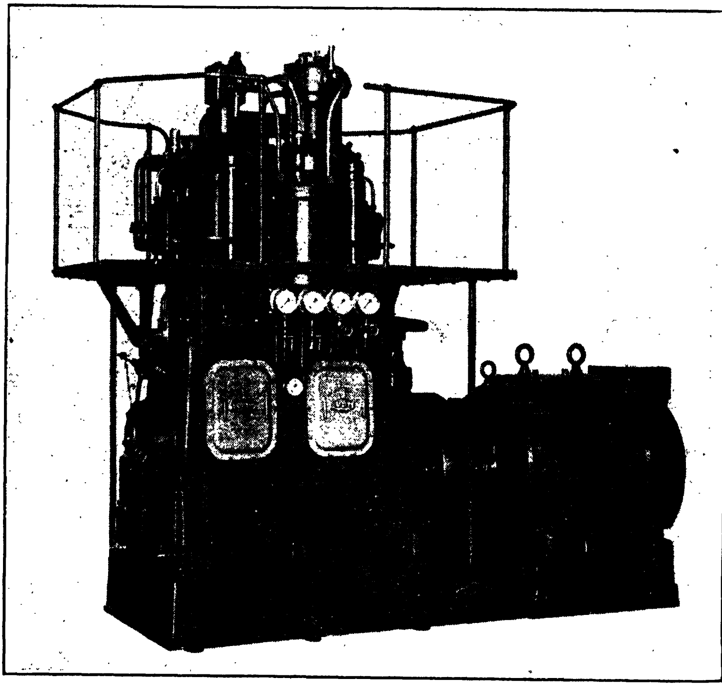


FIG. 1.—Weir four-stage, two-crank, motor-driven Air Compressor. 4,500 lb. per sq. in.

through the after-cooler. Oil and moisture separators are fitted to each intercooler and the after-cooler.

The pistons are of high-grade cast iron fitted with cast-iron rings of the Ramsbottom type With the perfected lubricating and cooling arrangements in the Weir compressor, the wear on these cast-iron rings over long periods of running is practically negligible, the tool marks being visible after years of running.

The air valves are of special design and the lift is exceedingly small, preventing undue shock and noise and obtaining a high degree of efficiency. The assembled valve units are arranged round the cylinder heads in such a manner as to reduce the cylinder clearance to a minimum, and give a resultant high volumetric efficiency.

All the running gear is forced lubricated by means of an oscillating type of oil pump. Cylinder lubrication is effected by a mechanical sight feed lubricator, feeding minute quantities of oil at

the correct points on the cylinder walls, and maintaining the thin even oil film so necessary with comparatively high-temperature conditions.

The foregoing description applies to compressors built for high pressure land service, but is also typical of three-stage compressors for pressures up to 1,000 lb. per sq. in., largely supplied for motor ship auxiliary duties, viz. starting, Diesel engine air-blast injection, manœuvring, etc. Other types include two-stage compressors for 450 lb. per sq. in. pressure, emergency tank-cooled compressors, and general service machines for all duties.

Fig. 2 illustrates a standard Weir four-stage, four-crank oxygen compressor for a delivery pressure of 4,000 lb. per sq. in. The first and second stages are double-acting, and the third and fourth stages single-acting, each crank carrying one stage only.

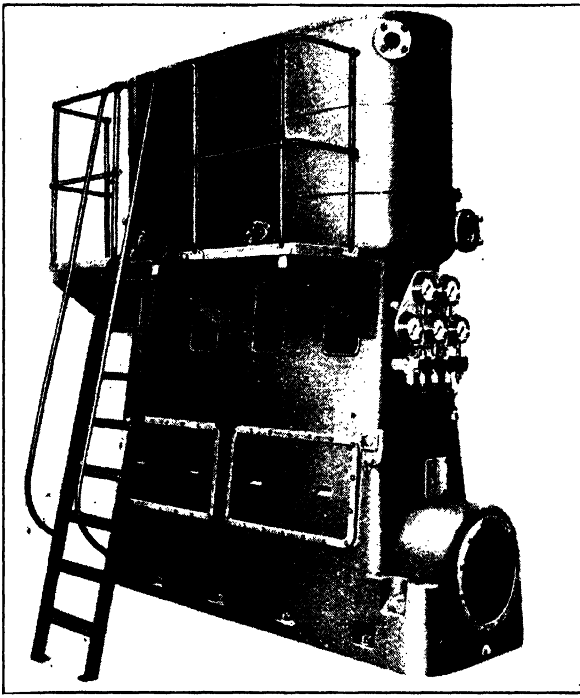


FIG. 2.—Weir four-stage Oxygen Compressor. 4,000 lb. per sq. in.

The whole of the cylinders and cooling coils are immersed in an open tank with an ample supply of cooling water to dissipate the heat of compression.

In compressing oxygen, oil must not be used for cylinder lubrication, as oil and oxygen form a highly explosive mixture. The cylinders therefore are lubricated with distilled water, which necessitates the use of corrosion-resisting materials and specially designed piston rings and liners to counteract undue wear caused by the poor lubricating properties of distilled water.

The piston rod glands are water sealed to prevent any leakage of oxygen to the atmosphere.

The running gear is totally enclosed and forced lubricated throughout by means of an oscillating type lubricating oil pump.

Compressors dealing with oxygen and other industrial gases for delivery pressure ranging to 6,000 lb. per sq. in. are constructed, and may be arranged for drive either by electric motor, independent steam engine, tandem steam engine, or oil engine.

Water Coolers.

(Heenan & Froude, Limited, Worcester, England.)

The P type cooler manufactured by this firm and described separately on p. 1273, is particularly useful for employment with air compressors on account of its capacity for providing very cool water even in hot weather. Owing to its operating on the principle of evaporation, its capacity is controlled much more by the prevailing atmospheric wet bulb temperature than the dry bulb temperature, and consequently by suitably proportioning it to its duty, the water can be cooled down to a temperature closely approaching the wet bulb temperature.

Thus in hot weather, in both temperate and tropical climates, cooled water can be provided considerably below the dry bulb temperature of the surrounding atmosphere; and as the efficiency and power absorption of air compressors depend to some considerable extent upon the temperature of the circulating water, this is a most useful feature of the cooler. The apparatus is constructed entirely of metal and is self-contained, and the space taken up is very small in relation to its duty.

The cooler can deal with the water of inter-coolers and after-coolers if required, thus reducing the final air temperature and minimising condensation in the exhaust from compressed-air tools, etc.

The air compressor is rendered almost independent of town's water, and the cooler effects a large saving in water costs.

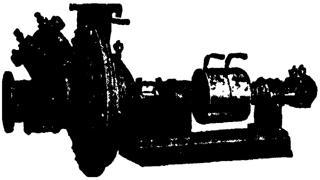
SECTION XXXVII

Sanitary Engineering.

THE 'STEREOPHAGUS' PUMP.

(The Pulsometer Engineering Co., Ltd., Nine Elms Works, Reading.)

The 'Stereophagus' pump is designed to deal with unscreened sewage, and, as it cuts up the solid matter in suspension to small pieces, the sewage as it leaves the pump is easily dealt with during subsequent treatment. The pump is simple and substantial in construction, and is low in installation cost as well as in maintenance.



It is equally suitable for dealing with trade effluents containing obstructive matter of any sort. The problem of pumping fluids containing flocculent or fibrous matter has always been a difficult one, and though a large variety of pumps has been tried, trouble from choking has always remained a pronounced feature. In reciprocating pumps, the choking of the valves is a great source of trouble, and in the case of centrifugal pumps, the clinging and accumulating of flocculent matter to the spindle and impeller necessitates frequent stoppages for cleaning. The 'Stereophagus' pump overcomes all these troubles.

SECTION XXXIX

PART I

Mining.

COAL FACE MACHINERY.

(Mavor & Coulson Limited, Glasgow and Sheffield.)

Longwall Coalcutters.—The modern chain coalcutter has become so powerful and adaptable that it has superseded all other types. The Samson coalcutter, for example, can cut longwall faces up to 8 ft. deep, and cuts at speeds up to 5 ft. a minute. Areas of 3,000 to 4,000 sq. ft. are regularly cut in the shift, and more on occasion.

The introduction by M. & C. of the spiral gummer has further improved longwall coalcutter performance. The gummer consists of a short worm conveyor, which takes away the holings as brought out by the cutter chain, and places them either at the side of the machine or behind it, according to the type of gummer. The advantages are: (1) Increased safety. The gummer completely encloses the cutter chain on the driving sprocket, both from the goaf side and from the end of the machine. There is no longer any need for a man to work close behind the machine, an especially important point under bad roof. (2) Greatly reduced fine dust. What dust is made is kept on the floor, buried under the larger holings, which come to the top. (3) The arduous work of shovelling away the holings as they are brought out is avoided. The gummer takes away at least 80 per cent. of the holings—far more than could be done by hand. (4) Power is saved, because the holings are no longer churned by the picks. (5) Picks keep sharp longer and cut further because they are not blunted by churning the holings. The time taken in changing picks is also reduced. (6) Other moving parts are given longer life, and the cost of replacements is correspondingly reduced. (7) Samson coalcutters with gummer travel freer and faster. (8) Better coal preparation. Any holings left in the cut are near the front, and the back, which is so hard to clean by hand, is left clear. The coal therefore shoots better, and explosives may also be saved.

Arcwall Coalcutters.—M. & C. arcwall and arcwall-and-sheering coalcutters commonly carry a jib 9 ft. long, making an arc cut up to 9 ft. deep at the end of a 'room' 21 ft. wide. Under favourable circumstances, the jib can be 11 ft. long, cutting 25 ft. wide. The machine travels from room to room, either on rail wheels or on crawlers, and may regularly cut more than 20 rooms in one shift. The machines usually have a 60 h.p. motor, and weigh from under 4 tons to over 7 tons.

Face Belt Conveyors.—A very widely used type of face conveyor is the flat belt running between side plates. The side plates make filling easier, reduce spillage, and protect the belt. When running at 150 ft. per minute with average loading, the belt has the following capacity:

Width of Belt.	Tons of Coal per Hour.
20 ins.	75
24 ins.	90
26 ins.	100

The structure of M. & C. face belts is formed of inverted troughs, which keep all dirt from reaching the return belt, no matter how coal is shovelled against the conveyor or how dirt rolls against it out of the goaf. The inverted troughs allow the belt to run under conditions where an unprotected trough would suffer severely.

Roadway Conveyors.—A pre-eminent conveyor for mining roadways is the troughed belt, introduced for large-scale underground conveying by M. & C. The capacity of M. & C. troughed belts handling run-of-mine coal is as follows:

Width of Belt.	160 ft./min.	260 ft./min.	340 ft./min.
24 ins.	80	130	170
26 ins.	100	165	210
30 ins.	140	230	300
36 ins.	200	330	430
42 ins.	—	490	640

As with the flat belts, these troughed belts are kept safe from damage by inverted troughing which completely covers them. The method by which the rollers are mounted on ball bearings reduces the friction to the lowest possible amount, and allows a smaller motor to be used or the

conveyor to extend further. When conveying up more than 1 in 16, the belt is kept from running back by a holdback. When conveying down steeper than 1 in 16, the belt is brought to rest by a brake, which is applied automatically when the electric or air supply is cut off the motor. The loading of large outputs into tubs is facilitated by a shaking chute, and the tubs can be handled beneath the discharge point by a tub-pusher. Extending the conveyor daily to keep up with the advancing face is simplified by the M. & C. loop take-up, which also gives an easy way of tensioning the belt.

M. & C. Loaders.—M. & C. Loaders are mobile machines for loading coal or other loose material. After the coal has been undercut and blown down, the loader advances on its crawlers and thrusts its gathering head into the heap. While it does so, two gathering arms, acting alternately, sweep and pull the coal on to the chain conveyor, which carries the coal to the end of a flexible jib and delivers it into tubs, shuttle cars, or a conveyor.

Hydraulic movements raise the gathering head off the floor for travelling, and slew or raise the jib, in order to deliver the coal just where required. These machines load large quantities of loose coal in mechanised room and pillar mining, and also have enabled stone-drifts to be driven at over 40 yards a week.

SECTION XLII

Paint and Painting Equipment.

GRAPHITE PAINT.

(*Graphite Products, Ltd.*, 52 *Battersea Church Road, S.W.* 11.)

The best and longest records of satisfactory protection against corrosion are said to be held by paints made with tough crystalline foliated graphite containing silica within its structure.

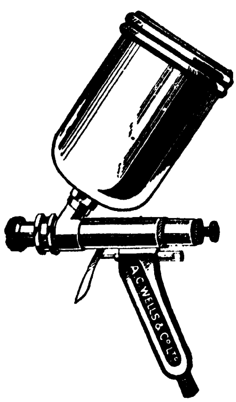
An example of this is the 'Foliac' Silica-Graphite Paint, manufactured by Graphite Products, Ltd., Battersea, which is British manufactured throughout, and has been specified for a large number of steel structures, including aerodrome buildings, oil fuel tanks, gas holders, docks, bridges, station roofs, railway wagons, etc.

The relative covering power of foliated silica graphite paint is very high, and owing to the light weight lubricating properties of the pigment the speed of application to the surface is very much more rapid than with other linseed oil paints. Owing to the large adhesion surface between the vehicle and the foliated graphite a very flexible tough paint is obtained which stands wearing action due to dust, for example, much better than paints which become brittle on drying. The resistance of the pigment to the action of the atmosphere found in industrial districts is due of course, to the fact that it is chemically inert.

SPRAY PAINTING EQUIPMENT.

(*A. C. Wells & Co., Ltd., Mount Street, P.O. Box* 5, *Hyde, Cheshire.*)

Spray Guns.—Of recent years the application of paints, varnishes, stains, fillers, french polishes, distempers, etc., by the spray method has become more general. The Spray Guns are fitted with a range of nozzles for different purposes, and consume 4/8 cub. ft. of compressed air per minute at pressures of 40/60 lb. per sq. inch. The Standard Gun is fitted with an adjustable nozzle whereby the deflector holes in the nozzle can be set to produce a round or flat spray. Generally the flat spray is about 6 ins. wide, but if the gun is being used with a Pressure Solution Container, this width can be increased to approximately 10 ins.



Solution Containers.—The solution to be sprayed is supplied to the gun either by a small container attached above the gun, when the solution falls by gravity, or below the gun where the solution is taken up by suction. It is not practicable, owing to the weight, to have these containers more than one pint or in exceptional cases, one quart capacity. In many instances, therefore, it is usual to supply the solution to the guns from gravity or pressure containers of suitable capacities. The latter type gives a wider spray, as the solution reaches the nozzle under pressure.

A full range of pressure containers is made in capacities up to 50 gallons.

Air Compressors.—The air for the gun is supplied from standard air compressor units which can be arranged either in portable or stationary form, to be driven by electric motor, petrol engine or other form of power.

Air Filters.—It is necessary to have clean air free from dust, oil or water for the spray gun, to ensure satisfactory action of the spray gun and a clean, unblemished finish of the work. In each equipment, therefore, an air filter is necessary. This is usually supplied with an air reducing valve, which enables pressure to be set to a suitable limit for the solution being sprayed. The air filter itself is in the form of a welded steel container with a perforated tube fitted with absorbent filtering material. This filtering material can easily be replaced or dried.

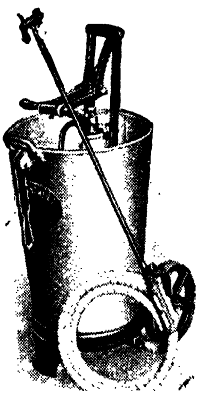
Air Receivers.—These welded steel vessels are placed in the air line between the compressor and air filter. The effect is to balance the impulses from the compressor and allow the water of condensation to settle out.

LIME SPRAYERS.

(A. C. Wells & Co., Ltd., Mount Street, P.O. Box 5, Hyde, Cheshire.)

These machines take the place of the old method of limewashing with a brush. Lime, whiting or cold water paints can be applied at a speed of from 10 to 20 sq. yds. per minute in a manner superior to brush work. One coat with this machine on rough or dirty surfaces is claimed to equal two applied with brushes. The material is applied in the form of a spray, and is driven into corners and difficult places where a brush would not reach.

Limewashing machines will spray all water preparations, but will not use oil-paints or varnishes. For the latter, compressed air is necessary to break the liquid up into a spray, necessitating special plant coupled to the user's own air main or suitable compressor.



Lime washing machines can be arranged to spray solignum, creosote and other wood preservatives, also insecticides of every description. In this connection the No. 6 machine is specially recommended.

These machines are designed to withstand rough usage. They are made of durable materials, which, although adding to the initial cost, effect a great economy in their life and upkeep.

The spraying nozzle is the nerve centre of a lime washing machine, and its construction is, therefore, a most important feature. The nozzle can be regulated to any degree of fineness by means of the spray regulator, while the patent filter prevents clogging. The whole can be cleaned easily, and replaced quickly. An additional and essential feature is the ability to swivel the nozzle to any angle, thus greatly assisting the operator and enabling a right-angled spray to be brought to bear on any surface.

The pump is simple and easily removable, while the bell-handle and trigger valve provide a comfortable grip and an immediate control over the spray by the mere pressure of the hand. Attention is called to the air chamber, which is constructed of sufficient size to enable an even spray to be maintained without continual pumping.

SECTION XLVII

(Unclassified.)

Diving Apparatus, etc.

DIVING APPARATUS; BREATHING AND RESUSCITATING APPARATUS, ETC.

(Siebe, Gorman & Co., Ltd., Davis Road, Tolworth, Surrey.)

The firm of Siebe, Gorman & Co., Ltd., was established in London by Augustus Siebe (1788–1872) in the year 1819. Siebe was a mechanical genius whose versatility covered the invention not only of diving apparatus, on which his fame chiefly rests, but also such widely different productions as printing machines, clock-work mechanisms, weighing machines (including the first height-and-weight machine, of which he supplied a number to the Government during the recruitment of troops for the Crimean war; one of these machines is still in the firm's possession), air pressure and water gauges, valves, air and water pumps, electrical apparatus, etc. He made, in 1850, the first practicable steam-driven ice-making machine (James Harrison's patents) produced

FIG. 1.

FIG. 2.

in England, and exhibited it at the great exhibition held in London in 1851. In 1823 he received the Silver Vulcan medal of the Royal Society of Arts for one of his inventions. He was an associate of the Institution of Civil Engineers.

Siebe made his first 'open' diving dress in 1819—*i.e.* a helmet attached to a jacket which extended to just below the waist, and functioned like a diving bell. In 1837 he invented the closed diving dress and helmet on the principle which is still in universal use. No matter what the shape of the helmet, type of valves, and method of securing to the rubber dress, the principle remains Siebe's throughout the world.

Siebe died in 1872, and was succeeded by his son, Henry Siebe (1830–1885) and son-in-law, W. A. Gorman (1834–1904), who had been Siebe's assistants for many years. The present head of the company is Sir Robert H. Davis (who entered the firm in January 1882 and served under

Henry Siebe and W. A. Gorman). With him as co-directors are his sons, R. W. Gorman Davis and W. Eric Davis; while his son, R. J. (Peter) Davis, is also in a managerial position.

In the firm's possession are the originals of a number of orders received by A. Siebe over a hundred years ago from the Admiralty for diving appliances, etc., for various naval establishments at home and abroad, and for work at the wreck of the *Royal George*, sunk at Spithead in 1782.

The firm is probably the oldest maker of air pumps (pressure and vacuum) in the world. It is certainly the oldest maker of diving and other submarine appliances, of gas masks and self-contained breathing apparatus for work in poisonous atmospheres, and of oxygen breathing apparatus for airmen flying at great altitudes

It had already for many years before 1879 been producing smoke helmets and breathing apparatus of the fresh air type, but in that year the firm turned its attention to the design (in collaboration with Henry A. Fleuss) and manufacture of breathing apparatus on the self-contained regenerative system, using compressed oxygen with CO_2 absorbents. In the same year it produced the first really practicable apparatus of the kind, and this was actually used at the

FIG. 3.

FIG. 4.

Killingworth Colliery, after a disastrous explosion there, in 1880; also at the Seaham Colliery. The same principle was also adapted to Siebe, Gorman & Co.'s diving dress with good results. It should be added that both apparatus on this principle, *i.e.* for work in poisonous atmospheres and under water, are the prototypes of all apparatus of the kind used throughout the world.

The Siebe-Gorman diving apparatus with air delivered to the diver by means of manually-worked pumps, or by air compressors—electrically, steam or oil engine driven—are so well known that we are not describing or illustrating it here. The following may, however, interest the reader:

Fig. 1 illustrates a self-contained diving apparatus consisting of the usual Siebe, Gorman & Co., diving dress and helmet with corselet; a steel cylinder charged with a mixture of pure oxygen and atmospheric air at a pressure of 1,800 lb. per sq. in.; a reducing valve and injector, delivering automatically the requisite quantity of oxygen, and circulating the air in the helmet and dress through the CO_2 absorbent.

The 'Proto' self-contained breathing apparatus (fig. 2) is used largely for rescue and recovery work in mines, and by fire brigades, gas works, chemical works, etc. It consists of a combined flexible breathing and CO_2 absorbent chamber carried in front of the wearer, a steel cylinder, charged with pure oxygen at 1,800 lb. per sq. in., carried on the back, with a reducing valve passing oxygen automatically into the breathing bag at a constant rate at all stages of pressure in the cylinder; a mouthpiece attached to flexible corrugated tubes fitted with inspiratory and expiratory valves connected to the breathing-absorbent chamber. The latter is divided by a

partition, the exhaled air passing through the expiratory valve, down one side of the partition, through the absorbent, and returning, purified, on the other side of the partition, through the inspiratory valve. Adjacent the latter valve is delivered the stream of fresh oxygen from the reducing valve. An oxygen by-pass is fitted in case of emergency, and a pressure gauge, attached

FIG. 5.

to a high pressure flexible tube, registers the amount of oxygen in the cylinder and the duration of its supply. The wearer's nose is clipped, breathing being done by the mouth only, so that, doing hard work, he is quite safe in the most poisonous or oxygenless atmosphere for at least two hours at a time.

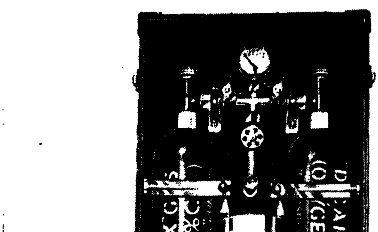


FIG. 6.

The 'Salvus' self-contained breathing apparatus, illustrated in fig. 3, is on the same principle as the 'Proto,' but in this case the apparatus is carried wholly in front of the wearer, and is suitable for only forty minutes' work in poisonous atmosphere. The CO_2 absorbent cartridge is separate from the breathing chamber.

Gas masks for use in all kinds of poisonous air encountered in industry, including carbon monoxide, as well as the various war gases are made (fig. 4).

Other forms of breathing apparatus are the 'Spirelmo' smoke helmet and 'Puretha' mask supplied with fresh air by means of bellows, blowers or pumps, and the 'Antipoys,' with which the wearer draws his supply of air by his own lung effort through a tube, the free end of which is kept in open fresh air; compressed air apparatus consisting of a cylinder or cylinders or atmospheric air with a lung governed breathing device, mask, etc.

Figs. 5 and 6 show the 'Novox' resuscitating apparatus for reviving the apparently drowned and persons suffering from shock. The apparatus consists of a cylinder of pure oxygen with 7 per cent. of carbon dioxide, and a lung-governed breathing device, in a flexible chamber, which gives the patient automatically more or less of the gaseous mixture, according to his needs. The addition of the 7 per cent. CO_2 stimulates the respiratory centre, causes the wearer to take in more oxygen and so hastens expulsion of the poison from the system.

Another important apparatus produced by the firm is the Davis submarine escape apparatus, the invention of Sir Robert Davis.

During the 1939–45 War much of the firm's effort was devoted to the design and production of special equipment for underwater warfare which included apparatus for the crews of 'Human Torpedos,' Midget Submarines and submersible canoes, mine recovery apparatus, etc.

The Admiralty Experimental Diving Unit, formed in the early part of the war, was located at the firm's works where the full resources of its Experimental and Research Department were placed at its disposal.

Linatex Rubber.

(*Wilkinson Rubber Linatex Ltd., Frimley Road, Camberley, Surrey.*)

Linatex is a special form of proprietary rubber manufactured direct from fresh rubber latex It has great resilience and a high resistance to abrasion and perishing. It can be built up by cold cementing processes to conform to any required shape.

Special cements are available for bonding Linatex to wood, metal, or concrete, and additional mechanical anchorage can be achieved by metal straps.

The following notes will give some indication of the many applications of this material :—

Air Conditioning Equipment.—For flexible trunkings, fan and fan-blade-coverings, Linatex 95 per cent. pure natural rubber can be employed. It is also suitable for the lining of ductings in ventilation and extraction plants.

Aircraft Accessories.—Flexatex hose is still used for aircraft petrol and nitrogen systems. A wide range of sealing sections for window frames and pressurised cabins can be supplied in either Linatex or Novatex, depending on location and working conditions.

Anti-Vibration Mountings.—Linatex 95 per cent. pure natural rubber has been found eminently suitable for damping vibration. The makers undertake to tackle each problem individually—from a steam hammer to a small camera mounting.

Centrifugal Pumps.—Pumps particularly suited for handling ash, sand, gravel and corrosive chemicals are built in Linatex from 1-in. to 9-in. bore. It is claimed that an impeller of practically solid Linatex rubber will outlast steel in such applications.

Coal and Ash Handling Plant.—The special anti-abrasion properties of Linatex, the makers state, can be employed in this sphere with good effect. Chute and hopper linings will outlast steel, and the Linatex pump cannot be used to better advantage than in the handling of ash sludge.

Cocks, Valves and Pipe Fittings.—Linatex cocks and valves are now being produced for the handling of slurries in sand, gravel, slime and corrosive chemicals. In their design attention has been paid to full-bore flow, positive seating and the need to withstand heavy abrasion.

Gaskets and Jointing Materials.—A wide range of gaskets made from Linatex 95 per cent. pure natural rubber is now in use. Where Linatex is unsuitable, gaskets in Novatex are supplied. This is a rubberlike material which effectively resists the action of oils and petrols.

Hose Pipes and fittings.—Hose lines of Flexatex continue to have a wide field of use in aircraft. This hose is made from a plastic material chosen for its resistance to oils, fuels and gases. It is also used in the handling of chemicals.

Lining (Rubber).—Linatex lining of gravel plants and chutes has proved of great benefit on the score of resistance to abrasion. It is also widely used for lining acid tanks, pipes, and ductings for fume-extraction plant.

Mills (Ball and Pebble).—Designed to use Linatex anti-abrasion rubber in compression, which is its best condition for resistance of abrasion, Linatex ball mills not only show better grinding times, but preclude the possibility of rubber linings working loose. The wear is negligible over a period of years.

Plastic Materials.—Novatex is a proprietary rubberlike material, suitable for extruded sections, oil and petrol-proof gaskets, and in general can be used where rubber is precluded.

Rubber.—The anti-abrasion qualities and resilience of Linatex 95 per cent. pure natural rubber have now established it firmly as a basic engineering material during twenty-five years of universal application in mining, chemical and engineering spheres.

Seals and Sealing Sections.—Any shape or size of seal can be fabricated in Linatex, and where excessive oil is present alternative materials can be used.

Pneumatic Despatch Tubes.

(*Lamson Engineering Co., Ltd.*, 6, 7 *&* 8 *Hythe Road, Willesden, London, N.W.* 10.)

Pneumatic Tubes are used by stores for handling cash and credit transactions, by industria concerns for conveying papers, samples, small articles, etc., between departments and buildings. They are also used in many Government departments, warships, aerodromes, etc.

There are two methods of effecting transmission by pneumatic tubes—the 'single' and 'double' tube systems. The former consists of a single tube and can be used for transmission in either direction by reversing the flow of air. To-day, this system is usually confined to long lines operated by pressure. The 'double' tube comprises two tubes, by which a continuous supply of air passes down one tube and returns by the other, to the exhauster. This system is usually worked on suction; carriers are thus allowed to travel in both outgoing and incoming tubes at the same time. Even the simplest communications, such as those connecting two points only, are carried out by double tubes with a small motor-driven exhauster applied to one end of the tube. The cost of such a system is low, and when equipped with push-button start and automatic stop, the amount of electric current used for intermittent operation is negligible. 'Double' tube systems may be either of the 'independent line' type when continuous service at will in either direction is provided, or they may be constructed on the 'shifting' plan, in which case the service is, to a certain extent, restricted. In either instance suction only is employed. 'Shifting' lines have one tube—common to two or more stations—on the incoming line to a central desk, but an independent outgoing tube to each out-station.

Any number of tubes may be connected to a central station, and these, in turn, through a main vacuum or pressure pipe, are connected to the exhauster, which usually operates continuously. Where there is a large number of stations, only a proportion of which are in simultaneous use, means can be provided to reduce automatically the speed of the exhauster, or to reduce the load on the machine, to suit the number of tubes in use and so effect a considerable reduction in power cost.

The motive power for operating the pneumatic tube system can be a turbine or Roots type blower. The usual practice, nowadays, is to use the turbine. The size and type of machine depend upon the diameter and length of tube or tubes through which the carriers have to travel, and the requirements of service. There is practically no limit to the length for a pneumatic tube, but for abnormal lines it is necessary to use pressure.

Tubes of 2¼ in. diameter are generally used. 40 mm. tubes are common in hotels, restaurants, etc., for conveying service chits and checks. Other typical sizes for samples, small parts, books and files, large documents, newspaper 'fudge' and matrices are 3 in., 4 in. and 10 in. diameter tubes, 7 × 4 in., 13 × 5 in. and 14 × 9 in. oval tubes.

Where cash is being handled, the central desk is usually designed as follows :—

'*Belt Operated*' *Type.*—Carriers arrive at the desk and are conveyed on a moving belt to the cashier; the change is made and carriers are placed on another belt, which takes them to despatching centre, whence they are returned to sales stations in the store. The flexibility of this type of desk allows the system to be operated by only one, or more than seventy cashiers, according to the size of the store or fluctuations in business. Patent carrier-operated separators automatically transfer credit and account transactions to a credit office for authorisation.

'*Gravity*' *Type.*—This desk is designed for smaller installations up to thirty stations. The incoming tubes are arranged at one or both ends of the desk. Carriers are delivered from terminals into a chute, down which they travel to operator. Despatch terminals are located at the centre or end of desk to facilitate speedy return of carriers.

With both 'Belt' and 'Gravity' operated desks, carriers are handled by operators in consecutive order of their arrival.

A new type of system now being installed in this country is the Carrier Operated Switch System, whereby carriers travel through an endless tube and are automatically discharged at any point by means of an electrical impulse which is set up through previous adjustment by the operator of interconnected indicating rings on the carriers.

The principal advantages of this system are that all stations inter-communicate without the necessity for a central desk and therefore the usual double tube between central desk and outstations is not needed.

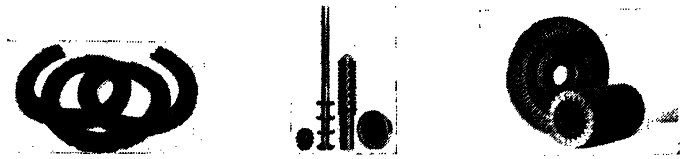
Brushware.

(*Kleen-E-Ze Brush Co. Ltd., Hanham, Bristol.*)

There was a time when, apart from catering to household and decorating needs, brushmakers met such demands as were made by industrial requirements from their limited stocks, either in

standard or slightly modified form. In other words, industry had to 'make do' with what was available. Now brushmaking is recognised as a specialised manufacturing industry.

Whilst there was a trend in the direction of creating brushes for specific industrial needs before hostilities, the war definitely gave impetus to this service and it is now possible to apply the expression 'industrial brushware of every description to customer's needs' or some such phrasing in actual truth!



Kleen-E-Ze Brushware.

Examples of specialised productions include:—

(a) *Component parts.*—'Brush-form' coils, 'Brush cloth' in many types, 'Multibrush' fibre units, brush guides, etc.
(b) *Processing 'tools.'*—Decarbonising brushes, wire wheels, polishing heads, scouring brushes, 'Kezas'—scalariform coiled units for cleaning unloaded supply tubes, internal tube burnishers, lacquer, paint and oil brushes, etc.
(c) *General production.*—Scruffing, burnishing, suds and tube brushes; polishing soap, bench, foundry, laboratory and maintenance brushware, etc.
(d) *Cleaning purposes.*—Platform and yard brooms, sweeper, dusting, polishing and hand brushes; scrubber, stencils, pipe, paint and varnish brushes, etc.
(e) *Personal use.*—Washroom brushware including hair, clothes and nail brushes, etc.

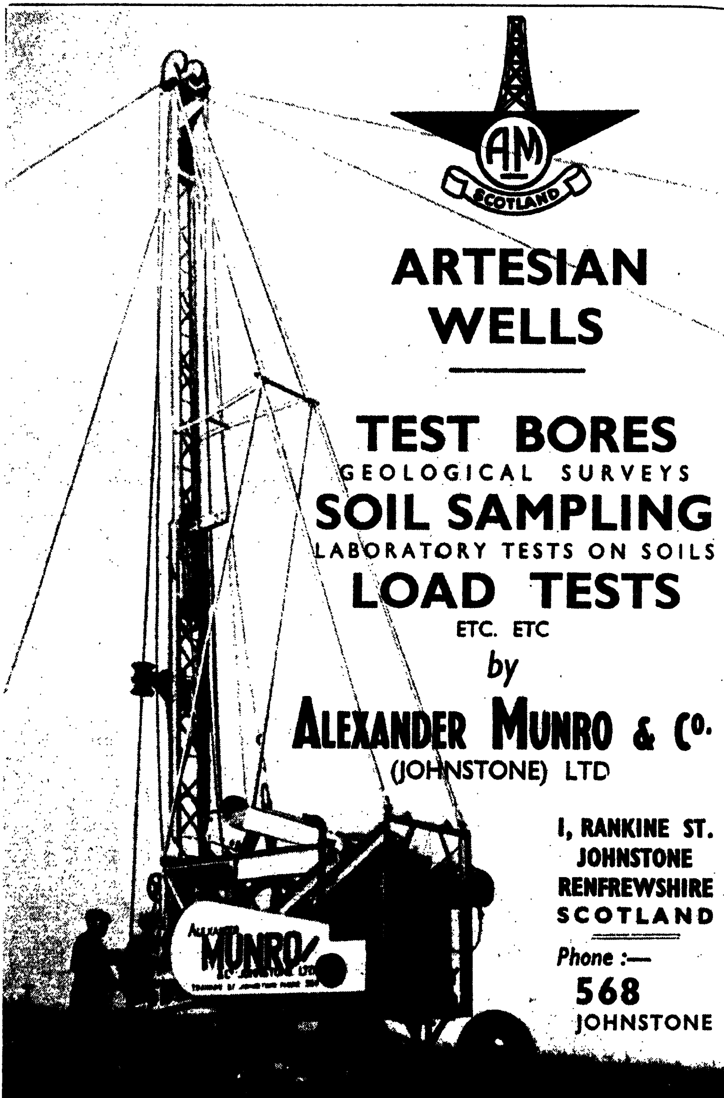
This firm's policy for the past twelve years has embodied a scheme of training technical representatives able to make first-hand investigation of specific requirements and to submit recommendations of brushware with full details of processing, etc. It has created over 9,000 types ranging from a small applicator with a ¼-in. head by ⅛-in. dia. to roller brushes (single-unit-type) 15 ft. long with a 6 ft. periphery. This has necessitated making scale drawings and models, often introducing mechanical operational arrangement, incorporating a variety of materials for laboratory tests to determine final needs and approval. Such a service to the customer has its appeal and means that the fullest possible experience is being given to solving his particular problem.

How many times has a mechanical device been in course of manufacture when it has been discovered that a small but essential unit does not function properly or has been designed on the drawing board and is technically wrong? Undoubtedly brushes come high in such a list of circumstances. Just now, for example, there is a spate of new designs, either in or just out of the prototype stage, requiring a brush component but it can be taken for granted that in only a small proportion of cases will the manufacturer have the advantage of preliminary discussion with a brush maker who is in the position to know all the snags of types, materials, lay-out, etc., and thus to learn the exact type around which the device can be built.

Brushing problems apply to the engineering world whether it be the aircraft, motor, shipbuilding, ordnance, locomotive or other classes. All have their separate and individual problems and would do well to take advantage of the services of any firm offering technical advice. A price-list will be a useful reference for ordinary needs, of course, but how can any firm give stock advice, literally or pictorially, that can be certain to fill a technical application?

Many cases can be cited when brushware designed for a special purpose has contributed towards the solution of problems widely varied in character yet closely allied in operation. One such was the result of an enquiry of an almost casual nature during practical experiments in high-altitude flying with liquid oxygen converters. It was suggested that a special component that was used as an oil-flow breaker might be modified to form a unit to fill a function that was not being obtained satisfactorily. Not only did the ultimate design meet the need, it reduced the vertical dimension of the apparatus by 50 per cent., a feature not even considered in the first instance!

To-day the brushmaking industry is expanding to meet the growing needs of engineers and manufacturers generally, who are finding new uses for brushware that can be made in so many forms to fulfil special purposes. This growth in the variety of brushware, has, of course, turned the trade from the simplest of processes to a complicated and precise manufacture embodying elaborate wood-working engineering and allied trades within its own. For example, a brush-head itself in the form of manufacture required, necessitates turning, spindling, routing, double-drilling mechanical dividing, mechanical knot picking, filling, trimming, capstan and centre-athe work, riveting and assembly.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INDEX

Note : The Roman numerals in parenthesis (I) and (II) refer to Volumes One and Two respectively.

ABRASION test of road stones, 548 (I)
—— for hardness, 1089 (I)
Abrasive cloths and papers, 1114 (I)
—— materials, 1097 (I)
—— wheels and blocks, 1098 (I)
———— dressers, 1106 (I)
———— dust collection, 1106 (I)
———— flanges, 1104 (I)
———— guards for, 1105 (I)
———— lubrication of, 1107 (I)
———— mounting, 1102 (I)
———— papers, 1114 (I)
———— rests for, 1106 (I)
———— selection of, 1098 (I), 1101 (I)
———— sizes of spindles, 1102 (I)
———— speed of, 1098 (I), 1101 (I)
———— storage of, 1102 (I)
Absolute zero of temperature, 50 (I)
Absorption, losses by, in reservoirs, 705 (I)
—— of sound, 73 (I)
—— rainfall, 663 (I)
Absorptive powers of coloured glass, 653 (I)
Accidents, road, 556 (I)
Accumulators, electric, 1476 (I)
———— acid for, 1477 (I)
———— charging, 1478 (I)
———— nickel-iron, 1840 (I)
—— hydraulic, 772 (I)
—— hydro-pneumatic, 774 (I)
—— steam, 776 (I), 183 (II)
Acetylene, 1367 (I), 1370 (I), 651 (II)
—— dissolved, 646 (I), 1087 (I)
—— for buoys, 646 (I), 653 (II)
—— lighthouses, 651 (I)
—— lighting, 654 (II)
—— piping, 1273 (I)
—— purification of, 1272 (I)
—— regulators, 1273 (I)
—— supply for welding, 1271 (I)
—— welding, 1267 (I)
———— apparatus, care of, 1272 (I)
Acid action on aluminium, 1207 (I)
—— for electric accumulators, 1476 (I)
Acid-resisting alloys, 1219 (I)
Ackerman steering gear, 371 (II)
Acme screw threads, 93 (I)
Acoustical defects, 74 (I)
—— treatment of buildings, Fibreglass, Ltd., 1097 (II)
Acoustics, 69 (I), 1097 (II)
Activated sludge process, 788 (II)
Adhesion of locomotives, 531 (II)
Adhesive cements, 353 (I)
—— strength of mortars, 181 (I)
Adiabatic expansion, gas, 1300 (I), 189 (II)
———— compression, 323 (II), 757 (II)
Adiabatic heat-drop, 118 (II)
Admiralty coefficients, 281 (II)
—— gun-metal, 1223 (I)
—— pier, Dover, 607 (I)
—— specification for oil fuel, 1361 (I), 1362 (I)
—— tests for iron, 164 (I)
———— steel, 164 (I)
—————— rivets, 165 (I)
—— white metal, 1213 (I)
'Adnic' white nickel alloy, 1220 (I
Adulteration of metals, to discover, by specific gravity, 121 (I)
Aeral, 1230 (I)
Aerial carrying ropes, 950 (I)
——ropeways, 831 (II)
———— carriers, 831 (II), 836 (II)
———— double rope or bi-cable system, 833 (II)
———— fixed clip single rope system, 838 (II)
———— jig back or to and fro, 839 (II)
———— limitations of types of, 839 (II)
———— single rope or mono cable system, 831 (II)
Aerodynamics, 390 (II)
—— wind channels, 390 (II)
Aero engines (*see* Aeroplanes)
Aerofoil theory, 399 (II), 405 (II)
Aerofoils, 393 (II) (*see also* Airscrews)
—— aerodynamic qualities of, 396 (II)
—— pressure distribution over, 397 (II)
—— slotted, 396 (II)
—— theory, 399 (II), 405 (II)
Aeroplanes, 389 (II)
—— beams, 423 (II)
———— offset moments in, 424 (II)
—— bulkheads, 428 (II)
—— climb, rate of, 414 (II)
—— dynamic similarity and scale effect, 392 (II)
—— electrical equipment of, 1455 (I)
—— engines, 445 (II)
———— air cooled *v.* liquid cooled, 457 (II)
———— cycle, 447 (II)
———— cold corrosion in, 455 (II)
———— cooling, 455 (II)
———— compression ignition, 465 (II)
—————— ratio effect on combustion, 448 (II)
———— cooling, 455 (II)
———— crankcases, 461 (II)

Aeroplanes, engines, cylinders, 462 (II)
——— detonation, 450 (II)
——— efficiency volumetric, 450 (II)
——— evaporative cooling, 456 (II)
——— fuel, 452 (II)
——— ideal air cycle, 447 (II)
——— installation weights, 457 (II)
——— jet propulsion, 265 (II), 448 (II), 464 (II)
——— liquid cooling, 445 (II)
——— mechanical efficiency, 445 (II)
——— ——— losses in, 446 (II)
——— performance figures, 464 (II)
——— petrol injection, 465 (II)
——— piston, 445 (II), 463 (II)
——— pistons, 461 (II)
——— power variation with altitude, 412 (II)
——— sleeve valve, 462 (II)
——— supercharging, 457 (II)
——— testing plant, 'Froude,' 1094 (II)
——— thermal efficiency, 446 (II)
——— turbine 418 (II), 464 (II)
——— valves, exhaust, 459 (II)
——— ——— cleaning, 460 (II)
——— vapour lock at high altitude, 454 (II)
——— volumetric efficiency, 450 (II)
——— weights, 457 (II)
—— estimation of H.P. available, 413 (II)
—— frames, 432 (II)
—— geometrical particulars of various, 433 (II)
—— horse-power required, 412 (II)
—— jet propelled, 265 (II)
—— 'Linatex' rubber applications, 1301 (II)
—— low drag wings, 398 (II)
—— materials of construction, 1241 (I), 414 (II)
—— monocoque construction 425 (II)
—— noise and insulation, 79 (I)
—— paints, 979 (II)
—— particulars of various, 433 (II)
—— performance, 411 (II)
—— ——— at different altitudes, 412 (II)
—— ——— curves, 414 (II)
—— ——— standard atmosphere, 411 (II)
—— riveting in construction, 418 (II)
—— speeds, 414 (II)
—— stiffness, 431 (II)
—— stressed skin construction, 425 (II)
—— stringers, 429 (II)
—— struts, 421 (II)
—— tension field web, 430 (II)
—— thin sections, torsion of, 429 (II)
—— using flaps, 395 (II)
—— welding in, 416 (II)
—— wiring lugs and sockets, 420 (II)
Afton dam, 695 (I)
Age-hardening of steel, 1179 (I)
Agriculture, electricity in, 1524 (I)
Air and oxygen required for combustion of gas, 1366 (I), 645 (II)
— blast cupola, (I)
— compressed, locomotives, 894 (II)
——— mining drilling by, 888 (II)
——— power distribution by, 889 (II)
——— water raising by, 826 (I)
— compression, 753 (II)
——— cost of, 1082 (I)
— compressors, 763 (II), 1288 (II)
——— Brotherhood, 1288 (II)
——— lubrication, 880 (I)
——— nozzles, 765 (II)
——— testing, 763 (II)
——— Weir, 1920 (II)
— conditioning, 676 (II), 1282 (II)
— coolers, Heenan, 1198 (II)
— cooling, 677 (II)
— cycle, aero-engine, 447 (II)
— density of atmospheric, 119 (I)
— discharge nozzles, 1116 (I)
— distribution of, for ventilation, 675 (II)
— draught stoker, Bennis Combustion, Ltd., 1220 (II)
— ducts, 679 (II)
— ejector, Weir, 1262 (II)
— expansion of, by heat, 63 (I)
— fans, 1341 (I)
— filtering, 676 (II)
— filters, 'Visco,' 1282 (II)
— flow of, through ducts, 679 (II)
——— nozzles, 1116 (I)
——— orifices, 756 (II)
——— pipes, 66 (II)
— gas, petrol, 654 (II)
— hammers, 1120 (I)
— heaters for boilers, 17 (II), 1226 (II)
——— efficiency of, 18 (II)
— humidity, 678 (II)
— lift pumps, 826 (I)
——— arrangement of compressing plant, 830 (I)
——— of pipes, 830 (I)
——— size of air main, 829 (I)
——— submergences, 829 (I)
— mains, 1078 (I)
——— loss of air due to friction, 757 (II)
— metering, 191 (I)
——— viscous flow, 201 (I)
— movement of, by fans, 681 (II), 1285 (II)
— petrol gas, 651 (II)
— preheating, 1226 (II), 1232 (II)
— pressure in pipes, 1079 (I)
— properties of, 56 (I)
— pumps, 178 (II), 1262 (II)
——— capacity of, 178 (II)
——— ejector or jet, 186 (II)
——— Weir ejector, 1262 (II)
— receivers, mine drill, 1078 (I), 889 (II)
— regulators, mining, 904 (II)
— required for combustion, 1361 (I), 18 (II)
——— gas, 1366, (I)
——— mine drilling, 882 (II)
——— ventilation, 675 (II)
— resistance of, to train, 526 (II)
— saturated, vapour in, 193 (II), 678 (II)
— thermal expansion, 63 (I)
— velocity and pressure of, 679 (II)
——— volume of, 679 (II)

Air velocity of sound through, 70 (I)
— vessels and alleviators, 78 (I)
— volume equivalent to free, 1079 (I)
— washing, 676 (II)
— water vapour in, 192 (II)
Aircraft (*see* Aeroplanes)
Airhydropump, Madan's, 1123 (II)
Airscrews, 401 (II)
—— Bernouilli's theorem, 405 (II)
—— characteristics of, 403 (II)
—— curves, 408 (II)
—— diameter required, 407 (II)
—— efficiency, 405 (II)
—— forces acting on blade, 406 (II)
—— Froude momentum theory, 401 (II)
—— high speed, 410 (II)
—— horse power, 408 (II)
—— inflow theory, 406 (II)
—— manufacture, 408 (II)
—— metal, 409 (II)
—— noise, 79 (I)
—— number of blades, 408 (II)
—— pitch, 402 (II)
—— slip, 402 (II)
—— speed, 408 (II)
—— theories, 401 (II)
—— thrust and torque, 405 (II), 410 (II)
—— variable pitch, 409 (II)
Albert canal, 602 (I)
Alclad, 1239 (I)
'Aldural,' 1171 (II)
Alkali hardeners for bearing metals, 1244 (I)
Alkalinity, caustic, of boiler water, 51 (II)
'Allautal,' 1235 (I)
Alldays & Onions, Ltd., power hammers, 1169 (II)
Allen's link motion valve gear, 100 (II)
Alleviators, hydraulic, 785 (I)
Alloys, 1140 (I), 1170 (II)
— acid-resisting, 1219 (I)
— adnic, 1220 (I)
— Alpax, 1232 (I)
— aluminium, 1225 (I), 1228 (I), 1251 (I)
—— casting, 1237 (I)
—— heat treatment and properties of, 1234 (I)
—— manganese, 1232 (I)
—— silicon, 1231 (I)
—— zinc, 1228 (I)
— A.M.F., 1218 (I)
— anti-friction, 1244 (I)
— bearing, 1242 (I), 1244 (I)
— beryllium, 1241 (I)
— bismuth-lead-tin, 1247 (I)
— brass, 1213 (I), 1249 (I)
— Bull's metal, 1174 (II)
— cast brass, 1213 (I)
— cemented tungsten-carbide, 1060 (I)
— copper, 1249 (I), 1253 (I)
—— and aluminium, 1225 (I), 1230 (I)
—— beryllium, 1227 (I)
—— cadmium, 1227 (I)
—— manganese, 1226 (I), 1251 (I)
—— nickel alloy 1226 (I), 1251 (I)
—— silicon, 1227 (I)
—— tin, 1250 (I)
—— zinc, 1212 (I)
Alloys, copper, nickel - aluminium, properties of work- and precipitation hardening of, 1235 (I)
— cutting tool, 1161 (I), 1253 (I)
— Delta brand, 1215 (I)
— die casting, 1264 (I)
— Dilver, 1218 (I)
— duralumin, 1233 (I), 1170 (II)
— electric resistance heating, 1220 (I)
— Elektron, 1240 (I)
— Elinvar, 1218 (I)
— eutectic, 1246 (I), 1252 (I)
— ferro, 1143 (I)
— fusible, melting points, 1246 (I)
— heat-resisting, 1219 (I), 1252 (I)
— Hiduminium RR., 1233 (I)
— Inconel, 1222 (I)
— iron nickel, 1217 (I)
— Kanthal, 1220 (I)
— lead base, 1243 (I), 1250 (I)
— Lipowitz's, 52 (I)
— lithium, 1241 (I)
— Magnalium, 1232 (I)
— magnesium, 1232 (I), 1239 (I)
— Magnuminium, 1240 (I)
— 'Melloid,' 1171 (II)
— melting-points of, 51 (I)
— Monel metal, 1220 (I)
— Mumetal, 1218 (I)
— nickel, 1217 (I)
—— barium, 1220 (I)
—— chromium, 1219 (I)
—— silver, 1218 (I)
— non-ferrous, 1212 (I), 1249 (I), 1254 (I), 1170 (II)
— non-magnetic, 1248 (I)
— Permalloy, 1217 (I)
— Permax, 1218 (I)
— physical properties of, 1253 (I), 1255 (I)
— Platinite, 1218 (I), 1220 (I)
— resistance, 1220 (I), 1379 (I)
— resistivities of, 1379 (I)
— 'RR,' 1233 (I)
— Smith No. 10, 1220 (I)
— specific gravities, 121
— steel specifications, 1161 (I), 1185 (I), 1187 (I)
— super hard cutting, 1161 (I)
— tin base, 1243 (I), 1250 (I)
— white metal, 1243 (I)
— Wood's, 52 (I)
— 'Y,' 1236 (I)
— zinc-aluminium, 1228 (I)
—— -tin, 1245 (I)
Alluvial dredging, 936 (II)
— mining, 862 (II) 932 (II)
Alpax alloys, 1232 (I)
Alternating-current motors, 1489 (I), 593 (II)
Altitude and rainfall, 660 (I)
— variation of engine power with, 412 (II)
Alumina, melting-point of, 371 (I)
Aluminium, 1206 (I)
— acid action on, 1207 (I)
— alloys, 1225 (I), 1228 (I), 1251 (I)
—— 'Allautal,' 1235 (I)
—— casting, 1237 (I)
—— heat treatment and properties of, 1234 (I)
—— 'Lautal,' 1235 (I)
—— sheet bending, 417 (II)
—— 'Silmalic,' 1235 (I)

Aluminium annealing sheet, 1207 (I)
——— anodic oxidation, 1239 (I)
——— bar and strip, weight and electrical value of, 133 (I)
——— brass, 1216 (I)
——— bronze, 1225 (I)
——— ——— die castings, 1226 (I)
——— ——— solder for, 1259 (I)
——— cables, steel-cored, 1405 (I), 1410 (I)
——— casting alloys, 1237 (I)
——— conductors, 1403 (I)
——— copper alloys, 1225 (I), 1230 (I), 1238 (I)
——— corrosion of, 1207 (I)
——— ——— protection, 1207 (I)
——— cylinder heads, 215 (II)
——— die quenching, 1235 (I)
——— extruded sections, 1238 (I)
——— flat bars, 133 (I)
——— foil for heat insulation, 68 (II)
——— influence on cast iron, 1148 (I)
——— manganese alloys, 1232 (I)
——— mechanical properties, 1207 (I)
——— nickel-silicon brass, 1217 (I)
——— overhead lines, 1411 (I)
——— paint, 1239 (I), 993 (II)
——— physical properties of, 1406 (I)
——— sheet, annealing, 1207 (I)
——— ——— weight of, 135 (I)
——— silicon alloys, 1232 (I)
——— soldering, 1259 (I), 1177 (II)
——— stress due to temperature, 177 (I)
——— strip, 133 (I)
——— tubes, weight of, 152 (I)
——— welding, 1280 (I)
——— zinc alloys, 1228 (I)
Amalgam solder, 1260 (I)
American high pressure boilers, 10 (II)
——— irrigation, 716 (I)
——— screw threads, 89 (I), 96 (I)
——— standard gas pipe threads, 97 (I)
——— ——— nuts and bolts, 99 (I)
——— ——— pipe flanges, 110 (I)
——— ——— screw threads, 89 (I), 96 (I)
——— wire gauge, 88 (I)
A.M.F. alloy, 1218 (I)
Ammeters, limits of errors in, 1507 (I)
Ammonia, properties of, 56 (I), 722 (II)
——— recovery, 1324 (I)
——— sulphate of, 644 (II)
Ammoniacal liquor gas, 635 (II)
Ammonium nitrate explosives, 950 (II)
Amsler traction dynamometer, 39 (I)
——— torsionmeter, 40 (I)
Amsterdam ship canal, 601 (I)
Anchor cables for suspension bridges, stresses in, 538 (I)
Anchors, rail, 485 (II)
Anderson, D., & Son, Ltd., 'Rok' and 'Flextile' roofing material, 1098 (II)
Anderton boat lift, 591 (I)
Anemometer, hot wire, 201 (I)
Angle and line bisector, 252 (I)
——— tee iron, 'Henderson & Glass,' 136 (I)
——— ——— steel ——— 137 (I)
——— iron, weight of, 136 (I)
——— steel, weight of, 137 (I)
Angles and lengths of slopes, 434 (I)
——— British standard, 243 (I)
Angles bulb, British standard, 244 (I)
——— cutting tool, 1062 (I)
——— of repose of soil, 435 (I)
——— ——— slopes, 431 (I)
——— roof, 301 (I), 302 (I)
Angus Smith's composition, 994 (II)
Animal power, 35 (I)
Animals, work done by, 35 (I)
Annealing steel, 1168 (I), 1192 (I)
——— sub-critical, 1168 (I)
——— wrought-iron chains, 176 (I)
Annual compass variation, 259 (I)
——— evaporation, 703 (I)
Anodic oxidation, 1239 (I)
Anthracite, 1327 (I), 856 (II)
——— wastes, 1329 (I)
Anti-freezing mixtures, 53 (I)
Anti-friction alloys, 1244 (I)
——— Delta metal, 1244 (I)
——— Fry's metal, 1244 (I)
——— grease, 995 (II)
Antimony, 1208 (I)
Antwerp, River Scheldt tunnel, 572 (I)
'Apexior' boiler compound, 1240 (II)
Apothecaries' measures, 6 (I), 8 (I)
Arbitrator, engineer as, 1033 (I)
Arbitrator's fees, 1033 (II)
Arc welding, 1286 (I), 1183 (II)
——— atomic hydrogen, 1289 (I)
——— cost of, 1289 (I)
——— equipment, 1287 (I), 1183 (II)
——— metallic, 1287 (I)
Arch bridges, 533 (I)
——— stresses, 534 (I)
——— temperature variation effect in, 536 (I)
——— dams, 683 (I), 697 (I), 698 (I)
Arched bridge, reinforced concrete, 409 (I)
Arches, 466 (I)
——— brick and masonry design, 467 (I)
——— centering for, 472 (I)
——— curve of least resistance, 469 (I)
——— eccentric loading, 471 (I)
——— fixed end, 410 (I)
——— keystones of, 468 (I)
——— reinforced concrete, 409 (I)
——— setting out, 466 (I)
——— stresses in, 410 (I), 583 (I)
——— thickness of, 468 (I)
——— to find depth of keystone of, 468 (I)
Architectural lighting, 707 (II)
Arcwall coalcutter, 1294 (II)
Area computation, Simson's rule for, 251 (I)
——— of circles, 1039 (II)
——— figures, 249 (I)
——— simple sections, 221 (I)
——— units of, 6 (I)
Areas, calculation of, from co-ordinates, 250 (I)
——— of regular and irregular figures, 249 (I)
Armature shafts, gearing on, 980 (I)
'Armco' ingot iron, 1141 (I)
Armengaud's screw thread, 95 (I)
Arromanches invasion harbour, 611 (I)
Artesian well pumps, 808 (I)
Articulated locomotives, 523 (II)
Artificial grindstones, 1099 (I)
——— marble, 358 (I)
——— stone, 358 (I)
Asbestos cement pipes, 353 (I)
——— roofing slates, 299 (I)

Asbestos fibre jointing, Turner Bros., 1258 (II)
——— sheets, 353 (I)
Ash emission from chimneys, 46 (II)
Ashlar masonry, 330 (I)
Ashwell & Nesbit, 'Iron Fireman' automatic stoker, 1228 (II)
Asphalt paving 564 (I)
Association, British, screw-thread, 90 (I), 93 (I)
Astronomy, 263 (I)
——— in field work, 264 (I)
Aswan Dam, 688 (I)
Atmosphere, standard, 411 (II)
——— temperature of, 54 (I)
Atomic hydrogen arc welding, 1289 (I)
——— weights, 120 (I)
Attrition test for road stones, 548 (I)
Audibility, limits of sound, 70 (I)
——— of signals at sea, 656 (I)
Audible sea signals, 653 (I)
Austenitic cast irons, 1145 (I)
——— stainless steels, 1200 (I)
Australian irrigation, 716 (I)
Automatic brakes, 578 (II)
——— Westinghouse, 578 (II)
——— CO_2 recorders, 1315 (I)
——— coal weighers, 206 (I)
——— metallic arc welding, 1288 (I)
——— refrigerating machines, 740 (II)
——— stokers, 1336 (I), 1220 (II), 1226 (II)
——— syphon spillways, 694 (I)
——— train control, 609 (II)
——— vacuum brake, 578 (II)
——— water weighers, 196 (I)
Automobiles, 321 (II), 1274 (II)
——— axles, 379 (II)
——— batteries, 1527 (I)
——— brakes for, 372 (II)
——— Bendix Cowdrey, 376 (II)
——— Girling, 374 (II)
——— Lockheed, 379 (II)
——— Servo systems, 374 (II)
——— stopping distance of, 373 (II)
——— clutches and flywheels, 356 (II)
——— coil ignition, 357 (II)
——— electric equipment of, 1527 (I)
——— engines, 329 (II), 340 (II)
——— balancing, 330 (II)
——— bearings, 331 (II)
——— camshaft drives, 336 (II)
——— carburettors, 350 (II)
——— clutches, 356 (II)
——— compression ignition, 340 (II)
——— fuel pumps, 343 (II)
——— ratio change effects, 323 (II)
——— connecting rods, 335 (II)
——— cooling, 198 (II), 346 (II)
——— crankshaft, 330 (II)
——— cylinder heads, 331 (II)
——— cylinders, 331 (II)
——— detonation in, 1352 (I)
——— efficiency of, 323 (II)
——— flywheels, 356 (II)
——— fuel, 327 (II)
——— distribution to cylinders, 349 (II)
——— pumps, 343 (II)
——— horse-power, 325 (II)
Automobiles, engines, horse-power, at various altitudes, 356 (II)
——— horse-power, rating formulæ of foreign countries, 326 (II)
——— ignition, 357 (II)
——— petrol types, 329 (II)
——— pistons, 335 (II)
——— power calculations, 324 (II)
——— supercharging, 347 (II)
——— thermodynamics, 323 (II)
——— valves, 332 (II)
——— springs, 333 (II)
——— timing, 334 (II)
——— frames, 322 (II), 369 (II)
——— gear boxes, 365 (II)
——— gears 363 (II)
——— lighting, 1528 (I)
——— lubrication of, 380 (II)
——— power required to negotiate gradients, 324 (II)
——— radiator cleanser, 1239 (II)
——— road regulations, 554 (I)
——— rolling resistance of roads, 324 (II)
——— springs for, 365 (II)
——— steering gear, 371 (II)
——— suspension, 365 (II)
——— weights of, 369 (II)
Aviation fuel, 452 (II)
Avonmouth docks, 614 (I)
Axial clutches, 929 (I)
Axial-flow pumps, 809 (I)
Axle boxes, railway, 576 (II)
Axles, automobile, 379 (II)
——— crank, of locomotives, 540 (II), 550 (II)
——— grease for, 995 (II)
——— straight, of locomotives, 550 (II)
Ayrton Perry torquemeter, 40 (I)
Azimuth, determination of, 266 (I), 271 (I)
——— graphic method of finding from sun or star, 271 (I)

B.Th.U. (British Thermal Unit), 54 (I)
Babbitt metal, 1243 (I)
Back pressure and extraction engines, 82 (II)
——— turbines (steam), 111 (II)
Bailey & Bowers' torquemeter, 41 (I)
Baking ovens, electric, 1522 (I)
Balanced draught in boilers, 16 (II)
Balancers, static, 1502 (I)
Balata belting, 910 (I)
Balk brake, 42 (I)
——— dams, 448 (I)
Ball bearings, 849 (I), 1113 (II)
——— capacity, 855 (I)
——— high temperature, 864 (I)
——— Hoffmann, 1140 (II)
——— housings, 860 (I)
——— journal, 850 (I)
——— lubrication of, 862 (I)
——— permissible loadson, 855 (I)
——— preloading, 864 (I)
——— Ransome & Marles, 1142 (II)
——— Skefko, 1146 (II)
——— thrust, 854 (I), 1135 (II)
——— cast iron, weight of, 127 (I)
——— mills, Krupp, gold mining, 931 (II)

Ballast, 485 (II)
——— **tamper, pneumatic**, 491 (II)
Balls, cast iron, weight of, 127 (I)
——— **for Brinell tests**, 1092 (I)
——— **metal, weight of, 127** (I)
Band clutches, 932 (I)
Bandsaws, 1039 (I)
Bar aluminium, weight of, 133 (I)
—— **brass, weight of, 131** (I)
—— **cast iron, weight of, 127** (I)
—— **copper, weight of, 131** (I)
—— **iron, tests for, 167** (I)
—— **lead, weight of, 131** (I)
—— **metals, weight of, 127** (I)
—— **steel, tests for, 166** (I)
Barff process for protecting iron, 1194 (I)
Barges, hopper, 638 (I)
Barimar Ltd., welding repairs, 1180 (II)
Barnes formula for weirs, 746 (I), **758** (I)
Barometer, height of, in millimetres, 13 (I)
Barrels for wire ropes, 939 (I)
Barrow and cart runs, 432 (I)
Bars, fire, 1333 (I), 291 (II)
——— **steel, Admiralty tests, 166** (I)
——— **T, 245** (I)
Bas Serani lighthouse, 653 (I)
Bascule bridges, 526 (I)
Base line, to measure, across a river, 251 (I)
Basic bricks, 372 (I)
——— **economy of dams, 698** (I)
Baths, tempering, for cutting tools, 1072 (I)
Batteries, automobile, 1527 (I)
———— **care of storage**, 1480 (I)
———— **nickel iron**, 1480 (I)
———— **primary**, 1476 (I)
———— **signalling and wireless**, 1480 (I)
———— **tests for impurities**, 1480 (I)
Battery charging, 385 (II)
———— **vehicles**, 384 (II)
Bauxite bricks, 371 (I)
Bazin's formula for channels for irrigation, 679 (I), **758** (I)
Beams, 211 (I), **241** (I), **311** (I), **393** (I)
——— **bending moments of continuous, 223** (I)
——— **building up, 312** (I)
——— **continuous, 223** (I)
——— **deflection of, 223** (I)
———————— **reinforced concrete, 405** (I)
——— **diagonal tension in, 404** (I)
——— **fishing, 311** (I)
——— **influence values, 214** (I), **218** (I)
——— **internal stresses, 215** (I)
——— **inverted concrete, 404** (I)
——— **jointing, 311** (I)
——— **mortise, 313** (I)
——— **rectangular, solid, 222** (I)
——— **reinforced concrete, 303** (I), **399** (I), **404** (I)
——— **safe loads on, 242** (I)
——— **scarfing, 311** (I)
——— **shouldered, 313** (I)
——— **straining-joint of, 315** (I)
——— **strength and stiffness of, 222** (I), **223** (I)
——— **stresses on, 211** (I), **222** (I)
——— **T, 312** (I), **402** (I)
——— **working stresses, 222** (I)
Bearers, rail and road, 513 (I)
Bearing capacity, 855 (I)
——— **metals, 1223** (I), **1242** (I), 462 (II)
——— **pressures, 866** (I), **963** (I), **249** (II)
Bearing **propeller shafts, 1272** (II)
——— **surfaces, locomotive**, 510 (II)
Bearings, **ball, 849** (I), 1131 (II)
———— Bound Brook, 1138 (II)
———— **bridge, 503** (I)
———— British Timken, 1131 (II)
———— **bronze, oil retaining**, 1138 (II)
———— **capacity, 855** (I)
———— **cooling compound for hot**, 996 (II)
———— **crane**, 810 (II)
———— **dimensions, 866** (I)
———— **engine**, 249 (II)
———— **graphited**, 1139 (II)
———— **high speed, 863** (I)
———— Hoffman, 1140 (II)
———— **housings, 860** (I)
———— **journal, 850** (I), **851** (I), **855** (I)
———— **lineshaft, 854** (I)
———— **locomotive**, 510 (II)
———— **lubrication of, 862** (I), **869** (I), **878** (I)
———— **metal, 1223** (I), **1245** (I)
———— Michell, 302 (II)
———— **motor-car engine**, 322 (II)
———— **needle, 853** (I)
———— **'Oilite,'** 1141 (II)
———— **oil retaining bronze**, 1138 (II)
———— **permissible load on, 866** (I)
———— **preloading, 864** (I)
———— **pressures, 866** (I)
———— **provision for thrust, 858** (I)
———— Ransome and Marles, 1142 (II)
———— Rigby's patent, **505** (I)
———— **roller, 849** (I), 1131 (II)
———————— **for bridges, 504** (I)
———————— **railway rolling stock**, 577 (II)
———— **self-lubricating bronze**, 1111 (II)
———— Skefko, 1146 (II)
———— **steam engine**, 87 (II)
———————— **turbine**, 133 (II)
———— **swing bridge, 527** (I)
———— **taper roller, 853** (I), 858 (I), 1131 (II)
———— Timken taper roller, 1131 (II)
———— **thrust, 854** (I), **858** (I), 1135 (II)
———— **tolerances for shafts and housings, 856** (I)
———— **turbine (steam)**, 133 (II)
Beckmann **thermometer, 1136** (I)
Bedplates **and bearings, bridge, 503** (I)
Bell **buoys, 645** (I)
——— **metal, 1223** (I)
Bellmouth **weirs and spillways, 694** (I) **751** (I)
Bells, **submarine, 656** (I)
Belt **conveyors**, 840 (II), 1294 (II), 1302 (II)
——— **driving, design, 913** (I)
——— **grease**, 996 (II)
——— **pulleys, 913** (I)
——— **tension, 915** (I)
Belts, **909** (I), 1152 (II)
——— **balata, 910** (I), 1152 (II)
——— **canvas, 910** (I)
——— **chain, 961** (I)
——— Dick, R. & J., Ltd., 1152 (II)
——— **efficiency, 914** (I)
——— **leather, 909** (I)
———————— **oiling and greasing**, 996 (II)
——— **length of, 914** (I)
——— **power transmitted by, 910** (I)
——— **short centre drives, 915** (I)
——— **solid woven, 909** (I)
——— **spring driving, 920** (I)

Belts, tensioning devices, 915 (I)
—— vee, 916 (I)
Bending moments, 216 (I), 225 (I), 509 (I), 531 (I), 893 (I)
Bendix Cowdrey brake system, 376 (II)
Bends for steam pipes, 60 (II)
—— corrugated steel, 60 (II)
Bennis Combustion Ltd., Air-Draught Stoker, 1220 (II)
—— Self-cleaning furnace, 1220 (II)
Benzine, flash temperature, 53 (I)
Benzol recovery, 1324 (I), 638 (II)
Bernouilli's theorem, 405 (II)
Bertand process, 1194 (I)
Beryllium, 1208 (I)
—— alloys, 1227 (I), 1241 (I)
Bevel gearing, 987 (I)
Bevis-Gibson torsionmeter, 39 (I)
Bibby couplings, 1269 (II)
Bi-cable ropeways, 833 (II)
Bigelow suspended furnace wall, 1236 (II)
'Bijur' lubrication system, Tecalemit Ltd., 1150 (II)
Bins, reinforced concrete, 405 (I)
Birchenough Bridge, 574 (I)
Birmingham gauge for sheets and hoops, 86 (I)
—— wire gauge, 85 (I), 87 (I)
Bisecton, line and angle, 252 (I)
Bismuth, melting point of alloy, 51 (I), 1247 (I)
—— solder, 1259 (I)
Bituminous cement, 353 (I)
—— materials, tests of, 549 (I)
—— —— experimental determinations, 550 (I)
—— paint, 981 (II)
Black patent grease, 996 (II)
—— powder or gunpowder, 949 (II)
Blade vibration on steam turbines, 149 (II)
Blading steam turbine, 114 (II), 124 (II)
—— materials for, 152 (II)
Blast-furnace Portland cement, 356 (I)
Blast furnaces, 1126 (I)
Blasting, 641 (I), 862 (II), 951 (II)
—— explosives, 862 (II), 951 (II)
—— rock, subaqueous, 641 (I)
—— sand, 1115 (I)
Blau gas, 646 (I)
Block clutches, 931 (I)
Block-in-Course masonry, 330 (I)
Blocks, abrasive, 1099 (I)
—— and terra-cotta, 369 (I)
—— concrete, 350, 352 (I)
—— furnace lining, 373 (I)
—— hollow, 369 (I)
Blowpipe welding, acetylene, 1274 (I)
Blue bricks, 364 (I)
—— patent grease, 996 (II)
Board of Trade regulations for electric tramways, 382 (II)
—— rules for safety valves, 56 (II)
Bodies, falling, 25 (I)
—— impact of, 25 (I)
—— rotating, energy of, 27 (I)
Boiler 'Apexior' compound, 1240 (II)
—— blocks, 369 (I)
—— central heating, 1217 (II)
—— chimney and flue dimensions, 13 (II), 44 (II), 667 (II)
—— —— emission of dust and ash from, 46 (II)
—— —— losses, 21 (II)
Boiler CO_2 Indicator, 1193 (II)
—— combustion and draught rate, 24 (II)
—— —— chambers, iron for screw stays of, 35 (II)
—— —— formulæ, 22 (II)
—— compounds, 18 (II), 1239 (II)
—— construction, 32 (II)
—— control, automatic, 15 (II)
—— corrosion, 1239 (II)
—— coverings, 68 (II)
—— dampers, 12 (II)
—— density tester, 1257 (II)
—— draught, 16 (II)
—— —— effect of altitude on, 23 (II)
—— —— for different fuels, 24 (II)
—— economisers, 17 (II)
—— efficiency, 20 (II)
—— feed and service pumps, 814 (I), 1557 (I), 1121 (II), 1127 (II)
—— —— water heaters, 1218 (II)
—— —— —— caustic alkalinity of, 51 (II)
—— —— —— conditioning, 52 (II)
—— —— —— oil elimination from, 53 (II)
—— —— —— pump, 1121 (II)
—— —— —— —— governor, 1244 (II)
—— —— —— —— regulators, 1243 (II), 1245 (II), 1247 (II)
—— —— —— treatment of, 18 (II), 1238 (II)
—— feeding, turbine pumps for, 814 (I), 1129 (II)
—— fittings, 54 (II), 1241 (II)
—— flue tubes, 33 (II)
—— flues, dimensions of, 13 (II), 44 (II)
—— fuel, coke as, 1326 (I)
—— —— calorific value of, 20 (II)
—— furnace linings, Bigelow, 1234 (II)
—— furnaces, 1407, 33 (II), 1220 (II)
—— —— linings, 1236 (II)
—— —— marine, 38 (II)
—— —— radiant heat transmission in, 25 (II)
—— —— stays, 35 (II), 39 (II)
—— gauge glasses, 58 (II)
—— gauges, 58 (II), 1241 (II)
—— graphite, 'Foliac,' 1240 (II)
—— heat balance, 1295 (I)
—— —— losses, 20 (II)
—— —— transfer in cross flow, 32 (II)
—— —— transmission, 24 (II)
—— heaters, tubular, 28 (II)
—— houses for heating installations, 666 (II)
—— injectors, 54 (II)
—— inspection, 53 (II)
—— lagging, 69 (II)
—— legislation, 54 (II)
—— linings, 1237 (II)
—— plants, high pressure, 10 (II)
—— plates, caustic embrittlement of, 51 (II)
—— —— detection of cracks in, 33 (II)
—— —— transmission of heat through, 25 (II)
—— power, 663 (II)
—— pressure required for pumping, 806 (I)
—— reducing valves, 59 (II)
—— riveting, pressure required for, 33 (II)
—— rules and regulations, 33 (II)
—— safety valves, 41, 55, 1249 (II)
—— scale, 48 (II)
—— setting, 11 (II)
—— settings, leaky, 12 (II)

Boiler shells, 32 (II)
——— **strength of,** 32 (II)
—— **stays,** 35 (II), 39 (II), 40 (II)
—— **steam pipes,** 60 (II)
—— **steel,** 163 (I), 32 (II)
—— **stokers,** 1336 (I), 1220 (II), 1226 (II)
—— **superheaters,** 16 (II), 296 (II), 1229 (II), 1278 (II)
—— **tube cleaners,** 1256 (II)
—— **tubes,** 25 (II), 41 (II)
——— **charcoal iron, lapwelded,** 42 (II)
——— **Fry's formulæ,** 25 (II)
——— **heat transmission through,** 25 (II), 31 (II)
——— **iron,** 41 (II)
——— **locomotive,** 42 (II)
——— **surface of,** 42 (II)
——— **weight of,** 41 (II)
—— **water, caustic alkalinity of,** 51 (II)
——— **de-aeration system,** 1228 (II)
——— **gauges,** 57 (II), 1241 (II)
——— **hardness of, soap test,** 19 (II)
——— **softening,** 49 (II), 52 (II)
———— **lime and soda process,** 49 (II)
——— **treatment,** 48 (II), 1238 (II)
Boilerine, 1239 (II)
——— **radiator cleanser,** 1239 (II)
Boilers, 9 (II), 293 (II), 1201 (II)
—— **air heaters for,** 17 (II)
——— **required for combustion,** 18 (II)
—— **Babcock & Wilcox,** 298 (II)
—— **central heating, McNeil,** 1217 (II)
—— **Clarke-Chapman,** 1201 (II)
—— **Cochran,** 1206 (II)
——— **'Sinuflo,'** 1209 (II)
—— **Cornish,** 13 (II)
—— **corrosion in, preventives,** 1239 (II)
—— **cylindrical,** 36 (II), 293 (II)
—— **cylindrical, length of,** 294 (II)
——— **marine,** 36 (II), 293 (II)
———— **diameter of tubes of,** 291 (II)
———— **heating surface,** 291 (II)
———— **superheaters for,** 296 (II)
———— **thickness of shells,** 294 (II)
—— **'Economic,'** 13 (II)
—— **effect of altitude on draught of,** 23 (II)
—— **evaporation of,** 1206 (II)
—— **exhaust gas,** 1208 (II), 1214 (II)
—— **forced circulation,** 15 (II)
—— **gas fired,** 1211 (II)
—— **heating for buildings,** 663 (II), 672 (II), 1217 (II)
——— **efficiency of,** 663 (II)
—— **high pressure American,** 10 (II)
—— **Johnson,** 298 (II)
—— **La Mont,** 15 (II)
—— **Lancashire,** 11 (II)
—— **life of,** 298 (II)
—— **locomotive,** 541 (II)
——— **fire boxes,** 546 (II)
——— **grate areas and heating surfaces of,** 515 (II)
——— **rivets for,** 558 (II)
——— **specification of plates for,** 557 (II)
——— **steel castings for,** 554 (II)
Boilers, locomotive, steel **tubes for,** 42 (II), 556 (II), 545 (II)
——— **superheaters,** 547 (II), 1278 (II)
——— **tubes for,** 42 (II), 545 (II), 556 (II)
—— **Loeffler,** 15 (II)
—— **Lopulco,** 1188 (II)
—— **marine,** 34 (II)
——— **cylindrical type,** 36 (II), 293 (II)
———— **length,** 294 (II)
———— **superheater,** 296 (II)
———— **thickness of shells,** 37 (II), 294 (II)
——— **draught of,** 293 (II)
——— **furnaces and firebars of,** 38 (II), 294 (II)
——— **heating surface of,** 291 (II)
——— **length of,** 294 (II)
——— **life of,** 298 (II)
——— **materials of construction,** 34 (II)
——— soot blowers, 1234 (II)
——— **steel for, 163** (I)
——— **valves,** 41 (II)
——— **water tube,** 295 (II), 297 (II)
——— **weight of,** 277 (II)
——— **working pressure, rules for determining,** 37 (II)
—— **McNeil central heating,** 1217 (II)
—— **mechanical stokers, 1336** (I), 1220 (II)
——— **draught,** 1337 (I)
—— **power station,** 10 (II)
—— **preventives of corrosion in,** 1239 (II)
—— **pulverised coal fired,** 1188 (II)
—— **rating of,** 10 (II)
—— **'Sinuflo,'** 1210 (II)
—— soot blowers for, 1234 (II)
—— **steel for, 163,** (I), 32 (II)
—— **transmission of heat in,** 24 (II)
—— **vertical, 13** (II)
—— **waste heat,** 625, (II) 1213 (II)
—— **water-tube for land service,** 14, (II), 1201, (II), 1217 (II)
——— **marine service,** 297(II)
——— **stokers,** 1220 (II)
——— **tubes for,** 43 (II)
—— **welded,** 33 (II)
—— **Yarrow,** 1217 (II)
Boiling-points of saturated solutions of salts, 52 (I)
——— **water, 52** (I)
Bolt heads, British standard hexagonal, 102 (I)
——— **round,** 104 (I)
——— **weight of, 154** (I)
—— **spacing,** 107 (I)
Bolts and nuts, 98 (I)
——— **American, 99** (I)
——— **British standard,** 99 (I)
——— **French standard, 101** (I)
——— **safe load on,** 100 (I)
——— **Whitworth's, 98** (I), 100 (I)
—— **connecting rod,** 209 (II)
—— **round-headed,** 104 (I)
—— **safe load on,** 100 (I)
Bombay harbour and docks, dredging, 639 (I)
Bonds for brickwork, 317 (I)
——— **masonry, 331** (I)
Booster transformers, 1503 (I)

Booth, J., & Co.'s 'Duralumin,' 1170 (II)
Bore-holes, 862 (II)
——— pumps, 809 (I)
Boring (mining), 861 (II)
——— core, 861 (II)
——— diamond rocks, 861 (II)
Boulder dam, 691 (I)
Bound Brook Bearings, 1138 (II)
Bower-Barff process, 1194 (I)
Box gates for docks, 619 (I)
—— keys, 105 (I)
—— spanners, 105 (I)
Boxes for pipe flanges, 68 (II)
Boyles' Law, 189 (II), 323 (II)
Brackett, F. W., & Co., Ltd., circulating water screens, 1263 (II)
Bradford steam trap, 1254 (II)
Brake horse-power, 75 (II), 755 (II)
—— linings, Ferodo, 1153 (II)
—— rims, cooling, 43 (I)
Brakes, 578 (II)
——— automatic, vacuum, 583 (II)
——— ——— Westinghouse, 579 (II)
——— automobile, 372 (II)
——— ——— stopping distance, 373 (II)
——— crane, 810 (II)
——— ——— Weston washer, 811 (II)
——— electro-mechanical, 1198 (II)
——— locomotive, 563 (II)
——— motor car, 372 (II)
——— power measurement, 41 (I)
——— ——— Brotherhood fluid friction, 43 (I)
——— ——— fan, 44 (I)
——— ——— Froude's, 43 (I)
——— ——— Hirn's rope, 41 (I)
——— ——— Prony, 42 (I)
——— railway, 578 (II)
——— ——— power required by, 579 (II)
——— regenerative and rheostatic, 382 (II)
——— Brakes, tramway, 372 (II)
——— vacuum automatic, 583 (II)
——— Westinghouse automatic, 579 (II)
Brass, aluminium, 1216 (I)
——— ——— effect in, 1216 (I)
——— ——— nickel-silicon, 1217 (I)
——— bars, weight of, 131
——— cast, 1213 (I)
——— ——— alloys, 1213 (I)
——— ductile, 1213 (I)
——— effect of other elements in, 1215 (I)
——— hot-forgeable, 1214 (I)
——— ——— pressings, 1214 (I)
——— iron, effect in, 1215 (I)
——— lead, effect in, 1215 (I)
——— manganese, effect in, 1216 (I)
——— naval, 1216 (I)
——— nickel, 1218 (I)
——— ——— effect in, 1216 (I)
——— pipes, weight of, 141 (I)
——— properties of, 1213 (I)
——— rolled, weight of, 131 (I)
——— screw, thread British standard 93 (I)
——— sheets and bars, weight of, 131 (I)
——— stress due to temperature, 177 (I)
——— tin, effect in, 1213 (I)
——— tubes for locomotive boilers, 556 (II)
——— ——— seamless, weight of, 151 (I)
——— welding, 1281 (I)
——— wire, weight of, 154 (I)
Brasses, 1249 (I)
Braziers' copper rivets, weight of, 156 (I)
Brazing or spelter solders, 1258, (I), 1179 (II)
Breakdown cranes, 806 (II)
Breakers, rock, 642 (I)
Breaking stresses of annealed and bright iron wire, 162 (I)
——— ——— hard drawn telegraph copper wire, 163 (I)
——— ——— wire and wire ropes, 162 (I)
Breakwaters, 605 (I)
——— construction of, 608 (I), 611 (I)
——— cost of, 606 (I)
——— Genoa, 609 (I)
——— mound, 608 (I)
——— wall, 609 (I)
——— wave action on, 607 (I)
Breathing apparatus, 'Proto,' 1299 (II)
Breeze concrete, 343 (I)
Brent reservoir, 694 (I)
Bretland track-laying machine, 491 (II)
Brett's Patent Lifter Co., Ltd., drop forging equipment, 1166 (II)
Brick arches, 467 (I)
—— chimney shafts, 464 (I)
—— proportions of, 46 (II)
—— paving data, 562 (I)
—— retaining walls, 460 (I)
—— roads, 562 (I)
—— walls, safe loads on, 322 (I), 437 (I)
Bricks, 180 (I), 316 (I), 364 (I)
——— bauxite, 371 (I)
——— blue, 364 (I)
——— building specification for, 365 (I)
——— cement-concrete, 364 (I)
——— chromite, 371 (I)
——— chuff, 364 (I)
——— colliery, 364 (I)
——— common, 364 (I)
——— concrete, 367 (I)
——— crozzle, 364 (I)
——— crushing strength of, 180 (I), 365 (I)
——— cutter, 364 (I)
——— engineering, 364 (I)
——— estimating numbers, 316 (I)
——— facing, 364 (I)
——— fire, 371 (I)
——— fireclay, specification for, 372 (I), 377 (I)
——— furnace, 373 (I)
——— glazed, 364 (I)
——— grizzle, 364 (I)
——— Hollow, 369 (I)
——— magnesia, 371 (I)
——— malm, 364 (I)
——— melting point of, 371 (I)
——— neutral, 372 (I)
——— panel, 364 (I)
——— paving, number required, 316 (I)
——— paviour, 364 (I)
——— place, 364 (I)
——— porosity of, 366 (I)
——— pressed, 364 (I)
——— rattler, test for, 366 (I)
——— rubbers, 364 (I)
——— sand-lime, 364 (I), 366 (I)
——— semi-bauxite, 371 (I)
——— semi-dry, 364 (I)
——— semi-plastic, 364 (I)
——— silica, 371 (I)
——— ——— specification for, 373 (I)

Bricks, sizes of, 365 (I)
——— specification for, 365 (I)
——— Staffordshire, 364 (I)
——— stiff plastic, 364 (I)
——— stock, 364 (I)
——— strength of, 180 (I)
——— tests of, 366 (I)
——— wire-cut, 364 (I)
Brickwork, 180, 316 (I)
——— bonds, 317 (I)
——— estimating, 316 (I)
——— of wells, 738 (I)
——— strength of, 180 (I)
Bridge welding, 1291 (I)
Bridge-meg testing set, 1195 (II)
Bridges, 477 (I) (*see also* Arches)
——— arch, 533 (I)
——— arched, reinforced concrete, 409 (I)
——— bascule, 526 (I)
——— bearers, road and rail, 513 (I)
——— bearings, 505 (I)
——— ——— Rigby's expansion rocker, 505 (I)
——— bedplates, 503 (I)
——— British standard steel for, 163 (I)
——— camber, 508 (I)
——— cantilever plate girder, 515 (I)
——— centrifugal force on, 499 (I)
——— Clyde, 489 (I)
——— concrete, 571 (I)
——— cross girders of, 512 (I)
——— deformation and deflection of, 514 (I), 523 (I)
——— economical span of, 478 (I)
——— expansion bearings, 503 (I)
——— flange plate curtailment, 513 (I)
——— ——— breadth of, 479 (I)
——— flooring of, 506 (I)
——— foundations of, safe loads on, 489 (I)
——— ——— ——— skin friction on, 489 (I)
——— girder, cantilever plate, 515 (I)
——— ——— specification for, 497 (I)
——— girders, circular, 531 (I)
——— ——— cross, 512 (I)
——— ——— economical proportions of, 478 (I)
——— ——— weights of, 492 (I)
——— highway, standard loadings for, 482 (I), 485 (I), 488 (I)
——— impact in, 500 (I)
——— lattice girder, 517 (I)
——— ——— girders of, 479 (I)
——— ——— load on, 517 (I)
——— ——— pin-jointed members, 520 (I)
——— ——— stresses on, 518 (I)
——— live loads on, 481 (I), 509 (I), 517 (I)
——— loadings on, 482 (I)
——— lubrication of swing, 530 (I)
——— piers of, 472 (I)
——— plate-girder, 478 (I), 509 (I)
——— portal bracing of, 521 (I)
——— rail and road bearers of, 513 (I)
——— railway, standard clearances, 479 (I)
——— ——— standard loadings, 482 (I), 485 (I)
——— reinforced concrete, arches of, 409 (I)
——— riveting, 508 (I)
——— road, bearers, 513 (I)
——— ——— standard clearances, 479 (I)
——— rolling load on, 481 (I)
Bridges, spans and standard clearances of, 479 (I)
——— standard loadings, 482 (I)
——— steel, 163 (I), 509 (I)
——— Stress Committee's recommendations for impact effect, 502 (I)
——— stress on, 493 (I), 503 (I), 517 (I), 533 (I), 537 (I)
——— suspension, 537 (I)
——— ——— self anchored, 539 (I)
——— swing and bascule, 525 (I)
——— ——— ——— bearings, 527 (I)
——— ——— ——— lubrication of, 529 (I)
——— ——— ——— pivots, 528 (I)
——— ——— ——— power required, 531 (I)
——— ——— ——— rollers, 523 (I)
——— temperature effect on, 503 (I)
——— ties, 520 (I)
——— train momentum on, 500 (I)
——— transporter, 527 (I)
——— types of, 477 (I)
——— waterway of, 490 (I)
——— webs and stiffeners, 509 (I)
——— weights of, 492 (I)
——— wind pressure on, 497 (I)
——— working stresses on, 493 (I)
Bridgework, British practice, 497 (I)
——— weights of, 492 (I)
Brightness (light), 686 (II)
Brine calcium, 709 (II)
Brinell hardness numbers, 1089 (I), 1090 (I), 1190 (I), 1254 (I)
Briquette fuel, 1330 (I)
——— coals and pitches for, 1331 (I)
——— process and plant, 1332 (I)
British Admiralty specification for oil for Diesel engines, 1362 (I)
——— ——— specification for oil fuel for the Navy, 1361 (I)
——— and Allied Manufacturers' Association, specification for turbine forgings, 151 (II)
——— Association screw threads, 93 (I)
——— bridgework practice, 497 (I)
——— canals, 585 (I)
——— horse power, 187 (II)
——— Isles, rainfall in, 659 (I)
——— locomotives, recent additions, 515 (II)
——— ——— types of, 522 (II)
——— Mesozoic ironstones, 859 (II)
——— Paints, Ltd., 'Apexior' boiler compound, 1240 (II)
——— railway rails, dimensions of, 472 (II)
——— Standard abrasive cloths and papers, 1114 (I)
——— ——— accumulators, 1476 (I)
——— ——— aluminium conductors, 1403 (I)
——— ——— angles, 243 (I)
——— ——— asbestos cement pipes, 353 (I)
——— ——— ——— sheets, 353 (I)
——— ——— boiler tubes, 41 (II)
——— ——— bolt-heads, 102 (I)
——— ——— bolts and nuts, 99 (I)
——— ——— braided copper overhead lines, 1409 (I)
——— ——— brass thread, 93 (I)
——— ——— ——— tubes, 556 (II)
——— ——— bricks, 365 (I)

British Standard bridge steel, 163 (I)
—— Brinell hardness numbers, 1091 (I)
—— building construction steel, 163 (I), 222 (I), 227 (I)
—— bullhead railway rails, 472 (II)
—— carbon and alloy steel, 1161 (I)
—— —— brushes, 1492 (I)
—— cast-iron classification, 1143 (I)
—— —— hydraulic power pipes, 786 (I)
—— —— water, gas, and sewage pipes, 114, (I) 147 (I)
—— castle nuts, 102 (I)
—— channels, 245 (I)
—— charcoal-iron lap-welded boiler tubes, 42 (II)
—— clay or marl plain roofing tiles, 367 (I)
—— compressor tests, 763 (II)
—— concrete flags, 352 (I)
—— —— kerbs, channels and quadrants, 350 (I)
—— —— pipes and tubes, 352 (I)
—— —— roofing tiles, 353 (I)
—— condenser tubes, etc., for marine purposes, 170 (II)
—— conduits, electrical wiring, 1419 (I)
—— copper conductors, 1380 (I)
—— —— plates, 555 (II)
—— —— rods, 555 (II)
—— creosoting of timber, 288 (I)
—— derrick cranes, 805 (II)
—— drills, 1057 (I)
—— electric clocks, 1529 (I)
—— —— equipment for automobiles, 1527 (I)
—— —— fans, 1483 (I)
—— —— lamps, 708 (II)
—— —— measuring instruments, 1506 (I)
—— —— traction, 585 (II)
—— electrical apparatus, 1150 (I)
—— —— machinery, 1180 (I)
—— —— motors and generators, 1182 (I)
—— electrical units and nomenclature, 1377 (I)
—— electricity meters, 1512 (I)
—— enamelled wire, 1112 (I)
—— field rheostats, 1500 (I)
—— fine screw threads, 92 (I)
—— fire extinguishers, 709 (II)
—— hose couplings, 712 (II)
—— fishbolts and nuts, 480 (II)
—— fishplates, 479 (II)
—— fractional h.p. motors, 1482 (I)
—— fuel oils, 1362 (I)
—— fuses, 1113 (I)
—— gas pipes, 114 (I)
—— girder bridges, 495 (I), 497 (I)
—— graphic instruments, 1511 (I)
—— hardness scales, 1094 (I)
—— heads for small screws, 93 (I)
British Standard hexagonal nuts and bolt heads, 102 (I)
—— high tensile structural steel, 496 (I)
—— hooks, 937 (I)
—— impact effect on bridges, 501 (I)
—— insulating materials, 1525 (I)
—— —— oils, 1525 (I)
—— insulators (porcelain), 1413 (I)
—— internal combustion engines, 249 (II)
—— keys, keybars and keyways, 117 (I)
—— lighting fittings, 708 (II)
—— limes, 340 (I)
—— loadings on railway and highway bridges, 482 (I), 485 (I)
—— locomotive axles, 550 (II)
—— —— materials, 550 (II)
—— —— springs, 551 (II)
—— —— steel forgings, 554 (II)
—— —— tubes, 42 (II)
—— —— tyre contours, 542 (II)
—— —— tyres, 550 (II)
—— measuring instruments, 1506 (I)
—— metallic resistance materials, 1379 (I)
—— motor starters and controllers, 1197 (I)
—— motors and generators, 1182 (I)
—— nuts and bolts, 99 (I)
—— paints, 980 (II), 984 (II)
—— pipe flanges, 108 (I)
—— —— threads, 98 (I)
—— poles for transmission lines, 1111 (I)
—— porcelain insulators, 1413 (I)
—— portable photometers, 708 (II)
—— Portland cement, 355 (I)
—— —— blast furnace, 356 (I)
—— primary cells, 1476 (I)
—— pump tests, 747 (I)
—— rails, 172 (II)
—— railway rolling stock material, 550 (II)
—— rheostats, 1500 (I)
—— rivets, 230 (I)
—— road materials, 551 (I)
—— tars, 553 (I)
—— rolled steel sections, 239 (I)
—— sand lime bricks, 366 (I)
—— screw heads, 94 (I)
—— —— threads, 90 (I)
—— screwed and socketed tubes, 146 (I)
—— sections, 239 (I)
—— sewer pipes, 114 (I), 116 (I), 769 (II)
—— shank hooks, 937 (I)
—— spanners, 105 (I)
—— steam turbines, 159 (II)

British Standard steel beams, working stresses, 222 (I)
—— castings, 554 (II)
—— electrical conduits, 1119 (I)
—— spigot and socket pipes, 116 (I)
—— tubes, 111 (I)
—— for bridges and buildings, 163 (I), 496 (I)
—— building construction, 163 (I), 222 (I), 227 (I)
—— general engineering (Schedule 970 En Series), 1171 (I)
—— marine boilers, 163 (I)
—— rails, 472 (II)
—— shipbuilding, 164 (I)
—— street lighting, 708 (II)
—— switch gear, 1195 (I)
—— taper pins, 117 (I)
—— tensile test pieces, 35 (II)
—— tiles, 367 (I)
—— tooth form for bevel gears, 989 (I)
—— traction motors, 593 (II)
—— transformers 1500 (I)
—— instrument, 1509 (I)
—— tubes screwed and socketed, 112 (I)
—— turbine oils, 880 (I)
—— twist drills, 1057 (I)
—— voltages for transmission and distribution systems, 1379 (I)
—— water pipes, 114 (I)
—— weights of stones, 335 (I)
—— Whitworth Castle and slotted nuts, 102 (I)
—— screw threads, 90, 91 (I)
—— wire gauge, 85 (I)
—— wiring fittings, 1418 (I)
—— working stresses for steel beams, 222 (I)
—— thermal unit (B.Th.U.), 54 (I), 187 (II)
—— Timken Ltd., taper roller bearings, 1131 (II)
Brittleness of chains at low temperatures, 176 (I)
Broaches and broaching, 1050 (I)
Bronze, aluminium, 1225 (I)
—— bearings, Bound Brook, 1138 (II)
—— 'Oilite' self-lubricating, 1141 (II)
—— 'Delta,' 1215 (I)
—— ductile, 1223 (I)
—— effect of certain metals on, 1223 (I)
—— lead, 1223 (I)
—— manganese, 1226 (I)
—— 'Melloid,' 1174 (II)
—— melting-point of, 1225 (I)
—— phosphor, 1224 (I)
—— silicon, 1225 (I)
—— welding, 1281 (I)
Brotherhood, Peter, Ltd., 1266 (II), 1288 (II)
—— compressors, 1288 (II)
Brotherhood, Ricardo oil engines, 1266 (II)
Brown & Sharpe system of involute gear teeth, 971 (I)
—— wire gauge, 88 (I)
Bruges and Crimp's formula for discharging capacity of sewers, 758 (I), 771 (II)
Bruges ship canal, 601 (I)
Brushes, carbon, 1192 (I)
Brussels-Charleroi Canal, 587 (I)
Bucket or ladder dredgers, 632 (I)
Buckets, Pelton wheel, 839 (I)
Building Acts, London, 318 (I), 322 (I), 381 (I), 393 (I)
—— blocks, hollow, 369 (I)
—— bricks, specification of, 365 (I)
—— foundations, 326 (I)
—— materials, effect of moisture changes on, 359 (I)
—— mechanical properties of, 180 (I)
—— strength of, 180 (I)
—— steel, 163 (I)
—— -up beams, 312 (I)
—— walls, 319 (I)
—— rules for, 319 (I)
Buildings, acoustical defects, 74 (I)
—— brickwork, estimating, 316 (I)
—— British standard steel for, 163 (I), 222 (I), 227 (I)
—— effect of weight of, on retaining walls, 457 (I)
—— electrical equipment of, Institution of Electrical Engineers regulations, 1127 (I)
—— fire resisting, 714 (II)
—— floors, 305, 318 (I)
—— heating, 659 (II) (*see also* Heating)
—— hot water supply of, 668 (II), 672 (II)
—— thermal control, 2762
—— illumination of, 685 (II)
—— reinforced concrete in; code of practice, 397 (I)
—— scaffolding, 448 (I), 1101 (II)
—— shoring, 450 (I)
—— sound insulation, 75 (I), 1097 (II)
—— structural steelwork for, London County Council Act, 322 (I)
—— superimposed loads, 318 (I)
—— vibration in, 76 (I)
—— walls, 319 (I)
—— weight effect on retaining walls, 457 (I)
—— wind pressure, 319 (I)
Bulb angles, British standard, 244 (I)
—— plates, British standard, 244 (I)
Bulkhead doors, hydraulic, 794 (I)
Bulkheads, aircraft, 428 (II)
Bull-head rails, 472 (II)
Bull's Metal & Melloid Co., Ltd., 'Melloid,' 1174 (II)
—— metals, 1174 (II)
Buoyancy, centre of, in ships, 273 (II)
Buoys, 645 (I)
—— bell, 645 (I)
—— channel demarcation by, 645 (I)
—— gas, 645 (I)
—— lenses for, 646 (I)
—— luminous, 645 (I)
—— whistling, 645 (I)
Bureau Veritas rules for safety valves, 57 (II)
Burge's formula for flood discharge, 666 (I)

Burman-Douglass steering gear, 371 (II)
Burners, gas, 1282 (II)
——— Gako, 1282 (II)
——— liquid fuel, 1361 (I)
——— pulverised fuel, 1340 (I), 1188 (II)
Burnhope reservoir, 694 (I)
Burtoning (cargo), 623 (I)
Burton's patent steel tubular scaffolding, 1101 (II)
Buses, electric trolley, 384 (II)
Bushings, terminal insulator, 1502 (I)
Butt welding, 1290 (I)
Button welding, 1291 (I)
Bypass valves, 782 (I)
By-product recovery, 1323 (I)
——— ovens, 1321 (I)

CABLE, cab-tyre sheathed, 1117 (I)
——— length, 1 (I)
——— reels, 1199 (II)
Cables, electric (*see* Electric cables)
——— suspension bridge, 537 (I)
Cab-tyre sheathed cable, 1117 (I)
Cadmium, 1208 (I), 1227 (I)
——— plating, 1193 (I)
Caesium, 1208 (I)
Cage-type induction motors, 1491 (I)
Cail screw thread, 95 (I)
Caisson disease, 643 (I)
Caissons, 445 (I), 622 (I)
Calcareous cements, 351 (I)
Calcium, 1208 (I)
——— brine, 730 (II)
Californian type dredgers (gold), 939 (II)
Callendar's tables of properties of saturated steam, 3 (II)
Calorie, 54 (I)
Calorific capacity of fuel, 55 (I)
——— intensity, 1135 (I)
——— properties of fuels, 1316 (I)
——— value of boiler fuels, 20 (II)
——— coal, 1309 (I), 856 (II)
——— gas, 1368 (I)
——— tests for, 649 (II)
——— commercial fuels, 1309 (I)
——— gaseous fuels, 1365 (I)
——— liquid fuels, 1365 (I)
——— refuse, 1349 (I)
——— solid, liquid, and gaseous fuels, tables, of, 1365 (I)
Calorimeter, throttling, 6 (II)
Calorising iron, 1195 (I)
Camber of bridge girders, 508 (I)
Cambridge optical pyrometer, 1138 (I)
Cams, internal combustion engine, 219 (II)
Camshaft control for motor-coach trains, 591 (II)
——— drives for automobile engines, 336 (II)
Canalisation of rivers, 583 (I)
——— cost of, 583 (I)
Canals, Albert, 603 (I)
——— Amsterdam, 601 (I)
——— bank protection in Italy, 597 (I)
——— British, 585 (I)
——— Bruges, 601 (I)
——— Brussels-Charleroi, 587 (I)
——— excavation, 717 (I)
——— capacities of, 585 (I), 679 (I)
——— Continental, 587 (I)
Canals, cost of, 590 (I)
——— critical velocity of, 708 (I)
——— curves, 595 (I)
——— dimensions of, 586 (I)
——— excavation, 717 (I)
——— flow in, 679 (I)
——— French, 587 (I)
——— German, 587 (I), 589 (I)
——— haulage on, 597 (I)
——— inclined planes, 593 (I)
——— irrigation, 708 (I)
——— Kiel, 599 (I)
——— lateral, 585 (I)
——— lifts, 590 (I)
——— locks, 590 (I)
——— loss of water on, 595 (I)
——— Manchester ship, 599 (I)
——— Mittelland, 602 (I)
——— Moscow-Volga, 602 (I)
——— New York, 590 (I)
——— New Welland, 602 (I)
——— Panama, 600 (I), 640 (I)
——— railway owned, 2137 (I)
——— Rhone-Marseilles, 587 (I)
——— ship, 599 (I)
——— silt borne by, 713 (I)
——— Suez, 600 (I)
——— Swedish, 587 (I)
——— traction on, 597 (I)
——— United Kingdom, 585 (I)
——— cost of, 590 (I)
Candle power, 685 (II)
Candles, International or standard, 685 (II)
Cannel coal, 855 (II)
Cans, lathe, A. C. Wells & Co., Ltd., 1151 (II)
Cantilever plate girder bridges, 515 (I)
Canvas belts, 910 (I)
Capacities of irrigation canals, mean velocity, 679 (I)
Capacity measures, 8 (I)
——— of canals, 585 (I)
Capped nuts, 104 (I)
Capping of winding ropes, 946 (I)
Caprotti valve gear, 540 (II)
Capstans, 808 (II)
Carat, the metric, 8 (I)
Carbide tools, tungsten, 1060 (I)
Carbon arc welding, 1286 (I)
——— brushes, 1492 (I)
——— combustion of, 60 (I)
——— conversion of, 60 (I)
——— filters, 733 (I)
——— heat generated by combustion of, 60 (I)
——— influence on cast iron, 1146 (I)
——— monoxide, 1367 (I)
——— properties of, 56 (I)
——— steel, 158 (I), 1063 (I), 1158 (I)
——— tools, 1063 (I)
——— cutting speeds, 1074 (I)
Carbonic acid gas fire protection, 715 (II)
——— properties of, 727 (II)
——— refrigerating machines, 728 (II)
Carbonising (gas) plant, 618 (II)
Carburettors, 350 (II)
Card indicator, to take, 79 (II)
Cardox, 615 (II)
Cargo burtoning, 623 (I)
Carriages, railway, 571 (II)
——— heating by steam, 578 (II)
——— wheels of, 577 (II)

Carriageway, width of, 560 (I)
Carriageways, concrete, 561 (I)
Carrier operated despatch system, 1302 (II)
Carriers for aerial ropeways, 831 (II)
Cart and barrow runs, 432 (I)
Cartridges, sheathed, 951 (II)
Carts and wagons, loads of, 432 (I)
Case-hardening, 982 (I), 1187 (I)
——— materials, 1187 (I)
————— gaseous, 1188 (I)
————— liquid, 1188 (I)
————— solid, 1187 (I)
————— thermal treatment of, 1188 (I)
——— nitrogen, 1189 (I)
——— rapid, 1189 (I)
——— Shorter process, 1189 (I)
——— Tocco process, 1189 (I)
Casein cements, 353 (I)
Cast brass, 1213 (I)
Cast iron, 1143 (I)
—— austenitic, 1152 (I)
—— balls, weight of, 127 (I)
—— bar, 127 (I)
—— chemical composition, 1146 (I)
—— classification, 1143 (I)
—— columns, 229 (I)
—— corrosion resistance, 1156 (I)
—— cutting, 1076 (I), 1285 (I)
—— elastic moduli, 1153 (I)
—— gears, 981 (I)
—— graphitisation, 1144 (I)
—— hardness, 1154 (I)
—— heat resistance, 1155 (I)
—— hydraulic piping, 114 (I), 786 (I)
—— impact test, 1153 (I)
—— machining, 1154 (I)
—— malleable, 1149 (I)
—— micro-constituents, 1145 (I)
—— packing, 1281 (II)
—— paving, 554 (I)
—— pillars, 229 (I)
—— pipes, 114 (I), 145 (I)
——— flanges of, 108 (I)
——— flow of water in, 760 (I)
——— standard sockets for, 114 (I)
——— weight of, 140 (I), 145 (I)
————— fittings, 147 (I)
—— properties of, 1119 (I), 1157 (I)
—— rail chairs, 490 (II)
—— sheet, bar, and ball, weight of, 127 (I)
—— soldering, 1260 (I), 1177 (II)
—— strength of, 162 (I), 1153 (I)
—— tensile strength, 1150 (I)
—— tranverse strength, 1151 (I)
—— vibration damping capacity, 1154 (I)
—— water-pipe fittings, weights of, 147 (I)
—— wear-resistance, 1155 (I)
—— weight of castings in, 154 (I)
—— welding, 1279 (I)
Castigliano's Theorem, 432 (I)
Casting and founding, 1261 (I)
—— die, 1264 (I), 1179 (II)
——— alloys used for, 1264 (I)
Castings, cleaning, 1263 (I)
—— cores, 1261 (I)
—— die, 1264 (I), 1179 (II)
——— alloys used for, 1264 (I)
—— dips for, 1263 (I)
—— iron for locomotives, 560 (II)
—— moulding sands, 1261 (I)
Castings, pattern composition for, 1262 (I)
—— sealing up cracks in, 1262 (I)
—— shrinkage of, 1263 (I)
—— steel, for locomotives, 554 (II)
—— weight of metal, compared with wood patterns, 154 (I)
—— wrought iron, 560 (II)
Castle nuts, British standard, 102 (I)
Catchment areas, 662 (I)
Caterpillar tractors for canal haulage, 598 (I)
Caulking hammers, 1078 (I)
Caustic alkalinity of boiler water, 51 (II)
—— embrittlement of boiler plates, 51 (II)
Cauvery Metur Dam, 687 (I)
Cavity bricks, 364 (I)
Cells, photo-electric, 1527 (I)
—— primary, 1476 (I)
Celluloid cements, 354 (I)
—— and cellulose regulations, 986 (II)
Cellulose lacquers, 986 (II), 988 (II)
Cement, bound roads, 563 (I)
—— bulk density, 125 (I)
—— concrete, 343 (I)
——— bricks, 364 (I)
—— electric, 354 (I)
—— Keenes', 357 (I)
—— mortar, 338 (I)
—— Portland, 357 (*see* Portland cement) (I)
—— rapid hardening, 357 (I)
—— refractory, 354 (I), 376 (I)
Cementation process, François, 870 (II)
Cemented carbide tools, 1060 (I)
Cementing steel, 1168 (I)
Cementite, 1145 (I)
Cements, 353 (I)
—— adhesive, 353 (I)
—— bituminous, 353 (I)
—— calcareous, 354 (I)
—— casein, 353 (I)
—— celluloid, 354 (I)
—— electric, 354 (I)
—— fishglue, 353 (I)
—— flour and starch, 354 (I)
—— gypsum, 354 (I)
—— mastic, 353 (I)
—— oxychloride, 354 (I)
—— Portland, 355 (I)
—— rapid hardening, 354 (I), 357 (I)
—— refractory and fire-resisting, 354 (I), 376 (I)
—— rubber, 354 (I)
—— strength, 180 (I)
—— water glass or silicate, 353 (I)
—— waterproof, 357 (I)
Centigrade-Fahrenheit conversion table, 49 (I)
Central heating boiler, Charles McNeil Ltd., 1217 (II)
Centre of buoyancy of ships, 273 (II)
—— flotation of ships, 274 (II)
—— gravity, 17 (I)
—— percussion, 25 (I)
Centrifugal fans, 1128 (I)
——— for cupolas, 1128 (I)
———— smiths' fires, 1128 (I)
—— force, 24 (I)
——— on bridges, 499 (I)
—— governor, 24 (I)
—— pump, Pulsometer, 1125 (II)
——— performance, 816 (I)
——— starting, 824 (I)
—— pumps, 809 (I), 1125 (II), 1130 (II)

Centrifugal pumps, design of, 822 (I)
—— erection of, 823 (I)
—— impeller vanes, 820 (I)
—— impellers for, 816 (I)
—— mine, 900 (II), 935 (II)
—— multi-stage, G. & J. Weir, Ltd., 1130 (II)
—— packing for, 823 (I)
—— stresses in drums and discs, 1521 (I)
Cerium, 1209 (I)
Chain adjustment, 966 (I)
— annealing, 176 (I)
— belts, 961 (I)
—— bearing pressures, 963 (I)
—— drive formulæ, 966 (I)
——— dimensions, 964 (I)
—— pitch in relationship to shaft speed, 962 (I)
—— power transmitting capacity, 963 (I)
— brittleness at low temperatures, 176 (I)
— drives, Morse Chain Co., Ltd., 1155 (II)
—— Renold & Coventry Chain Co., Ltd., 1158 (II)
— driving, 961 (I), 1155 (II)
—— bearing pressures, 963 (I)
—— centre distance between shafts, 964 (I)
—— crane, 169 (I)
— Gunter's, 1 (I), 249 (I)
— iron, strength of, 169 (I)
— lubrication, 966 (I)
— rings, 171 (I)
— roller proportions, 962 (I)
— short-link, 170 (I)
— slings, 172 (I)
— speed (driving), 962 (I)
— strength, 169 (I)
— surveying, Gunter's, 1 (I), 249 (I)
— swivels, 172 (I)
— testing, 175 (I)
— tooth cutters, 966 (I)
— types of driving, 961 (I)
— wheel teeth, 962 (I)
—— materials, 965 (I)
—— pitch diameter, 964 (I)
— wrought iron, annealing and normalising, 176 (I)
Chaining, 249 (I)
— over sloping ground, 253 (I)
Chairs, cast-iron rail, 490 (II)
Chambon dam, 696 (I)
Chamier's formula for flood discharge, 666 (I)
Champlain canal, U.S.A., 694 (I)
Channel curves, 582 (I)
— demarcation of, by buoys, 645 (I)
— training works, 580 (I)
Channels, British standard, 245 (I), 350 (I)
— flow of water in, 578 (I), 675 (I), 756 (I)
— velocity of water in, 578 (I)
— wind, 390 (II)
Charcoal iron lapwelded boiler tubes, 12 (II)
— wood, 1330 (I)
Charging accumulators, 1478 (I)
— battery vehicles, 385 (II)
Charles' gas law, 188 (II)
Chassis lubrication, 381 (II)
Chemical action and fatigue, 186 (I)
— and Insulating Co., Ltd., heat insulating materials, 1102 (II)
Chemical elements, atomic weights and specific gravities, 120 (I)
—— melting points of, 51 (I)
— fire extinguishers, 709 (II)
— purification of water, 733 (I)
— resistant enamels, 989 (II)
— treatment of foundations, 436 (I)
— weed killers on railways, 494 (II)
'Chezy's' formula for flow of water in pipes and channels, 675 (I), 756 (I)
Chilian mills (mining), 930 (II)
Chimney area, gasworks, 622 (II)
— losses, determination of, 21 (II)
— shafts, stability of, 464 (I)
Chimneys, 464 (I)
— brick, 46 (II)
— dimensions of, 465 (I), 11 (II), 667 (II)
— emission of dust and ash from, 46 (II)
— house heating, 667 (II)
— L.C.C. regulations, 466 (I)
— stability, 464 (I)
— steel, 466 (I)
— table of sizes of, 45 (II)
— tall, 465 (I)
— temperature of flues of, 44 (II)
— wind pressure on, 464 (I)
China clay, 363 (I)
Chip breakers, 1061 (I)
Chisel steels, 1181 (I)
Chisels, rock breaking, 642 (I)
Chlorate explosives, 951 (II)
Chlorination and chloramination of water, 733 (I)
Chrome, nickel steel, 1164 (I)
— steel, high, 1187 (I)
— Vanadium steels, 1167 (I)
Chromite bricks, 371 (I)
Chromium, 1206 (I)
— effect of, in steel, 1183 (I)
— influence on cast iron, 1148 (I)
— nickel-alloys, 1219 (I)
— plated tools, 1161 (I)
— steels, 1167 (I)
Chucks, magnetic, 1063 (I)
Chuff bricks, 364 (I)
Ciment fondu, 354 (I)
Cippoletti weirs, 748 (I)
Circles, area of, and squares, 1046 (I)
— circumference and area of, 1039 (I)
— geometrical and inertial properties of, 31 (I)
Circular girders, 531 (I)
— reservoirs or bins, reinforced concrete, 405 (I)
Circulating pumps, condenser, 180 (II)
— water screens, 1263 (II)
Clamps, wire rope, 948 (I)
Clarke, Chapman & Co., Ltd., coal-pulveriser, 1186 (II)
—— feed pump, 1121 (II)
—— Raymond bowl mill, 1187 (II)
—— steam separator and dryer, 1203 (II)
—— water-tube boiler, 1201 (II)
Clay, china, 363 (I)
Clay or marl roofing tiles, 367 (I)
Clays and earths, 363 (I)
— aluminous, 371 (I)

Clays, siliceous, 371 (I)
Cleaners, boiler tube, 1256 (II)
Cleaning castings, 1263 (I)
Clock and sundial time variation, 261 (I)
—— oil, 996 (II)
—— springs, 897 (I)
Clocks, synchronous, 1529 (II)
Closets, water, 797 (II)
Cloths and papers, abrasive, 1114 (I)
Club end method of socketing wire ropes, 947 (I)
Clutches, Coil, 1152 (II)
———— cone, 929 (I)
———— Ferodo linings, 1153 (II)
———— friction, 929 (I)
———————— principles, 931 (I)
———— motor car, 356 (II)
Clyde Blowers Ltd., soot blowers, 1231 (II)
—— bridge, 489 (I)
CO_2 Recorders, 1315 (I), 1193 (II)
Coal, 1313 (I), 298 (II)
—— analysis of, 1317 (I)
—— anthracite, 855 (II)
—— briquette, 1330 (I)
—— by-product recovery, 1323 (I)
—— calorific value of, 1309 (I), 858 (II)
—— cannel, 855 (II)
—— classification of, 851 (II)
—— cleaning, 908 (II), 913 (II)
———————— dry, 916 (II)
—— coking, 1321 (I)
—— combustion and smoke, 1334 (I)
—— commercial selection of, 1317 (I)
—— composition of, 1316 (I), 856 (II)
—— constitution, 856 (II)
—— consumption of, in collieries, 883 (II)
———————————— locomotives, 562 (II)
—— cost of getting, 879 (II), 885 (II)
—— cutters, 876 (II), 1294 (II)
—— cutting, cost of, 879 (II), 885 (II)
—— decomposition, temperature of, 1319 (I)
—— distillation products of, 631 (II)
—— dust burners, 1340 (I)
———— feeders, 1341 (I)
———— in mines, 906 (II)
———— petroleum mixtures, 1363 (I)
—— face machinery, 1294 (II)
—— firing at sea, 298 (II)
—— gas (*see* Gas)
———— cost of manufacturing, 635 (II)
———— dilution of, with producer gas, 621 (II)
———— fuel, 1368 (I)
—— impurities in, 1319 (I)
—— low temperature distillation of, 1325 (I)
—— meters, 191 (I), 204 (I)
———— chain type shute, 206 (I)
—— mine legislative enactments, 817 (II)
—— mines, electric signalling in, 908 (II)
———— explosives, 950 (II)
———— firedamp detectors in, 901 (II)
—— mining, 853 (II)
———— subsidences, 875 (II)
—— power compared with water power, 719 (I)
—— pre-gasification of, 1341 (I)
—— pulverised fuel, 1337 (I), 1188 (II)
———————— advantages of, 1346 (I)
———————— at sea, 1342 (I)
———————— combustion chambers for, 1346 (I)
———————— firing boilers with, 1188 (II)
Coal, pulverised for locomotives, 1345 (I)
———————— Lopulco system, 1188 (II)
—— pulverisers, 1339 (I), 1341 (I), 1186 (II)
—— sampling, 1317 (I)
—— shipment, 623 (I)
—— sizing, 857 (II)
—— specific gravity of, 857 (II)
—— storage, 1319 (I)
———— gasworks, 632 (II)
———— loss of thermal value by, 1320 (I)
—— stowage space of, 1320 (I)
—— tar, 1323 (I), 1356 (I)
—— washing, 913 (II)
—— weighers, 206 (I)
—— weight of, 857 (II)
Coaling appliances, 623 (I)
Coalite, 1325 (I)
Coatings for iron and steel, 1192 (I)
Cobalt, 1209 (I)
Cochran & Co., Annan, Ltd., boilers, 1206 (II)
———————— evaporation of, 1206 (I)
———————— exhaust gas, 1208 (II), 1211 (II)
———————— gas fired, 1214 (II)
———————— 'Sinuflo,' 2121 (II)
———————— thermal storage (Ruths), 1215 (II)
———————— waste heat, 1211 (II)
Cocks, injection, condenser, 181 (II)
—— Klinger, 1211 (II)
Coefficient, gravity, 27 (I)
Coefficients, Admiralty, 281 (II)
———————— applied to fuel consumption, 282 (II)
———— of displacement, 272 (II)
———— friction, 932 (I)
———————— motor car tyres, 321 (II)
Coffer or puddle dams, 447 (I)
Coffering of mines, 867 (II)
Coignet system of concrete construction, 398 (I)
Coil Clutch Co., Ltd., clutch, 1152 (II)
Coils, wireless tuning, 969 (II)
Coke, 1321 (I), 1323 (I), 857 (II)
—— as a boiler fuel, 1326 (I)
—— composition of, 857 (II)
—— furnace, 1128 (I), 1129 (I)
—— storage of, 633 (II)
—— used in gas manufacture, 633 (II)
Coke-oven gas, 1325 (I), 1369 (I)
Coking, 1321 (I)
———— by-product recovery, 1323 (I)
———— coals, 1321 (I), 855 (II)
———— ovens, 1321 (I)
Cold solder, 1260 (I)
—— storage, 731 (II)
Collar-beam roof, 303 (I)
Colliery bricks, 364 (I)
———— haulage gear, 889 (II)

Colliery leases, 847 (II)
——— tubs, 891 (II)
Colloidal graphite, 880 (I)
Colour of smoke, 1336 (I)
——— light signalling on railways, 606 (II)
——— ——— signals, 606 (II)
Coloured glass, absorptive powers of, 653 (I)
——— mortars, 342 (I)
Colours, light, 64 (I)
——— of different temperatures, 50 (I)
——— tempered steel, 1072 (I)
Columbia River project, 691 (I)
Columbium (Niobium), 1209 (I)
Columns, 227 (I)
——— cast iron, 229 (I)
——— eccentric loads on, 229 (I)
——— loads on, 227 (I), 437 (I)
——— reinforced concrete, 396 (I)
——— safe load on, 437 (I)
——— strength of, 227 (I)
Combination of colours (light), 64 (I)
Combustion, 60 (I), 1310 (I)
——— air required for, 1361 (I), 1392 (I), 615 (II)
——— and heat, 54 (I)
——— chambers, refractory linings for, 1346 (I)
——— computation, 1314 (I)
——— draught and rate of, 1311 (I), 22 (II)
——— excess air, 1312 (I), 19 (II)
——— external, oil mixtures for, 1363 (I)
——— formulæ, 21 (II)
——— gas engine, 251 (II)
——— heat generated by one pound of carbon, 60 (I)
——— heat of, 55 (I)
——— indicator, 1193 (II)
——— of carbon, 60 (I)
——— ——— heat generated by, 60 (I)
——— ——— coal, and smoke, 1334 (I)
——— ——— gas, weight of air or oxygen necessary for, 1366 (I), 615 (II)
——— ——— hydrocarbon fuels, 1312 (I)
——— rate of, 22 (II)
——— spontaneous, in mines, 907 (II)
——— table of heat of, 55 (I)
Common bricks, 361 (I)
Communication, radio, 957 (II)
Commutator alternating current motors, 1192 (I)
Compass, prismatic, 259 (I)
——— variation, annual, 259 (I)
——— ——— to find, 259 (I)
'Compo' oil-retaining bearings, 1139 (II)
Composition, nails, weight of, 156 (I)
Compound dredgers, 636 (I)
——— locomotives, 536 (II)
——— pendulum, 27 (I)
Compressed air, horse-power of, 751 (II)
——— liquid fuel burners, 1361 (I)
——— locomotives, 891 (II)
——— pumping by, 826 (I)
——— water-raising by, 826 (I)
Compressibility of water, 744 (I)
Compressing plant arrangement for air lift pumps, 830 (I)
Compression, air, 753 (II)
Compression, air, ammonia, system of refrigeration, 720 (II)
——— cost of, 1082 (I)
——— gas, adiabatic, 1300 (I), 188 (II), 757 (II)
——— ignition, motor-car engine, 310 (II)
——— ——— aero engines, 465 (II)
——— or expansion, isothermal, 1299, (I), 188 (II)
——— ratio, 190 (II)
Compressors, air, 753 (II), 1288 (II)
——— ——— brake horse power required at shaft, 755 (II)
——— ——— Brotherhood, 1288 (II)
——— ——— coolers for, 1292 (II)
——— ——— efficiency of, 1302 (I)
——— ——— filters for, 234 (II)
——— ——— lubrication, 889 (I)
——— ——— nozzles, 765 (II)
——— ——— testing, 763 (II)
——— ——— Weir, 1290 (II)
——— gas, 258 (II)
——— indicator diagrams of, 721 (II)
——— refrigeration, 721 (II), 736 (II)
Concave lens, 64 (I)
Concentrations, stress, 187 (I)
Concrete, 343 (I)
——— beton, 343 (I)
——— blocks, 350 (I)
——— breeze, 343 (I)
——— bricks, 367 (I)
——— bridges, reinforced, 571 (I)
——— carriageways, 561 (I)
——— cement, 343 (I)
——— channels, 350 (I)
——— coal residues in, effect of, 349 (I)
——— colouring, 348 (I)
——— condensation, on, 348 (I)
——— cracking, prevention of, 348 (I)
——— crushing strength of, 347 (I)
——— dams, 696 (I)
——— effect of pigments on, 348 (I)
——— ——— sea water on, 348 (I)
——— flags, 350 (I)
——— floors, 306, 398 (I)
——— ——— dusty, treatment of, 349 (I)
——— kerbs, 350 (I)
——— lime, 343 (I)
——— materials for, 345 (I)
——— mixers, 345 (I), 569 (I)
——— ——— capacities and leading dimensions of, 346 (I)
——— mixing, 345 (I)
——— partition slabs, 309 (I)
——— permeability of, 348 (I)
——— piles, moulded-in-place, 438 (I)
——— ——— reinforced, 438 (I)
——— pipes, reinforced, 352 (I), 723 (I)
——— placing, 346 (I)
——— prevention of cracking and scaling of, 348 (I)
——— proportions for, 344 (I)
——— quadrants, 350 (I)
——— rapid hardening cement for, 357 (I)
——— raw materials for, 344 (I)
——— reinforced, 343 (I), 381 (I) (*see* Reinforced concrete)
——— ——— floors, 398 (I)
——— ——— roads, 561 (I)
——— roads, 561 (I)

Concrete, rust, prevention of, 349 (I)
—— scaling, prevention of, 318 (I)
—— sea-water, effect on, 318 (I)
—— slag, 343 (I)
—— sleepers, 488 (II)
—— slump test, 344 (I)
—— strength of, 347 (I)
—— tiles, 350 (I), 353 (I)
—— volume of, 347 (I)
—— walls, permissible pressures on, 322 (I)
—— waterproofing of, 349 (I)
—— weight of, 347 (I)
Condensation on concrete, 348 (I)
Condensed steam, measurement of, 181 (II)
Condenser air pumps, 178 (II), 180 (II), 1262 (II)
—— circulating pumps, 180 (II)
—— injection cocks, 181 (II)
—— tube plates, 169 (II)
—— tubes, 169 (II)
—— —— cleaning of, 169 (II)
Condensers, 167 (II), 1261 (II)
—— air in, 177 (II)
—— cooling surface, 172 (II)
—— —— water for, 175 (II)
—— ejector, 177 (II)
—— electrical, 1505 (I)
—— gas, 625 (II)
—— jet, 167 (II)
—— power-station, 169 (II)
—— radio, reactance, 965 (II)
—— refrigerating, 736 (II)
—— surface, 168 (II), 1261 (II)
—— —— heat transmission in, 170 (II)
—— water cooled, 626 (II)
—— —— flow resistance in, 176 (II)
—— Weir regenerative surface, 1261 (II)
Conditioning of air, 676 (II)
Conductivity, thermal, 63, 310 (I)
—— —— of various materials, 63 (I)
—— —— —— metals, 1204 (I)
Conductor rails for electrified track, 478 (II)
Conductors, earth continuity, 1392 (I)
—— electrical, aluminium, 1403 (I)
—— —— copper, 1380 (I), 1407 (I)
—— —— —— British Standard, 1380 (I)
—— —— standard stranded, 1408 (I)
—— lightning, rules for, 1516 (I)
Conduits, electrical, 1118 (I)
—— dimensions of steel, 1119 (I)
—— steel, British standard specification for, 1419 (I)
Cones, Seger, 1136 (I)
Conic frustum, 19 (I)
Conical coverings, reinforced concrete, 408 (I)
—— helical springs, 896 (I)
—— pendulum, 24 (I)
Connecting rods, internal combustion engine, 209 (II), 215 (II), 335 (II)
—— locomotive, 538 (II)
Considere system of concrete construction, 398 (I)
Constantan alloy, 1226 (I)
Continental canals, 587 (I)
Continental screw threads, 94 (I)
'Contra-Lock' wire ropes, 1154 (II)
Contract for works, engineer's rights and powers, 1030 (II)
—— specification for locomotive construction, 561 (II)
Control gear, electrical, 1497 (I)
—— —— crane, 813 (II)
—— train, 609 (II)
—— valves, hydraulic, 777 (I)
Convergence of meridians, 272 (I)
Conversion of degrees Centigrade into Fahrenheit, 49 (I)
—— tables, 3–9 (I)
—— —— English, Metric, 3–9 (I)
Convex mirrors, 64 (I)
Conveyor volume meters, 204 (I)
—— weight meters, 205 (I)
Conveyors, and elevators, gas works, 635 (II)
—— belt, 840 (II), 1302 (II)
—— pneumatic tube, 1302 (II)
Coolers, air, tubular, 1198 (II)
—— oil, for the heat treatment of steel, 1174 (II)
—— Visco, 1264 (II)
—— water, 1264 (II), 1273 (II) 1287 (II), 1292 (II)
Cooling aero engines, 455 (II)
—— air, 677 (II), 1198 (II)
—— automobile engines, 346 (II)
—— brake rims, 43 (I)
—— compounds for hot bearings, 996 (II)
—— internal combustion engines, 198 (II)
—— ponds, 181 (II)
—— surface of condensers, 172 (II)
—— towers, water, 182 (II), 1264 (II)
—— water for condensers, 175 (II)
—— —— internal combustion engines, 1273 (II)
Copes Flowmatic regulator, 1243 (II)
—— pump governor, 1244 (II)
—— Regulators, Ltd., feed water regulator, 1243 (II)
Copper, 1203 (I)
—— alloys, 1212 (I), 1226 (I), 1253 (I)
—— —— properties of, 1253 (I)
—— aluminium alloys, 1225 (I), 1230 (I)
—— balls, weight of, 127 (I)
—— bars, weight of, 131 (I)
—— base alloys brasses, 1249 (I)
—— —— bronzes, 1250 (I)
—— beryllium alloys, 1227 (I)
—— cadmium alloys, 1227 (I)
—— 'clothing,' 1193 (I)
—— conductors, 1380 (I), 1407 (I), 1409 (I)
—— electrolytic, elasticity of, 159 (I)
—— fuse wire, 1413 (I)
—— hydrogen-electric welding, 1293 (I)
—— impurities in, 1204 (I)
—— influence on cast iron, 1148 (I)
—— manganese alloys, 1226 (I), 1251 (I)
—— nails, weight of, 155 (I)
—— nickel alloys, 1226 (I), 1235 (I), 1251 (I)
—— ore, 859 (II)
—— overhead lines, electrical, 1407 (I), 1409 (I)
—— physical properties, 1253 (I)
—— pipes, weight of, 141 (I)
—— plates for locomotive fire-boxes, 555 (II)
—— rivets, weight of, 156 (I)

Copper, rods for locomotive stay-bolts, 555 (II)
——— rolled, weight of, 131 (I)
——— sheet and bar, weight of, 131 (I)
——— silicon alloys, 1227 (I)
——— steels, 1202 (I)
——— strength of, 1203 (I)
——— stress due to temperature, 177 (I)
——— tensile strength, 186 (I)
——— tin alloys, 1250 (I)
——— tubes, for locomotive boilers, 555 (II)
——————— seamless, weight of, 149 (I)
——— weight of castings in, 154 (I)
——— welding, 1281 (I)
——————— enamelled, 1412 (I)
——————— hard-drawn, 163 (I)
——————— tinned fuse, 1413 (I)
——————— weight of, 154 (I)
——— zinc alloys, 1212 (I)
Cords, flexible electric, 1383 (II)
Core boring, 861 (II)
Cores, sand, 1261 (I)
Cork insulation for refrigeration, 733 (II)
Cornish boilers, 13 (II)
Corrosion in boilers, preventives of, 1239 (II)
——— of aluminium, 1207 (I)
——————— iron, 178 (I)
——————————— pipes, 734 (I)
——————————— ropes, 178 (I)
——————— metals, 178 (I), 1192 (I)
——————— soft steel, wrought iron and nickel steel, 178 (I)
——————— steel, iron, and Delta metal, 178 (I)
——————————— pipes, 734 (I)
——— protection methods for metals, 1192 (I)
——— relative, 1196 (I)
——— resistance of cast iron, 1156 (I)
Corrugated-iron galvanised fixings, 236 (I)
——————————— sheets, 239 (I)
——————————— roofing, 233 (I), 236 (I), 239 (I), 301 (I)
——————— steel sheets, 238 (I)
——————————— steam tube bends, 61 (II)
Coslettising iron, 1195 (I)
Cost of arc-welding, 1289 (I)
——— breakwaters, 606 (I)
——— canals, 583 (I), 717 (I)
——————— in Great Britain, 590 (I)
——————— lateral, 585 (I)
——— coal cutting, 876 (II), 879 (II)
——————— gas, 635 (II)
——— compressing air, 1082 (I)
——— docks, 611 (I), 626 (I)
——— dredging, 582 (I), 639 (I), 938 (II)
——————— plant, 638 (I)
——— drilling rock, 641 (I)
——— electric baking ovens, 1523 (II)
——————— vehicles, 588 (II)
——— floating docks, 626 (I)
——— gas purifiers, 635 (II)
——————— manufacture, 615 (II)
——— gasworks, 615 (II)
——— goldmill equipment, 929 (II)
——— graving docks, 611 (I), 626 (I)
——— heating, 665 (II)
——— irrigation, 712 (I)
——— lateral canals, 585 (I)
——— light railways, 500 (II)
——— lighthouses, 649 (I)
——— mining operations, 865 (II), 868 (II), 876 (II), 879 (II)
Cost of ploughing, 38 (I)
——— pulverised coal mill house, plants, etc., 1348 (I)
——— quay walls, 617 (I)
——— railway electrification in Great Britain, 588 (II)
——— railways, 498 (II)
——— river canalisation, 583 (I)
——— road maintenance, 547 (I)
——— roads, 546 (I), 565 (I)
——— sinking mining shafts, 865 (II), 868 (II)
——— transport, 37 (I)
——— tubbing in mines, 868 (II)
——— water softening, 732 (I)
Cotton ropes, 920 (I)
——————— drives, 920 (I)
——————————— fluctuating loads on, 926 (I)
——————— splicing, 927 (I)
Counterbalancing locomotive driving-wheels, 512 (II)
Coupling rods, locomotive, 538 (II)
Couplings, 1269 (I)
——————— Bibby, 1269 (II)
——————— chain, 1157 (II)
——————— flexible, 243 (II), 1269 (II)
——————— hose, 712 (II)
——————— pipe, 578 (II), 1118 (II)
——————— railway, 578 (II)
——————— tube, 1081 (I)
Coverings, domes and conical, reinforced concrete, 408 (I)
——————— roof, 233 (I)
——————— steam pipe, loss of heat from, 66 (II)
——————————— relative value of, 68 (II)
Cracks in steel plates, detection of, 33 (II)
Crane bearings, 810 (II)
——— brakes, 810 (II)
——— chains, 170 (I), 810 (II)
——— controllers, 813 (II)
——— electro-lifting magnets, 815 (II)
——— gearing, 809 (II)
——— grabs, 807 (II)
——— hooks, 937 (I)
——— lifting tackle, 170 (I)
——— materials, 809 (II)
——— pulleys and drums, 936 (I)
——— rings, 171 (I)
——— ropes, 936 (I)
——— shafts, 810 (II)
——— slings, 172 (I)
——— solenoids, 814 (II)
——— structures, 809 (II)
——— switchboards, 814 (II)
——— swivels, 172 (I)
Cranes, 622 (I), 801 (II)
——— breakdown steam, 806 (II)
——— derrick, 805 (II)
——— electric, 802 (II)
——————— wiring, 814 (I)
——— electrical equipment of, 812 (II)
——— floating, 623 (I)
——— general data of, 801 (II)
——— hand, overhead travelling, 802 (II)
——— locomotive, 805 (II)
——— mobile, 806 (II)
——— oil engined, 801 (II)
——— overhead electric travelling, 802 (II)
——— quay, 622 (I), 806 (II)
——— shipbuilding, 808 (II)

Cranes, travelling, 802 (II)
——— wharf, 806 (II)
Crank axles of locomotives, 540 (II), 550 (II)
——— cases, 226 (II), 461 (II)
——— connecting rod mechanism, 87 (II)
——— effort and turning moment, 89 (II)
——— shafts, 209 (II), 227 (II)
——— ——— automobile, 330 (II)
——— ——— torsional vibration, 235 (II)
Crankchamber oils, 880 (I)
Creep of steel at high temperatures, 155 (II)
——— prevention of rail, 181 (II)
Creosote oil, 285 (I)
Creosoting plants, 285 (I)
——— timber, 285 (I)
Cribs, 448 (I)
Crimp and Bruges formula for flow in sewers, 758 (I), 771 (II)
Critical speed in steam turbines, 118 (II)
——— speeds of gearing, 981 (I)
——— velocity, irrigation canals, 708 (I)
——— ——— in rivers, 578 (I)
Crockatt, W., & Sons' Salinometer, 1256 (II)
——— Simplex terrazzo floor polisher, 1100 (II)
——— Simplex turbine tube cleaners, 1256 (II)
——— valve reseaters, 1255 (II)
Crosby Valve & Engineering Co., Ltd., desuperheater, 1249 (II)
——— feed-water regulator, 1245 (II)
——— indicator (steam), 1095 (II)
——— nozzle type safety valve, 1249 (II)
Cross flow, heat transfer in, 31 (II)
Crossings (*see* Railways and Tramways)
Crosthwaite Furnaces & Scriven Machine Tools Ltd., mechanical stoker, 1221 (II)
Croton dam, the, 687 (I)
Crozzle bricks, 364 (I)
Crowds, weight of, 36 (I)
——— pressure of, 36 (I)
Crucible melting furnaces, 1125 (I), 1128 (I)
Crucibles, 375 (I)
Crushing mills, coal, 909 (II)
——— gold, 929 (II)
——— strength of bricks, 365 (I)
——— ——— mortars, 181 (I)
Cubes, squares, roots, reciprocals, 1062 (II)
Culverts, discharge through submerged orifices of, 671 (I)
Cup leathers, friction of, 808 (I)
Cupolas, 1127 (I)
——— centrifugal fans for, 1128 (I)
Cupro-nickel alloys, 1226 (I)
Current meters, 578 (I), 680 (I), 756 (I)
——— transformers, 1509 (II)
Curves, canal, 595 (I)
——— polar (light), 687 (II)
——— railway (*see* Railway curves)
Cushioning valves, 782 (I)
Cutter dredgers, suction, 636 (I)
Cutters, coal, 876 (II)
——— milling, 1044 (I)
——— ——— speed of, 1045 (I)
Cutting, electric, 1293 (I)
——— off machine, tube, 1165 (II)
——— oxy-acetylene, 1283 (I)
——— speeds of tools, 1071 (I)
——— tool alloys, 1161 (I), 1253 (I)
——— ——— angles of, 1070 (I)
——— ——— carbon steel, 1063 (I)
Cutting tool, high speed, 1068 (I)
——— ——— speeds, 1071 (I)
——— ——— tempering, 1069 (I)
——— ——— tungsten carbide, 1060 (I)
Cuttings, to determine volume of earth in, 421 (I)
Cyanide process, 928 (II)
Cycle engineers' screw threads, 93 (I)
Cyclic irregularity in internal combustion engines, 230 (II)
Cycloidal teeth, 974 (I)
Cyclone type dust extractor, 17 (II)
Cylinder heads, internal combustion engine, 215 (II), 331 (II), 462 (II)
——— ——— marine engines, 287 (II), 289 (II)
——— sinking by pneumatic process, 444 (I)
Cylinders, foundation, 443 (I), 489 (I)
——— gas, 656 (II)
——— hollow, strength of thick, 168 (I)
——— of compound and simple locomotives of equivalent power, 536 (II)
——— ——— compound locomotives, 536 (II)
——— ——— internal combustion engines, 199 (II)
——— ——— locomotives, 536 (II)
——— ——— marine engines, 287 (II), 289 (II)
——— ——— steam engines, 85 (II), 87 (II)
——— sinking, by pneumatic process, 444 (I)
——— thick, strength of, 168 (I)

DAMPERS, boiler, 12 (II)
——— dynamic, 957 (I), 465 (II)
Damping devices for vibration, 77 (I), 242 (II)
Dampness in walls, to remedy, 359 (I)
Dampney's 'Apexior,' 1210 (II)
Dams, 446 (I), 683 (I), 697 (I), 698 (I)
——— arch, 683 (I), 697 (I), 698 (I)
——— ——— strains under load, 693 (I)
——— Aswan, 688 (I)
——— balk, 448 (I)
——— basic economy of, 698 (I)
——— Boulder, 691 (I)
——— Cauvery Metur, 687 (I)
——— Chambon Barrage, 696 (I)
——— coffer, or puddle, 447 (I)
——— concrete, 696 (I)
——— Deer Creek, 696 (I)
——— earth, 683 (I), 695 (I)
——— failures, 698 (I)
——— Fort Peck, 695 (I)
——— Grand Coulee, 696 (I)
——— gravity, 683 (I), 684 (I), 696 (I)
——— Huffman, 693 (I)
——— hydraulic fill, 685 (I), 693 (I)
——— laboratory experiments, 695 (I)
——— Ladybower, 695 (I)
——— Laggan, 691 (I)
——— literature, 699 (I)
——— masonry, 686 (I), 696 (I)
——— multiple arch, 683 (I), 698 (I)
——— puddle, 447 (I), 684 (I)
——— recently built, 695 (I)
——— Rio Salado, Mexico, 695 (I)
——— rock-fill, 685 (I), 696 (I)
——— Ryburn, 688 (I)
——— Sardis, 695 (I)
——— shrinkage of, 693 (I)
——— single balk, 448 (I)
——— sluices, 721 (I)

Dams, spillway, 691 (I)
—— Stevenson's Creek, 693 (I)
—— Tal-y-bont, 695 (I)
—— water losses through, 708 (I)
—— Woranorah, 688 (I)
Danel Weir, 695 (I)
Danube river training works, 581 (I)
Darlington magnesia coverings, 1102 (II)
Dartford-Purfleet Tunnel, 571 (I)
Davis submarine escape, 1301 (II)
Daylight, artificial, 705 (II)
De Meletin's process, 1195 (I)
Deacon water meter, 1107 (II)
Dead-weight of ships, 279 (II)
De-aerator, Weir, 1238 (II)
Deceleration lanes, 561 (I)
Decibel scale, 71 (I)
Decomposition of light in the spectrum, 64 (I)
Deer Creek dam, 696 (I)
Diversion Weirs, 699 (I)
Diving, 643 (I)
—— apparatus, Siebe-Gorman, 1298 (II)
—— bells, 643 (I)
Docks, 611 (I)
—— Avonmouth, 611 (I)
—— bolted sectional, 629 (I)
—— cost of, 611 (I)
—— entrance locks, 617 (I)
—— floating, 625 (I)
———— largest, 627 (I)
—— gates, 617 (I)
———— box, 619 (I)
———— stresses in, 619 (I)
—— Gladstone, 611 (I)
—— graving, 625 (I)
—— King George V, 611 (I)
—— offshore, 629 (I)
—— sectional box, 619 (I)
———— pontoon, 629 (I)
—— Singapore floating, 628 (I)
—— sluices and lock gates, hydraulic, 792 (I)
—— Southampton floating, 627 (I)
—— walls, 615 (I)
———— stresses in, 615 (I)
Domes and conical coverings, reinforced concrete, 408 (I)
Domestic uses of electricity, 1520 (I)
Doors, bulkhead, hydraulically controlled, 791 (I)
—— fire-resisting, 714 (II)
Dorman, Long & Co., steel sleeper, 483 (II)
Double helical gearing, 993, (I), 160 (II)
———— gears, turbine, 160 (II)
Dover breakwater, 607 (I)
Dowelling, masonry, 331 (I)
Draft of ships, 271 (II)
Dragline excavators, 424 (I)
Drain pipes, 370, (I), 769 (II)
Drainage, house, 792 (II)
—— mine, 899 (II)
Drains, flushing, 776 (II)
Draught, boiler, 1337, (I), 16 (II)
—— effect of altitude on, 23 (II)
—— mechanical, 1337 (I)
—— of ships' boilers, 296 (II)
—— pressures, rate of combustion for, 24 (II)
—— required for efficient combustion of fuel, 21 (II)
Dredgers, compound, 636 (I)
—— dipper, 633 (I)
Dredgers, gold, 937 (II)
—— grab or grapple, 631 (I)
—— hopper service, 638 (I)
—— hydraulic, 635 (I)
—— ladder, 632 (I), 638 (I)
—— river, 637 (I)
—— sand pump, 635 (I)
—— suction, 635 (I)
———— cutter, 636 (I)
—— tin, 937 (II)
Dredging, 632 (I)
—— Bombay Harbour, 639 (I)
—— cost of, 639 (I)
———— plant, 638 (I)
—— for alluvial minerals, 936 (II)
—— Port Natal, 639 (I)
—— river, 582 (I)
Dressers, abrasive wheel, 1106 (I)
Drill bits, sharpening steel, 887 (II)
Deflection of beams, 223 (I), 405 (I)
—— bridges, 514 (I), 523 (I)
—— girders, 216 (I)
Degree, length of, in latitude and longitude, 262 (I)
Degrees of temperature, conversion of, 49 (I)
—— to radians tables, 1017 (II)
Dehydration of gas, 639 (II)
Delta brand alloys, 1215 (I), 1244 (I)
Demolition blasting, 952 (II)
Densities of bulk materials, 125 (I)
Depositing floating dock, 632 (I)
Deposition of metal on worn parts, 1082 (I)
Depreciation of plant and machinery, 1017 (II)
Derrick cranes, 805 (II)
Designs, registration of, 1008 (II)
Despatch tubes, pneumatic, 1302 (II)
Destructors, refuse, 1349 (I)
———— Heenan, 1191 (II)
De-Superheaters, Crosby, 1249 (II)
Detection of cracks in steel plates, 33 (II)
Deterioration of water mains, 759 (I)
Detonators, 863 (II), 952 (II)
Development of water-power, 719 (I)
Dewpoint, 678 (II)
'Dextramite' high temperature coverings, 1103 (II)
Diagram factor (steam indicator), 75 (II), 288 (II)
—— of steam indicator, to interpret, 81 (II)
—— the indicator, 80 (II)
Diameter of pipes, 114 (I), 734 (I), 738 (I)
Diamond rock-boring, 861 (II)
Dick, R. & J., Ltd., belting, 1152 (II)
Dicken's formula for flood discharge, 665 (I)
Die-casting, 1264 (I), 1179 (II)
Die-castings, alloys used for, 1264 (I)
———— aluminium bronze, 1226 (I)
———— steel, non-shrinking, 1185 (I)
Diesel engines (*see* Oil engines, Diesel)
—— exhaust-gas fired boiler, 1208 (II)
—— fuel oil, 1355 (I)
———— compressibility of, 341 (II)
Dikes, longitudinal, 581 (I)
Dilver, 1218 (I)
Dionic water tester, 1197 (II)
Dip, magnetic, 260 (I)
—— of horizon, 255 (I)
Dipper dredgers, 633 (I)
Dips for castings, 1263 (I)
Direct-current motors, 1487 (I)
———— control of, 591 (II)

Direct-current motors, efficiencies of, 1489 (II)
——— speed regulation of, 1489 (II)
——— traction, 593 (II)
Disc drum type turbines, 129 (II)
—— friction in turbines, 131 (II)
Discharge of water from reservoirs, 721 (I)
——— through orifices, 671 (I), 745 (I)
Discs, centrifugal stresses in, 145 (II)
Displacement (ship), 272 (II)
Disposal, refuse, 1349 (I), 1191 (II)
Dissolved acetylene, 646 (I), 653 (II)
Distance at which sounds can be heard, 70 (I)
——— measurements, 251 (I)
——— of objects at sea, 263 (I)
——— sun from earth, 263 (I)
——— visible from lighthouses, 649 (I)
Distempers, 986 (II)
Distillation of coal, low temperature, 1325 (I)
——— products from, 1326 (I), 634 (II)
——— tar, 1356 (I), 641 (II)
Distillers, tar, 1356 (I), 641 (II)
Drill gauge, Morse wire, 1057 (I)
Drilling, 886 (II)
——— and blasting rock, 641 (I)
——— diamond, mining, 861 (II)
——— machines, high-speed, 1028 (I)
——— portable pneumatic, 1030 (I)
——— power for, 1031 (I), 888 (II)
——— mining, 886 (II)
——— notes on machine, 1057 (I)
——— test, 1090 (I)
Drills, 1055 (I)
—— electric, 1031 (I)
—— pneumatic, 1030 (I), 1078 (I)
—— twist, 1055 (I)
——— British Standard specification for, [1057 (I)
——— grinding, 1055 (I)
——— Morse taper measures, 1056 (I)
Drive, hydraulic, Keelavite, 1110 (II)
Driving belts, 909 (I), 1152 (II)
——— spring, 920 (I)
——— chains, 961, (I), 1154 (II)
——— ropes, 916 (I), 920, (I)
——— vee, 916 (I)
——— steel sheet piling, 444 (I)
Drop forgings, 1121 (I)
—— hammers, 1122 (I), 1166 (II)
—— stamps, 1120 (I)
——— Brett's, 1166 (II)
Drum disc type steam turbines, 129 (II)
Drums, centrifugal stresses in, 145 (II)
——— wire rope, 936 (I), 939 (I)
Dry bulb hygrometer, 678 (II)
—— measure, 8 (I)
Drying apparatus, timber, 291 (I)
Ducommun screw thread, 95 (I)
Ducter low resistance testing set, 1197 (II)
Ductile brass, 1213 (I)
——— bronze, 1223 (I)
Ducts, flow of air through, 679 (II)
Dumpy level, 258 (I)
Dunwan reservoir, 695 (I)
Duplex spot welding, 1292 (I)
Durability of concrete, 417 (I)
——— timber, 284 (I)
Duralumin alloy, 1233 (I), 1170 (II)
——— age hardening of, 1233 (I)
Duralumin, heat treatment, 1233 (I)
——— mechanical properties, 1233 (I)
Dust, coal, 906 (II)
—— collecting from grinding wheels, 1106 (I)
——— by electrostatic precipitation, 47 (II), 1236 (II)
—— collector, Visco Beth, 1283 (II)
—— emission, from chimneys, 46 (II)
—— extraction plant, 47 (II)
Dykes, longitudinal, 581 (I)
Dynamatic dynamometers, 45, (I), 1094 (II)
Dynamic dampers, 957, (I), 165 (II)
Dynamics, 22 (I)
Dynamite, 949 (II)
Dynamometers, 39 (I), 1093 (II)
——— absorption, 41 (I)
——— Amsler, 39 (I)
——— dynamatic, 1094 (II)
——— eddy current, 45 (I)
——— electro-, 45 (I), 1094 (II)
——— fan, 44 (I)
——— Froude's water, 41 (I), 43 (I)
——— Hyatt, 39 (I)
——— hydraulic, 39 (I), 1093 (II)
——— pendulum, 39 (I)
——— spring type, 39 (I)
——— strain gauge, 39 (I)

EARTH, angle of repose of, 435 (I)
——— bulk density, 125 (I)
——— coefficient of friction, 21 (I)
——— connections for lightning conductors, 1517 (I)
——— continuity conductors, 1392 (I)
——— dams, 446 (I), 683 (I), 695 (I)
——— data regarding sun and the, 263 (I)
——— equatorial radius of the, 263 (I)
——— excavation of, 423 (I)
——— magnetic force of, 260 (I)
——— moving machines, 568 (I)
——— polar radius of, 263 (I)
——— pressure of, on retaining walls, 458 (I)
——— radius of, and distance from sun, 263 (I)
——— resistance, measuring, 1196 (II)
——— temperature of, 54 (I)
——— transporting, 425 (I)
——— volume of, in cuttings, 421 (I)
Earths and clays, 363 (I)
——— specific gravity of, 125 (I)
Earthwork, 421 (I), 573 (I)
——— tables, 422 (I), 426 (I)
East Coast iron, 1142 (I)
—— true, to determine, 261 (I)
Ebonite, machining, 1077 (I)
——— sheet and rod, weight of, 139 (I)
Eccentric loads, 229 (I)
Economic ' boiler, 13 (II)
Economisers, 17 (II)
Economy of dams, 698 (I)
Eddystone lighthouse, 649 (I)
Efficiency, mechanical, 75 (II)
——— ratio, 76 (II)
Effluents, sewage, standard of purity of, 781 (II
Ejector air pumps, 180 (II), 1262 (II)
——— Weir, 1262 (II)
——— condenser, 177 (II)
Elasticity of metals, 159 (I), 1242 (I)
——— timbers, 179 (I)

ELE

Electric accumulators, 1476 (I)
—— **ammeters, voltmeters and wattmeters, limits of errors**, 1507 (I)
—— **arc cutting**, 1293 (I)
———— **welding**, 1286, (I), 1183 (II)
—— **automobile equipment**, 1527 (I)
—— **baking ovens**, 1522 (I)
—— **batteries**, 1478 (I)
———— **vehicles**, 384 (II)
—— **booster transformers**, 1503 (I)
—— **brakes**, 810 (II), 1198 (II)
—— **building equipment**, 1427 (I), 1459 (I), 1520 (I)
—— **butt welding**, 1290 (I)
—— **cables, aluminium**, 1403 (I)
———— **steel cored**, 1406 (I), 1410 (I)
———— **braided copper**, 1409 (I)
———— **cab type sheathed**, 1417 (I)
———— **colliery**, 899 (II)
———— **copper circular**, 1380 (I)
———— **flexible**, 1382 (I)
———— **government specification for,**
———— **high voltages**, 1411 (I), 1416 (I)
———— **installation work**, 1416 (I)
———— **lead covered**, 1388 (I)
———— paper, lead-covered, 1389 (I)
———— reels, self-winding, 1199 (II)
———— standard rubber, 1415 (I)
—————— sizes, old and new, 1382 (I)
—————— stranded, 1381 (I), 1408 (I)
———— steel - cored aluminium, 1406 (I), 1410 (I)
———— vulcanised rubber, 1384 (I)
—— carbon brushes, 1492 (I)
—— cement, 354 (I)
—— clocks, 1529 (I)
—— commutator motors, 1492 (I)
—— conductivity of metals, 1204 (I)
—— conductor's aluminium, 1403 (I)
———— copper, 1380 (I), 1408 (I)
———— earth continuity, 1392 (I)
———— lightning, 1516 (I)
—— conduits and fittings, 1418 (I)
—— controllers, 1497 (I)
—— cords, flexible, 1383 (I)
—— crane control gear, 813 (II)
—— cranes, 802 (II)
—— cutting, 1293 (I)
—— drills, portable, 1031 (I)
—— dynamometers, 45 (I), 1094 (II)
—— enamelled wire, 1412 (I)
—— equipment of buildings, 1427 (I), 1459 (I), 1520 (I)
———— factories, 1459 (I)
———— motor cars, 1527 (I)
———— ships, 1453 (I), 1528 (I)
—— extra high pressure, regulations, 1427 (I)
—— farming, 1524 (I)
—— field rheostats, 1500 (I)
—— furnaces, 1125 (I), 1129 (I)
———— arc, 1132 (I)
———— power required for, 1133 (I)
———— resistance, 1125 (I), 1129 (I)
—— fuses, 1413 (I)
—— heating of soil, 1525 (I)
—— high pressure mains, 1427 (I)

ELE

Electric high voltage cables, 1411 (I)
—— indicating instruments, 1506 (I)
—— induction motors, 1489 (I)
———— regulators, 1504 (I)
—— industrial trucks, 385 (II), 1274 (II)
—— instrument transformers, 1509 (I)
—— instruments, 1506 (I), 1194 (II)
———— graphic, 1511 (I)
———— Meg and Megger testing, 1194 (II)
—— insulating materials, 1525 (I)
———— oils, 1501 (I)
—— insulator bushings, 1502 (I)
—— insulation testing instruments, 1194 (II)
—— insulators, porcelain, 1413 (I)
—— lamps, 704 (II)
———— architectural, 707 (II)
———— artificial daylight, 705 (II)
———— discharge, 701 (II)
———— fluorescent tubular, 705 (II)
———— gas-filled, lumens rating, 704 (II)
———— mercury vapour, 705 (II)
———— sodium, 705 (II)
———— spacing, 696 (II), 707 (II)
—— lawn mowers, Ransomes, 1277 (II)
—— lifting magnets, 815 (II)
—— lifts, 817 (II)
—— light cables, 1382 (II)
—— lighting of factories, 693 (II), 695 (II)
———— units, 685 (II)
———— conductors, 1516 (I)
—— locomotives, 589 (II), 894 (II)
—— lorries, 385 (II)
—— machinery, British standards for, 1480 (I)
———— efficiency of, 1481 (I)
—— measuring instruments, 1506 (I)
—— mercury arc rectifiers, 1485 (I)
—— Metadyne control, 592 (II)
—— metallic resistance materials, 1379 (I)
—— meters, 1511 (I), 1515 (I)
—— motor coach trains, 591 (II)
—— motors, 1487 (I), 1492 (I) (*see* Motors electric)
—— moving-coil regulator, 1504 (I)
—— National Grid Transmission Lines, 1410 (I)
—— nomenclature, 1377 (I)
—— ovens, 1522 (I)
—— overhead lines, 1407 (I)
———— aluminium, 1411 (I)
———— copper 1407 (I), 1409 (I)
———— regulations, 1420 (I), 1425 (I)
—— porcelain insulators, 1413 (I)
—— power factor improvement, 1505 (I)
———— meters, 1511 (I)
———— stations, fuel consumption of, 7 (II)
—— primary cells, 1476 (I)
—— propulsion of ships, 292 (II), 1528 (I)
—— pyrometers, 1138 (I)
—— railway signalling, 601 (II)
—— railways, 585 (II)
—— rectifiers, mercury arc, 1485 (I)
—— regulators, induction, 1504 (I)
———— moving coil, 1504 (I)
———— voltage, 1504 (I)

Electric resistance, 1379 (I)
———————— heating alloys, 1220 (I)
———————— metals, 1379 (I)
———————— pyrometers, 1138 (I)
———————— temperature effect, 1379 (I)
———————— testing Instruments, 1194 (II)
——— resistivity of metals and alloys, 1134 (I), 1379 (I)
——— rheostats, field, 1500 (I)
——— rubber cables, 1415 (I)
——— salinometer, 1256 (II)
——— ship propulsion, 292 (II), 1528 (II)
—————— equipment, 1453 (I), 1528 (I)
——— signalling in coal mines, 908 (II)
——— soil heating, 1525 (I)
——— spot welding, 1291 (I)
——— standard practice, 1484 (I)
——— starters and controllers, 1497 (I)
——— static balancers in three-phase circuits, 1502 (I)
——— sub-standard ammeters, limits of errors, 1507 (I)
——— switch gear, 1495 (I)
——— switchboards, 1495 (I)
——— symbols, 1378 (I)
——— synchronous motors, 1489 (I)
——— tariffs, off peak, 1529 (I)
——— terminal insulator bushings, 1502 (I)
——— testing instruments, 1194 (I)
——— thermo pyrometers, 1138 (I)
——— traction, 585 (II)
—————— battery, 384 (II)
—————— Metadyne control, 592 (II)
—————— motors, 593 (II)
—————— oil, 385 (II)
—————— on canals, 598 (I)
———————— long - distance railways, 585 (II)
———————— suburban railways, 591 (II)
———————— tramways, 593 (II)
—————— petrol, 385 (II)
—————— switchboard, 383 (II)
—————— trolley bus, 384 (II)
——— tramways, 381 (II)
———————— Board of Trade regulations, 382 (II)
———————— braking, rheostatic and regenerative, 382 (II)
———————— cost of, 588 (II)
———————— tractive resistance on, 382 (II)
——— transformers, 1500 (I), 1509 (I)
———————— booster, 1503 (I)
———————— insulating oils for, 1501 (I)
———————— outdoor for National Grid, 1502 (I)
——— transmission lines, National Grid, 1410 (I)
————————— poles for, 1411 (I)
———————— turbines (steam), 292 (II)
——— trolley buses, 384 (II)
———————— motors and controls, 384 (II)
——— trucks, industrial, 385 (II), 1274 (II)
——— units and nomenclature, 1377 (I)
——— vehicles, battery, 385 (II)
Electric vehicles, battery, operating costs, 386 (II)
——— voltages, standard, 1379 (II)
——— voltmeters, wattmeters, and ammeters, limits of errors in, 1507 (I)
——— ware, 370 (I)
——— welding, 1286 (I), 1183 (II)
—————— building up worn rail crossings by, 492 (II)
——— winding, 897 (II)
—————— plants, mining, 897 (II)
——— wire, enamelled copper, 1412 (I)
——— wiring, fittings, 1418 (I)
—————— Institution of Electrical Engineers, rules for, 1427 (I)
——— X-rays, 1525 (I)
Electrical dynamometers, 45 (I)
——— engineering, 1377 (I), 1194 (II)
——— equipment, methods of declaring efficiency, 1481 (I)
—————— noise due to, 81 (I)
—————— of aircraft, 1455 (I)
——————— buildings, I.E.E. regulations, 1427 (I), 1459 (I)
——————— cranes, 812 (II)
——————— factories, 1459 (I)
——————— motor cars, 1527 (I)
——————— ships, I.E.E. regulations, 1453 (I)
——— heating of soil, 1525 (I)
——— torsion meters, 40 (I)
Electricity Commissioners, regulations, 1420(I)
———————— for overhead lines, 1425 (I)
——— domestic uses of, 1520 (I)
——— generation, standard practice, 1484 (I)
——— in agriculture, 1524 (I)
——— meters, 1511 (I), 1515 (I)
——— supply, regulations of Electricity Commissioners, 1420 (I)
Electro- deposition, saving worn parts by, 1082 (I)
——— galvanising, 1193 (I)
——— magnetic and electro-pneumatic control for motor-coach trains, 592(II)
——— magnets, lifting, 815 (II)
——— mechanical brakes, 1198 (II)
——— percussive welding, 1292 (I)
Electrodes, welding, 1183 (I)
Electrodynamometers, 1094 (II)
'Electrofeeder' multistage pump, 1130 (II)
Electro- galvanisng, 1193 (I)
——— plating, 1193 (I)
Electrolytic capacitors, 967 (II)
Electrostatic precipitation, 48 (II), 1236 (II)
'Elektron' alloy, 1240 (I)
Elevators, gasworks, 635 (II)
——— Giant, mining 935 (II)
——— grain, 624 (I)
——— hydraulic mining, 934 (II)
——— mechanical, mining, (935 II)
——— Ruble, mining, 935 (II)
Elinvar alloy, 1218 (I)
Eloxal process, 1239 (I)
Elliston, Evans & Jackson's electro-mechanical brakes, 1198 (II)
Embankments, labour upon, 38 (I)
—————— settlement of, 561 (I)
—————— shrinkage of materials in, 425(I)

Embankments, slopes of, 433 (I)
—————— volume of earth in, 421 (I)
Embrittlement, caustic, of boiler plates, 51 (II)
Emery polishing, power for, 1115 (I)
——— wheel guards, 1105 (I)
——————— speeds, 1099 (I), 1101 (I)
Emulsifier, 569 (I)
Emulsion paints, 986 (II)
Enamelled copper wire, 1412 (I)
Enamels, 982 (II)
——— cellulose, 986 (II), 988 (II)
——— chemical resistant, 989 (II)
——— synthetic resin, 988 (II)
Endless chain and endless rope haulage, 893 (II)
Energy and work, 25 (I)
Engineering bricks, 364 (I)
——— ropes, 934 (I)
——— Standards Committee's specifications (*see* British Standards)
Engineers as arbitrators, 1033 (II)
——— witnesses, 1029 (II)
——— employment of, 1028 (II)
——— evidence before committees, 1030 (II)
——— fees, 1027 (II)
——— legal notes for, 1027 (II)
——— legal status of, 1027 (II)
——— liability for negligence, 1030 (II)
——— rights *re* inventions, 1029 (II)
———, powers, and duties, under contract, 1030 (II)
Engines, aero (*see* Aeroplane engines)
——— automobile (*see* Automobile engines)
——— Brotherhood-Ricardo, 1266 (II)
——— gas (*see* Gas engines)
——— Gleniffer, 1266 (II)
——— internal combustion (*see* Internal combustion)
——— marine (*see* Marine engines)
——— motor-car (*see* Automobile engines)
——— oil (*see* Oil engines)
——— petrol (*see* Petrol engines)
——— steam (*see* Steam engines)
——— testing plants, 1093 (II)
——— winding (*see* Winding engines).
English lengths, 1 (I)
——— standard screw thread, 89 91 (I)
Entropy, 1302 (I)
Epicyclic gears, 974 (I)
Equation of time, 261 (I)
Equatorial parallax of sun, 265 (I)
——— radius of earth, 265 (I)
Equilibrium of retaining walls, 456 (I)
Erosion of river banks, 579 (I)
Estimate sheet for ships, 282 (II)
Estimating depreciation, 1019 (II)
Ethyl fluid, 454 (II)
Ethylene glycol for aero engines cooling, 455 (II)
Eutectic alloys, 1252 (I)
Evaporation and humidity, 702 (I)
——— annual, 703 (I)
——— boiler, 2684
——— Californian experiments on, 703 (I), 706 (I)
——— Egyptian experiments on, 702 (I)
——— from irrigation canals, 708 (I)
——— soil, 706 (I)
——— water surface, 701 (I)
——— losses by, 705 (I)
——— monthly, 702, 704 (I)
Evaporation of rain, 663 (I)
——— ratio of pan evaporation to open water, 707 (I)
Evaporative cooling of aero-engines, 456 (II)
———power of fuels, 1320 (I)
Evaporators, 1238 (II)
——— refrigerating, 737 (II)
——— Weir, 1238 (II)
Evershed & Vignoles, Ltd., Electrical instruments, 1194 (II)
Exactor, 'Self-Sealing' pipe coupling, 1118 (II)
——— pipe-line change-over switch, 1119 (II)
Excavation, 423, 568 (I)
——— bulking of materials, 425 (I)
——— canal, 717 (I)
——— shrinkage of materials, 425 (I)
Excavations, slopes of, 433 (I)
Excavator machines, 423, 568 (I)
Exhaust gases, 1313 (I)
——— gas fired boilers, 1208 (II), 1214 (II)
——— pipes, 63 (II)
——— silencers, 80 (I)
——— steam, heating by, 673 (II)
——— oil in, 84 (II)
——— steam pipes, 63 (II)
——— valves, 459 (II)
Exhausters, gas, 626 (II)
Expansion, isothermal and adiabatic, 1299 (I), 188 (II)
——— of air by heat, 63 (I)
——— steam, 73 (II), 287 (II)
——— substances by heat, 62 (I)
——— thermal, 62 (I)
——— type thermometer, 1136 (I)
Explosion pressures in petrol engines, 340 (II)
Explosive signals, sea, 656 (I)
Explosives, 919 (II)
——— blasting, 949 (II)
——— mining, 862 (II), 950 (II)
——— properties of typical, 953 (II)
——— safety, 861 (II)
——— sheathed, 863 (II), 951 (II)
Extinguishers, fire, (709 II)
Extraction dust, 48 (II)
Extraction engines, 82 (II)
——— pump, Weir, 1262 (II)
Extruded copper, 1215 (I)
——— sections, aluminium, 1238 (I)
Eyebolts, 171 (I)
Eytelwein's formula for pipe diameters, 734 (I

FACING bricks, 364 (I)
Factories, electricity regulations, 1459 (I)
——— floors, 307 (I)
——— planning and layout, 905 (I)
Factors, 970 (II)
Fahrenheit-Centigrade conversion table, 49 (I)
Falling bodies, 25 (I)
Fan dynamometer, 44 (I)
Fans, centrifugal, 1128 (I)
——— cupola, 1128 (I)
——— electric, 1483 (I)
——— movement of air by, 681 (II), 1285 (II)
——— ventilating, in mines, 903 (II)
——— silencing exhaust from, 80 (I), 683 (I)
——— wing, 1285 (II)
Farming, electric, 1524 (I)

Fatigue, chemical action and, 186 (I)
——— hysteresis in, 188 (I)
——— limit, 183 (I)
——— ——— effect of temperature on, 185 (I)
——— of metals, 181 (I)
——— ——— methods of testing, 182 (I)
——— ——— testing machines, 182 (I)
——— riveted joints, 188 (I)
——— springs, 186 (I)
——— welds, 187 (I)
——— wires, 186 (I)
Feed pumps, 814 (I), 1121 (II), 1127 (II)
——— Clarke Chapman, direct-acting, 1121 (II)
——— governors, Copes, 1244 (II)
——— Weir, 1127 (II)
——— water, 48 (II)
——— ——— conditioning, 52 (II)
——— ——— de-aerator, 1238 (II)
——— ——— heating, 1248 (II)
——— ——— ——— locomotive, 548 (II)
——— ——— oil elimination from, 53 (II)
——— ——— regulators, 1243 (II)
——— ——— ——— Copes, 1243 (II)
——— ——— ——— Crosby, 1245 (II)
——— ——— ——— Mumford, 1248 (II)
——— ——— ——— 'Thermofeed, 1247 (II)
——— ——— ——— Weir's, 1247 (II)
——— ——— softening, 49 (II)
——— ——— ——— zeolite process, 52 (II)
——— ——— treatment of, 48 (II), 1238 (II)
Fees, arbitrator's, 1035 (II)
——— engineer's, 1027 (II)
Feet, decimals of, inches reduced in, 2 (I)
Ferodo Ltd., brake linings, 1153 (II)
——— clutch linings, 1154 (II)
Ferranti moving coil regulator, 1503 (I)
Ferrite, 1140, 1145 (I)
Ferro alloys, 1143 (I)
Fibre machining, 1078 (I)
Fibreglass Ltd., glass silk, 1097 (II), 1103 (II)
Field rheostats, 1500 (I)
——— work, astronomy in, 264 (I)
Filters, for engines and compressors, 234 (II)
——— oil, 115 (II)
——— Pulsometer, 1108 (II)
——— sewage, 787 (II)
——— Visco, 1282 (II)
——— water, 730 (I), 1108 (II)
Filtration, air, 676 (II), 1282 (II)
——— of oil, 878 (I), 1151 (II)
——— water, 729 (I), 1105 (II)
Filtrators Ltd., control of water softening plant, 1105 (II)
Fine screw threads, British standard, 92 (I)
Fire appliances, first-aid, 709 (II)
——— ——— fixed installations, 712 (II)
——— ——— major, 711 (II)
——— ——— mobile secondary aid, 711 (II)
——— bars, 1333 (I), 294 (II)
——— boxes, locomotive, 546 (II)
——— ——— life of, 546 (II)
——— bricks, 371, 377 (I)
——— damp detectors in coal mines, 901 (II)
——— extinction, 709 (II)
——— ——— in mines, 908 (II)
——— extinguishers, portable, 709 (II)
——— hose, 712 (II)
——— ——— couplings, 712 (II)
——— insurance and depreciation, 1029 (II)
——— prevention at sea, 715 (II)
Fire proofing, 376 (I)
——— protection with carbonic acid gas, 715 (II)
——— resistance, reinforced concrete, 398 (I)
——— resisting buildings, 714 (II)
——— ——— cements, 354, 376 (I)
——— ——— doors, 714 (II)
——— ——— floors, 306 (I), 398 (I)
——— ——— materials, 376 (I), 715 (II)
——— ——— paint, 980 (II)
——— streams, 713 (II)
Fireclay bricks, specification for, 372 (I)
Fireclays, 371 (I)
Firedamp detectors in coal mines, 901 (II)
Fires, oil, 715 (II)
Fish bolts and nuts, British standard for, 480 (II)
——— glues, 353 (I)
——— plates, 479 (II)
——— ——— reconditioned, 479 (II)
Fishing beams, 311 (I)
Fittings, boiler, 54 (II)
Fixed capacitors, 965 (II)
——— resistors, 963 (II)
Flags, concrete, 352 (I)
Flame characteristics, oxy-acetylene, 1275 (I)
——— hardening (Shorter process), 1189 (I)
——— machining, 1283 (I)
Flanges of pipes, 108, 788 (I), 68 (II)
Flanging of boiler plates, 33 (II)
Flaps, aircraft, 1763
Flash temperatures, 53 (I)
——— ——— of oils, 53 (I)
Flattened strand ropes, 941 (I)
'Flexatex' non-rubber hose, 1301 (II)
Flexible cables, 1382 (I)
——— couplings, 243 (II), 1269 (II)
——— shaft drives, 1032 (I)
——— tubes, 1273 (I)
'Flextile' roofing material, 1098 (II)
Flight at altitude, 414 (II)
Float method of measuring water flowing in stream, 578, 680 (I)
Floating cranes, 623 (I)
——— docks, 625 (I)
——— ——— cost of, 626 (I)
——— ——— list of largest, 627 (I)
——— ——— notes on, 629 (I)
——— landing stages, 644 (I)
Floats for measuring water flowing in streams, 578, 680 (I)
Flood control, 668 (I)
——— discharges, rainfall, 665 (I)
Floor slabs, reinforced concrete, 393 (I)
Flooring, bridge, 506 (I)
Floors, 305, 398 (I)
——— bridge, 506 (I)
——— concrete, 398 (I)
——— ——— treatment of dusty, 349 (I)
——— equivalent dead loads on, 318 (I)
——— factory, 307 (I)
——— fire resisting, 306, 398 (I)
——— jointless, 305 (I)
——— L.C.C. regulations, 318 (I), 381 (I)
——— reinforced concrete, 381 (I), 398 (I)
——— safe load on, 437 (I)
——— sound proof, 307 (I)
——— terrazzo, polisher 1100 (II)
——— trough for bridges, 507 (I)
——— wooden, 305 (I)
Flotation machines, mining, 917 (II)
——— process, 917 (II)

Flour paste, 359 (I)
Flow nozzles, 752 (I)
—— of air through nozzle, 1116 (I)
—————— orifices, 756 (II)
—————— pipes, 1070 (I), 66 (II)
—— gas in mains, 645 (II)
—— rivers, 578, 668, 716 (I)
—— steam through orifices, 202 (I), 6 (II), 107 (II)
—————— pipes, 64 (II), 66 (II)
—————— turbines, 103 (II)
—— water over notches and weirs, 672 (I), 746 (I)
—————— salt velocity method of measuring, 756 (I)
—————— through blown-out joints, 762 (I)
—————— flumes and channels, 675 (I), 756 (I)
—————— orifices, 671 (I), 745 (I)
—————— pipes, 675 (I), 735 (I), 760 (I), 804 (I)
—— sheets, mining, 918 (II)
Flowmatic regulator, Copes, 1243 (II)
Flue boiler dimension, 13 (II), 44 (II)
—— dust extraction by electrostatic precipitation, 1236 (II)
—— gases, 1312 (I)
—————— analysis of, 1314 (I)
—————— composing of, 1314 (I)
—————— grit concentration in, 46 (II)
—————— heat losses in, 1313 (I)
—————— purification, 46 (II)
—————— quantity of, 1312 (I)
—————— sampling of, 1315 (I)
—————— sulphur, removal from, 48 (II)
—— tubes, 33 (II)
Flumes, flow of water in, 675 (I), 749 (I)
—— standing wave, 749 (I)
—— Venturi, 198 (I)
Fluorescent tubular lamps, 705 (II)
Flushing chambers, 776 (II)
—— sewers, 776 (II)
Fluxes for acetylene welding, 1277 (I)
—— soldering, 1257 (I), 1175 (II)
Flywheels, 90 (II), 230 (II)
—— gas engine, 230 (II)
—— motor car, 356 (II)
—— speed of, 93 (II)
—— steam engine, 90 (II)
Foam fire extinguishing, 715 (II)
Focal length of concave lens, 64 (I)
—————— convex mirror, 64 (I)
'Foliac' boiler graphite, 1240 (II)
—— graphite paint, 1296 (II)
—— lubricting graphite, 1148 (II)
'Fondu' electric cement,. 354 (I)
Fontana project, 697 (I)
Footpath construction, 567 (I)
Foot-valves and strainers, 803 (I)
Footways, 567 (I)
Force, 22 (I)
—— by screw, 20 (I)
—— centrifugal, 24, 499 (I)
—— pumps, 181 (II)
Forced circulation boilers, 15 (II)
Forces, parallelogram of, 30 (I)
—— polygon of, 30 (I)
—— triangle of, 20, 30 (I)
Foreign horse-powers, 826 (II)
—— weights and measures, 10 (I)
Forests and rainfall, 667 (I)
Forging presses, 790 (I)
Forgings, drop, 1121 (I)
—— steam turbine, 151 (II)
—— steel, locomotive, 554 (II)
Forming machines, wire and strip, 1160 (II)
Fort Peck dam, 695, 698 (I)
Fottinger torsionmeter, 39 (I)
Foundations, 326 (I), 435 (I), 459 (I)
—— bridge, 489 (I)
—— building, 326 (I)
—— chemical treatment of, 436 (I)
—— cylinders for, 443 (I)
—— engine, 248 (II)
—— freezing process of sinking, 445 (I)
—— loads on, 436 (I)
—— on different soils, 436 (I)
—— piled, 437 (I)
—— pneumatic, frictional resistance of, 445 (I)
—— retaining wall, 459 (I)
—— safe loads on, 436 (I)
—————— bridge, 489 (I)
—— skin friction on bridge, 489 (I)
—— steam hammer, 1120 (I)
—— wall, 326 (I), 459 (I)
Founding and casting, 1261 (I)
Foundry cupolas, 1127 (I)
—— ladle crane hooks, 937 (I)
—— pig iron, 1142 (I)
Four-stroke cycle, 197 (II)
Fourth powers, table of, 1074 (II)
Fractional h.p. motors, 1482 (I)
Frames, aircraft, 432 (II)
—— engine, 86 (II)
—— motor car, 369 (II)
Framing and covering, steel roof, 233 (I)
Francis formula for weirs, 746 (I)
François cementation process, 870 (II)
Free-cutting steels, 1162 (I)
'Freon 12,' refrigerant, 722 (II)
Freezing mixtures, 53 (I)
—— prevention of, 53 (I)
—— process of sinking foundations, 445 (I)
—————— mines, 869 (II)
—— temperatures, 50 (I)
French bolts, 101 (I)
—— catch, 54 (I)
—— nuts, 101 (I)
—— screw threads, 95 (I)
—— standard bolts and nuts, 101 (I)
Frequency tables, radio, 958 (II)
—— modulation, 959 (II)
—————— receivers for, 959 (II)
—————— experimental transmissions, 960 (II)
Fret-sawing machines, 1043 (I)
Friant dam, 697 (I)
Friction clutches, 929 (I)
—— coefficients of materials, 21 (I)
—— experiments in, on swing bridges, 529 (I)
—— in gas mains, 646 (II)
—— leather packing, 808 (I)
—— lubrication of swing bridges, 529 (I)
—— skin, of bridge foundations, 489 (I)
Frictional resistance of pneumatic foundations, 445 (I)
Froude aircraft engine testing plant, 1094 (II)
—— dynamometers, 1093 (II)

Froude electrodynamometers, 1094 (II)
—— momentum theory, 404 (II)
—— water dynamometer, 41 (I), 43 (I), 1093 (II)
Frustum, 19 (I)
—— pyramidal, 19 (I)
Fry's anti-friction alloys, 1244 (I)
—— heat transmission formulæ, 28 (II)
—— solders and soldering fluxes, 1175 (II)
Fuel, acetylene, 1367 (I)
—— anthracite, 1327 (I)
—— aviation, 452 (II)
—— benzine, 1352 (I)
—— briquette, 1330 (I)
—— calorific capacity of, 55 (I)
—— —— values of, 1309 (I), 1316 (I), 1365 (I), 20 (II)
—— charcoal, 1330 (I)
—— coal, 1315 (I)
—— —— dust petroleum mixtures, 1363 (I)
—— —— tar, 1323 (I), 1356 (I)
—— coalite 1325 (I)
—— coke, 1321 (I), 633 (II)
—— —— as a boiler, 1326 (I)
—— combustion of, 1310 (I)
—— —— air required for, 1361 (I)
—— —— draught required for, 22 (II)
—— consumption of, in locomotives, 562 (II)
—— —— in electric power stations, 10 (II)
—— —— —— heating buildings, 665 (II)
—— —— ships, 282 (II), 299 (II)
—— Diesel engine, 1355 (I)
—— economisers, 17 (II)
—— ethyl fluid, 454 (II)
—— evaporative power of, 1320 (I)
—— gas engine, 1368 (I), 200 (II)
—— —— oil, 1354 (I)
—— —— turbine, 265 (II)
—— gaseous, 1365 (I), 1366 (I)
—— —— calorific value of, 1365 (I)
—— —— for engines, 1368 (I), 200 (II)
—— heating boiler, 644 (II)
—— hydrocarbon, combustion of, 1312 (I)
—— hydrogen, 1367 (I)
—— internal combustion engine, 200 (II)
—— kerosene, 1354 (I)
—— lignite, 1329 (I)
—— liquid, 1321 (I
—— —— burners, 1361 (I)
—— —— for locomotives, 561 (II)
—— —— furnace firing with, 1360 (I)
—— —— vapour tension in, 192 (II)
—— marsh gas, 1367 (I)
—— motor, 327 (II)
—— oil, combustion of, air required for, 1361 (I)
—— —— distribution to cylinders, 349 (II)
—— —— for heavy oil engines, 1362 (I)
—— —— —— marine engines, 298 (II)
—— —— —— boilers, 298 (II)
—— —— the Navy, 1361 (I)
—— —— properties of, 1357 (I)
—— —— residual, 1354 (I)
—— olefiant gas, 1368 (I)
—— peat, 1329 (I)
—— petrol, 1352 (I)
—— petroleum, 1351 (I)
—— —— tar mixtures, 1363 (I)
—— pulverised coal, 1337 (I), 1188 (II)
Fuel pumps for heavy oil engines, 1358 (I), 343 (II)
—— solid, 1315 (I)
—— spontaneous ignition temperatures, 1363 (I)
—— tar oil, 1323 (I)
—— wood, 1330 (I)
Fuels, 1309 (I)
—— and their relative costs, 663 (II)
Furnaces, 1125 (I)
—— Bennis self-cleaning, 1220 (II)
—— 'Bigelow' suspended walls for, 1236 (II)
—— blast, 1126 (I)
—— boiler, 33 (II)
—— —— radiant heat transmission in, 25 (II)
—— bricks, 373 (I)
—— classification, 1125 (I)
—— coke, 1128 (I)
—— construction, 1134 (I)
—— Crosthwaite, 1222 (II)
—— crucible, melting, 1125 (I), 1128 (I)
—— cupolas, 1127 (I)
—— efficiency, 1135 (I)
—— electric, 1125 (I), 1129 (I)
—— hardening control, 1160 (I)
—— hearth, 1125 (I)
—— lining for, 373 (I), 1236 (II)
—— 'Liptak,' 1236 (II)
—— liquid fuel firing, 1360 (I)
—— marine boiler, 38 (II), 294 (II)
—— metallurgical, 1125 (I)
—— muffle, 1128 (I)
—— oil-fired, 1129 (I), 1360 (I)
—— producer gas, 1129 (I)
—— reverberatory, 1125 (I), 1128 (I)
—— shaft, 1125 (I)
—— temperature, control of, 1125 (I)
—— tilting, 1129 (I)
—— ventilation of mines, 903 (II)
Fuses, electrical, 1413 (I), 968 (II)
Fusible alloys, melting points, 52 (I), 1247 (I)
—— metals, 1246 (I)
—— solder, 1178 (II)
Fusion of metals, latent heat of, 1135 (I)
—— heat and electrical energy required, 1135 (I)
—— pyrometers, 1136 (I)

GAKO gas burner, 1282 (II)
Galvanised iron corrugated sheets, 239 (I)
—— —— roofing, paint for, 239 (I), 993 (II)
—— steel corrugated sheets, 238 (I)
Galvanising, 1192 (I)
Gas, acetylene, 646 (I), 1367 (I), 1370 (I), 651 (II)
—— adiabatic expansion of, 188 (II)
—— air and oxygen necessary for combustion, 1366 (I)
—— ascension pipes, 624 (II)
—— benzol from, 1324 (I), 638 (II)
—— Blau, 646 (I)
—— boilers, waste heat, 623 (II), 1213 (II)
—— Boyle's law, 188 (II)
—— buoys, 645 (I)
—— burners, 'Gako,' 1282 (II)

GAS

Gas, calorific value of coal, 1368 (I), 645 (II), 649 (II)
—— caps of safety lamps, 905 (II)
—— carbon monoxide, 1367 (I)
—— carbonic acid, fire protection by, 909 (II)
—— carbonising plant, 618 (II)
—— Charles' law, 188 (II)
—— cleaning by electrostatic precipitation, 1236 (II)
—— coal, 1368 (I), 615 (II)
———— air necessary for combustion of, 1366 (I), 645 (II)
———— calorific value of, 1368 (I), 645 (II), 649 (II)
———— cost of purification of, 635 (II)
———— dilution of, with producer gas, 621 (II)
———— storage, 632 (II)
—— coke oven, 1325 (I), 1329 (I), 1369 (I)
———— storage, 633 (II)
—— compression, 188 (II)
———— ratios, 190 (II)
—— compressors, 1288 (II)
—— condensers, 625 (II)
—— consumption of, per head of population, 618 (II)
—— cylinders, 656 (II)
—— data relating to, 615 (II)
—— dehydration, 639 (II)
—— distillation, products of, 634 (II)
—— dry purification plant, 628 (II)
—— engines, 204 (II) (*see also* Internal combustion engines)
———— combustion, 254 (II)
———— compression ratios, 190 (II)
———— connecting rods, 212 (II)
———— cooling, 198 (II)
———— cycles, 197 (II)
———— cylinder heads, 215 (II)
———— efficiency, 194 (II), 195 (II)
———— firing methods, 200 (II)
———— flywheels, 230 (II)
———— fuels for, 1368 (I), 200 (II)
———— gaseous fuels for, 1368 (I)
———— high speed, 210 (II)
———— lubrication, 878 (I)
———— parts of, 204 (II)
———— piston cooling, 206 (II)
———— rofs, 206 (II)
———— pistons, 207 (II), 210 (II)
———— starting, 207 (II)
———— stresses in, 204 (II)
———— timing, 216 (II)
———— types of, 197 (II)
———— valves of, 216 (II)
—— exhaust, 1313 (I)
—— exhausters, 626 (II)
—— expansion, adiabatic and isothermal, 1299 (I), 189 (II)
—— fired boilers, 1214 (II)
—— flash temperature, 53 (I)
—— flow of, in mains, 645 (II)
—— flue, 1312 (I) (*see* Flue gases)
—— formulæ relating to perfect, 1299 (I)
—— high temperatures, specific heat of, 61 (I)
—— holder tanks, 630 (II)
—— holders, 630 (II)
———— pressure, 631 (II)
———— waterless, 631 (II)
—— hydrogen, 1367 (I)
—— impurities in, 636 (II)

GAS

Gas, isothermal expansion of, 1299 (I), 188 (II)
—— laws, 187 (II)
—— liquor, 644 (II)
—— mains, 646 (II)
———— friction in, 646 (II)
—————— Pole's formula, 645 (II)
—————— Unwin's formula, 645 (II)
———— hydraulic, 624 (II)
—— marsh, 1367 (I)
—— masks, 1300 (II)
—— memoranda, 644 (II)
—— meters, 198 (I), 648 (II), 629 (II)
———— drum type, 198 (I)
———— heat injection, 199 (I)
———— hot wire anemometer, 201 (I)
———— orifice, 200 (I)
———— rotary type, 200 (I)
———— shunt type, 200 (I)
———— station, 629 (II)
———— Venturi type, 199 (I)
—— naphthalene removal, 638 (II)
—— natural, 1368 (I)
—— oil, 1354 (I)
—— Olefiant, 1368 (I)
—— Parliamentary regulations in reference to, 615 (II)
—— perfect, 1299 (I)
—— petrol air, 654 (II)
—— pipes, 112 (I), 114 (I), 116 (I)
———— British standard, 112 (I)
———— cast iron, 114 (I)
———— dimensions of, 114 (I)
———— weight of, 140 (I), 145 (I)
—— producer, 1370 (I)
———— composition of, 1370 (I), 623 (II)
———— heat value of, 1371 (I)
—— properties of, 54 (I), 56 (I)
—— purification, 627 (II), 635 (II)
———— of power station, 48 (II)
—— recorders, 1315 (I), 1193 (II)
—— referee's tests, 649 (II)
—— retort house, 625 (II)
—— retorts, 374 (I), 619 (II)
———— ascension pipe joints, 624 (II)
———— bench fittings, 624 (II)
———— foul main, 625 (II)
———— heating of, 621 (II)
———— moulded, 374 (I)
———— specification for, 374 (I)
———— vertical, 619 (II)
—— scrubbers and washers, 627 (II)
—— sold in Great Britain, 618 (II)
—— specific volume, 191 (II)
—— storage capacity, 630 (II)
———— in cold storage rooms, 731 (II)
—— sulphur, organic, removal from town gas, 639 (II)
—— sulphuretted hydrogen removal, 637 (II)
—— supply regulations, 615 (II)
—— tar, 641 (II)
———— removal, 625 (II)
—— tests, 649 (II)
—— thermal conductivity, 63 (I)
—— town, contamination of, in storage, 641 (II)
—— turbines (*see* Turbines, gas), 253 (II)
—— undertakings in Great Britain, Acts, 616 (II)

Gas, undertakings in Great Britain, statistics, 617 (II)
—— volume of oxygen and air to burn, 645 (II)
—— washers and scrubbers, 627 (II)
—— waste heat, 1372 (I)
—— water, 642 (II)
—— works coal and coke storage, 632 (II)
———— condensers, 625 (II)
—————— atmospheric, 626 (II)
—————— water-cooled, 626 (II)
———— conveyors and elevators, 635 (II)
———— exhausters, 626 (II)
———— fire bricks used on, 373 (I)
———— grate and chimney area, 622 (II)
———— mains, 645 (II)
———— Parliamentary Acts, 615 (II)
———— retort house of, 625 (II)
———— temperatures, 623 (II)
———— terms, 645 (II)
Gaseous case-hardening materials, 1188 (I)
—— fuels, 1366 (I)
Gases, mine, 901 (II)
—— specific gravities of, 56 (I), 120 (I), 125 (I)
Gasoline, 1352 (I)
Gates, dock, 617 (I)
—— head, 722 (I)
Gauge glasses, 58 (II)
Gauges, hydraulic pressure, 799 (I)
—— orifice, 754 (I)
—— railway, 499 (II)
—— rain, 669 (I)
—— water (boiler), 1241 (II)
———— defects in, 58 (II)
—— wire and sheet metal, 85 (I)
———— drill, 1057 (I)
Gauging gear teeth, 1000 (I)
—— streams, 674 (I)
Gear, bevel, 987 (I)
—— cast-iron, 981 (I)
—— chordal pitch, 975 (I)
—— circular pirch, 975 (I)
—— cutters, involute, 973 (I)
—— cycloidal teeth, 974 (I)
—— double helical, 993 (I), 160 (II)
—— epicyclic, 974 (I)
—— face width, 970 (I)
—— generating process, 986 (I)
—— grinding, 994 (I)
—— hardening of, 982 (I)
—— helical, 990 (I)
—— Hindley or globoidal worm, 1001 (I)
—— internal, 980 (I)
—— lubrication, 1008 (I)
—— mild steel, 981 (I)
—— motor-car gear box, 365 (II)
—— non-resonant, 981 (I)
—— pinions, 981 (I)
—— pitch, circle diameter, 1009 (I)
———— line velocity of a, 975 (I)
—— power transmitted, 975 (I)
—— reduction, 996 (I), 160 (II), 290 (II)
—— rolling mill, 972 (I)
—— slide valve, 1470
—— spiral bevel, 989 (I)
—— spur, 967 (I), 975 (I), 365 (II)
—— strength of, 977 (I), 979 (I), 994 (I)
—— teeth, cycloidal, 974 (I)
———— failure, 982 (I)
———— gauging, 1000 (I
Gear, teeth, grinding of, 984 (I)
———— involute, setting out, 970 (I)
———— lapping, 987 (I)
———— load, 975 (I)
———— Logue, 972 (I)
———— machine moulded, 972 (I)
———— number of, 999 (I)
———— profiles, 968 (I)
———— Sellers, 972 (I)
———— strength, 979 (I)
—— triple helical, 996 (I)
—— turbine, 996 (I)
———— reduction, 996 (I), 160 (II), 290 (II)
—— variable speed, 1114 (II)
———— hydraulic, 1114 (II)
—— wheel arms, 972 (I)
———— bosses, 972 (I)
———— rims, 972 (I)
———— turbines, 290 (II)
Gearing, 967 (I)
—— armature shaft, 980 (I)
—— back lash, 970 (I)
—— bevel, 987 (I)
—— crane, 809 (II)
—— critical speeds of, 981 (I)
—— definitions, 967 (I)
—— helical, 990 (I)
—— interference, 969 (I)
—— Lewis formula, 976 (I)
—— lubrication, 1008 (I)
—— mill, 972 (I)
—— noises, 981 (I)
—— pitch circle diameters, 1009 (I)
—— power transmitted by, 975 (I)
—— spiral bevel, 989 (I)
—— spur, 967 (I), 969 (I), 365 (II)
———— corrections for interference in, 969 (I)
—— tooth, 967 (I)
—— turbine reduction, 996 (I), 160 (II), 290 (II)
—— variable speed, 1114 (II)
—— worm, 1001 (I)
———— applications, 1007 (I)
———— nomenclature, 1003 (I)
Genissiat dam, 697 (I)
Genoa breakwater, 609 (I)
Geodetic formulæ, 271 (I)
Geographical measures, 1 (I)
Geometrical and inertial properties of squares, octagons, hexagons and circles, 31 (I)
Geophysics for the location of minerals, 862 (II)
German canals, 587 (I), 595 (I)
—— waterways, 598 (I)
German waterways, capacities of, 587 (I)
Gesner protective process, 1194 (I)
Giant elevators, mining, 935 (II)
Giants or Monitors, 933 (II)
Girder bridge specification, 495 (I), 497 (I), 509 (I)
—— bridges, lattice, 479 (I), 517 (I)
———— plate, 478 (I), 509 (I), 515 (I)
———— standard loadings for, 482 (I)
Girders, bending moments of, 216 (I), 225 (I)
—— bridge, 478 (I)
—— circular, 531 (I)
—— continuous, 223 (I), 225 (I)
—— cross, 512 (I)

Girders, economical proportions of bridge 478 (I)
——— joints and splices, 510 (I)
——— reinforced concrete, 309 (I)
——— riveted, 229 (I)
——— steel, 241 (I)
——— ——— deflection of, 216 (I), 523 (I)
——— stresses on, 211 (I)
——— supporting combustion chambers, 39 (II)
——— weights of, 492 (I)
Girder-work, riveted joints in, 229 (I)
Girling brake-system, 374 (II)
G.K.N. steel sleepers, 483 (II)
Gladstone docks, Liverpool, 614 (I)
Glands, water, steam turbine, 131 (II)
Glasgow refuse disposal, 1350 (I)
Glass, coloured, absorptive powers of, 654 (I)
——— silk for heat insulation, 1103 (II)
——— ——— sound insulation, 1097 (II)
——— steam gauge, 58 (II)
——— thermal expansion, 62 (I)
Glasses, gauge, 58 (II)
Glazed bricks, 364 (I)
Glazing of lighthouses, 653 (I)
Gleniffer Engines, Ltd., cold starting engines, 1266 (II)
Gloss paints, 982 (II)
Glue, holding power of, 180 (I)
Glues, 354 (I)
Goggles for welding, 1276 (I)
Gold and silver tables, 932 (II)
——— dredging, 936 (II)
——— milling, 928 (II)
——— ——— cyanide process, 928 (II)
——— mills, cost of, 929 (II)
——— mines, 868 (II)
——— stamps, 929 (II)
——— wire, gauge for, 86 (I)
Gooch's link motion valve gear, 99 (II)
Gourley and Crimp formula for weirs, 747 (I)
Governor, Copes pump, 1244 (II)
Governors, 93 (II)
——— boiler feed pump, 1244 (II)
——— centrifugal, 24 (I)
——— internal combustion engine, 250 (II)
——— relay, 93 (II)
——— retort house, 625 (II)
——— steam engine, 93 (II)
——— turbine (water), 844 (I)
——— ——— power of, 844 (I)
Grab, 425 (I)
——— or grapple dredger, 634 (I)
Grabs, capacity and weights of, 807 (II)
Gradients, railway, 512 (II)
——— ——— resistance due to, 529 (II)
Grain elevators, 624 (I)
——— growth and size control (steel), 1168 (I)
Grand Coulee dam, 696 (I)
Granular substances, bulk densities, 125 (I)
——— ——— specific gravity of, 124 (I)
Graphic instruments, 1511 (II)
Graphite, for boilers, 'Foliac,' 1240 (II)
——— lubrication, 880 (I), 1148 (II)
——— paint, 994 (II), 1296 (II)
——— Products, Ltd., 1148 (II), 1240 (II), 1296 (II)
Graphited bearings, 1139 (II)
Graphitisation, 1144 (I)
Grapple or grab dredger, 634 (I)
Grate area, gasworks, 622 (II)
——— in locomotives, 545 (II)
Graving docks, 625 (I)
——— cost of, 626 (I)
Gravity, centre of, 17 (I)
——— dams, 683, 696 (I)
——— docks and quay walls, 615 (I)
——— motion under, 22 (I)
——— operated despatch tubes, 1302 (II)
——— specific (*see* Specific gravity)
Grease lubrication, 1150 (II)
Greases, 995 (II)
Great Britain, gas sold in, 617 (II)
——— Ordnance maps, 254 (I)
——— railway electrification, 588 (II)
——— roads in, 544 (I)
——— Western Railway, locomotives, 518 (II), 520 (II)
——— ——— rolling stock, 572 (II), 574 (II)
——— ——— train control, 609 (II)
Green and Boulding, Ltd., Buffalo and Bison injectors, 1257 (II)
——— ——— Buffix separator, 1258 (II)
Grinding, 1098 (I)
——— allowances, 1111 (I)
——— cylindrical, 1107 (I)
——— drills, 1055 (I)
——— dry versus wet, 1107 (I)
——— external, 1107 (I)
——— gear teeth, 984 (I)
——— ——— formed wheel process 984 (I)
——— internal, 1109 (I)
——— lubrication, 1107 (I)
——— machines, 1029 (I)
——— ——— power required for, 1035 (I)
——— surface, 1110 (I)
——— tools, 1055 (I)
——— wet versus dry, 1107 (I)
——— wheel guards, 1105 (I)
——— wheels, 1098 (I)
——— ——— classification of, 1099 (I)
——— ——— composition of, 1098 (I)
——— ——— dressers, 1106 (I)
——— ——— dust collection, 1106 (I)
——— ——— flanges, 1104 (I)
——— ——— grit sizes, 1099 (I)
——— ——— guards, 1105 (I)
——— ——— lubrication of, 1107 (I)
——— ——— mounting of, 1102 (I)
——— ——— rests, 1106 (I)
——— ——— selection, 1101 (I)
——— ——— sizes, 1099 (I)
——— ——— speed, 1098 (I), 1101 (I)
——— ——— spindle size, 1102 (I)
——— ——— storage, 1102 (I)
Grindstones, 1098 (I)
——— artificial, 1099 (I)
——— to true, 1099 (I)
Grit concentration in flue gases, 1420
Grizzle bricks, 364 (I)
Ground sills, 582 (I)
——— sloping, chaining over, 253 (I)
Grouting roads, 565 (I)
——— work for dams, 693 (I)
Groynes, 580 (I)
——— on River Missouri, 581 (I)
Guard for abrasive wheels, 1105 (I)

Guides and rubbing ropes, 950 (I)
Guns, spray, A. C. Wells & Co., Ltd., 1296 (II)
Gun-metal, 1223 (I)
——— weight of castings in, 154 (I)
Gunpowder, 949 (II)
Gunter's surveying chain, 1 (I)
Gypsum cements, 354 (I)
Gyration, radius of, 22, 26 (I)

HÆMATITES, 1141 (I), 859 (II)
Haigh testing machine, 182 (I)
Hammers, 1078 (I), 1166 (II)
——— Brett's drop, 1120 (I), 1122 (I), 1166 (II)
——— caulking, 1078 (I)
——— foundations, 1120 (I)
——— Massey's steam, 1118 (I)
——— ——— pneumatic power, 1120 (I)
——— pneumatic power, 1120 (I)
——— ——— Alldays & Onions Ltd., 1169 (II)
——— ——— riveting, 1078 (I)
——— steam, 1117 (I), 1166 (II)
——— ——— dimensions, weights, etc. 1118 (I)
——— ——— energy and velocity of a blow, 1117 (I)
——— ——— steam consumption, 1118 (I)
Hand cranes, 802 (II)
Handling machinery, hydraulic, 798 (I)
Hangers, steam pipe, 63 (II)
Hans Renold metal protection process, 1194 (I)
Harbours, 605 (I)
Hard soldering, 1258 (I), 1179 (II)
Hardening and tempering steel, 1069 (I), 1160 (I) 1179 (I), 1187 (I), 1196 (I)
——— case, 962 (I), 1187 (I)
——— furnaces, 1150 (I)
——— gears, 982 (I)
——— high-speed tools, 1069 (I)
——— nitrogen, 983 (I), 1189 (I)
——— Shorter process, 1189 (I)
——— Tocco process, 1189 (I)
Hardness numbers, Brinell, 1089 (I), 1090 (I), 1190 (I)
——— of materials, 1088 (I), 1191 (I)
——— water, 731 (I), 49 (II)
——— scales, approximate comparison of, 1094 (I)
——— ——— relations between, 1190 (I)
——— test for drilling, 1090 (I)
——— tester, Herbert, 1092 (I)
——— ——— Rockwell, 1093 (I)
——— ——— sclerometer, 1090 (I)
——— ——— Shore scleroscope, 1089 (I)
——— values of metals, 1094 (I), 1190 (I)
Hardy Spicer needle bearing propeller shafts and universal joints, 1272 (II)
Haulage canal, 597 (I)
——— cost of, 890 (II)
——— endless chain and endless rope, 893 (II)
——— mining, 889 (II)
——— pulleys, 898 (II)
——— self-acting incline, 892 (II)
——— tail, 893 (II)
——— tubs, 891 (II)
——— underground, cost of, 890 (II)
Haulage wire ropes for, 951 (I)
Hauling and winding, ropes for, 942 (I), 951 (I)
Hazen and Williams formula for flow in pipes, 758 (I)
Head gates, 722 (I)
——— races of turbine (water), design of, 722 (I)
Headgears, mining, 898 (II)
Hearth furnaces, 1125 (I)
Heat, 49 (I)
——— absorbing factors, 1135 (I)
——— and combustion, 54 (I)
——— balance, 21 (II)
——— cycle, 197 (II)
——— drop, adiabatic, in steam turbines, 118 (II)
——— engine, the efficiency of a, 1301 (I)
——— expansion of air by, 63 (I)
——— graphic calculation of, 30 (II)
——— insulating materials, 375 (I), 68 (II), 1102 (II)
——— ——— aluminium foil, 68 (II)
——— ——— 'Dextramite,' 1103 (II)
——— ——— glass silk, 1103 (II)
——— ——— magnesia, 1102 (II)
——— ——— value of roofing material, 299 (I)
——— latent, 62 (I)
——— ——— of fusion of metals, 1135 (I)
——— loss of, from steam pipes, 66 (II)
——— losses in boilers, 20 (II)
——— ——— flue gases, 1313 (I)
——— ——— heating buildings, 659 (II)
——— of combustion, table of, 55 (I)
——— producing factors, 1135 (I)
——— radiation pyrometers, 1137 (I)
——— required for the fusion of metals 1135 (I)
——— required for heating buildings, 659 (II)
——— resistance of cast irons, 1155 (I)
——— resisting alloys, 1219 (I), 1252 (I)
——— ——— paints, 980 (II)
——— ——— steels, 1201 (I)
——— specific, 61 (I)
——— transfer in cross flow, 31 (II)
——— transference, 738 (II)
——— ——— through partitions, 310 (I)
——— transmission in boilers, 24 (II)
——— ——— boiler tubes, 25 (II), 31 (II)
——— ——— radiant in boiler furnaces, 25 (II)
——— ——— surface condensers, 170 (II)
——— ——— through roofing materials, 300 (I)
——— treatment of aluminium alloys, 1234 (I)
——— ——— gas cylinders, 656 (II)
——— ——— metals, 1256 (I)
——— ——— rails, 478 (II)
——— ——— steels, 1160 (I), 1167 (I), 1180 (I)
——— ——— wrought iron, effect of 1141 (I)
——— units of, 49 (I)
——— value of producer gas, 1371 (I)
——— waste, recovery of, 1372 (I), 623 (II)
——— tubular 28, (II)

Heater, 'Usco' air, 1226 (II)
Heaters, air, for boilers, 17 (II), 1226 (II)
—— efficiency of, 18 (II)
—— boiler feed water, 1248 (II)
—— Ljungström, 18 (II)
Heating, 659 (II)
—— boilerhouse requirements, 666 (II)
—— buildings, 659 (II)
———— boiler-holse requirements, 666 (II)
———— boilers for, 663 (II), 1217 (II)
———— boiler power required, 663 (II)
———— contract temperatures and their equivalents, 663 (II)
———— fuels and their relative costs, 663 (II)
—— buildings, horticultural, 1525 (I)
—— chimney sizes, 667 (II)
—— electric, of soil, 1524 (I)
—— exhaust steam, 673 (II)
—— feed water, 548 (II), 1248 (II)
—— fuel consumption, 665 (II)
—— fuels and their relative costs, 663 (II)
—— hot water, 668 (II), 672 (II)
———— boilers, 672 (II)
———— storage and distribution 672 (II)
—— mains, size of, 671 (II)
—— pipes, 669 (II)
—— railway carriages by steam, 2212
—— soil by electricity, 1524 (I)
—— steam, 670 (II)
—— surface of locomotives, 516 (II)
———— marine boilers, 291 (II)
———— radiators, 661 (II)
—— systems, 669 (II)
—— vacuum system, 671 (II)
Heenan & Froude's aero-engine testing plant, 1094 (II)
———— air cooler, 1198 (II)
————dynamometer, 13 (I), 1093 (II)
———— dynamatic dynamometer, 1094 (II)
———— electric dynamometers, 1094 (II)
———— forming machines for strip and wire, 1160 (II)
———— hangar stand, 1095 (II)
———— oil coolers for heat treatment of steel, 1174 (II)
———— refuse destructors, 1911 (II)
———— water cooler, 1273 (II), 1287 (II), 1292 (II)
Hefner candle, 685 (II)
Height of an object, to measure, 253 (I)
Heights, table of dip and distance of horizon, 225 (I)
Helical gearing, 990 (I), 160 (II)
——springs, 895 (I), 898 (I), 369 (II)
———— for locomotives, 553 (II)
—— tooth-gearing, 990 (I)
Helium, 56 (I)
Helmets, smoke, 1299 (II)
Hemp ropes, strength of, 176 (I)
Hennebique system of concrete construction, 398 (I)
Herbert, E. G., Ltd., hardness tester, 1092 (I)
Herbert, E. G., Ltd., 'Rapidor' sawing equipment, 1161 (II)
Hexagons, geometrical and inertial properties, of, 31 (I)
Hiduminium R.R. alloy, 1233 (I)
High-speed bearings, 863 (I)
—— drilling machines, 1028 (I)
—— drills, 1069 (I)
—— engines, 1591
—— steels, 1067 (I)
—— tool steel, 1067 (I), 1160 (I)
———— hardening, 1069 (I), 1160 (I)
Highways) (*see* Roads)
Hiley's formula for safe load on piles, 411 (I)
Hindley or globoidal worm gear, 1001 (I)
Hirn's rope brake, 41 (I)
Hodgson's fan dynamometer, 44 (I)
Hoffmann Manufacturing Co., Ltd., bearings, 1140 (II)
Hoists, wire ropes for, 939 (I)
Holding power of glue, 180 (I)
———— spikes in wood, 179 (I)
Hollow blocks, 369 (I)
Holyhead breakwater, 605 (I)
Honigmann shaft-sinking method, 870 (II)
Hook's coupling, 2172 (II)
Hooks, crane, 937 (I)
Hoop-iron, dimensions and weight of, 153 (I)
Hopkinson-Thring torsion meter, 39 (I)
Hopper barges, 638 (I)
—— service dredging, 638 (I)
Horizon, dip of, 255 (I)
Horizontal force of earth's magnetism, 260 (I)
—— sun-dial, 272 (I)
Horse-power, 38 (I), 75 (II), 187 (II)
—— automobile, 325 (II)
—— brake, 75 (II)
—— for haulage, 889 (II)
———— machine tools, 1032 (I)
—— in compressed air, 754 (II)
—— indicated, 75 (II)
—— of shafting, 889 (I)
———— steam engines, 75 (II)
———— water, 720 (I)
—— skin, effective, of ships, 280 (II)
—— transmitted by belts, 910 (I) 963 (I)
———— gears, 975 (I), 1007 (I)
—— wave, effective, of ships, 280 (II)
—— worm gear, 1007 (I)
Horse traction on canals, 597 (I)
Horses, power of, 35, 37 (I)
Horticultural heating, 1524 (I)
Hose, air, 1081 (I)
—— couplings, 1081 (I), 712 (II)
—— fire, 712 (II)
—— reels, 1200 (II)
Hot bearings, cooling compound for, 996 (II)
—— brass pressings, 1214 (I)
—— forgeable brass, 1214 (I)
—— water boilers, 672 (II), 1217 (II)
———— and boiler power required, 672 (II)
———— heating by, 668 (II)
———— supply, 672 (II)
—— wire anemometer, 201 (I)
Hour angle, to find, 271 (I)

House drainage, 791 (II)
——— heating, 659 (II)
Houses, all electric, 1521 (II)
——— struts between, 450 (I)
Housings, ball and roller bearing, 860 (I)
Howden, James, & Co., Ltd., Howden-Ljungström air preheater, 1232 (II)
——— Centicell dust collector, 1283 (II)
Hrabak's formula for bending stresses, 945 (I)
Huffman dam, 693 (I)
Humidity, 678 (II)
——— and evaporation, 702 (I)
Hutchison automatic penetrometer, 550 (I)
Hyatt dynamometer, 39 (I)
Hydraulic accumulators, 772 (I)
——— actuating machinery, 792 (I)
——— alleviators, 785 (I)
——— control waves, 777 (I)
——— dredgers, 635 (I)
——— dynamometer, 'Froude,' 39 (I), 43 (I), 1093 (II)
——— elevators, mining, 934 (II)
——— feeds for machine tools, 1022 (I)
——— fill dams, 685, 693 (I)
——— handling machinery, 798 (I)
——— intensifiers, 776 (I)
——— leathers, 808 (I)
——— lining and slewing machinery for bridges, 530 (I)
——— limes, 339 (I)
——— mains, gas, 624 (II)
——— ——— velocity in, 788 (I)
——— ——— ventilation of, 788 (I)
——— mean depth, 709 (I)
——— mining, 933 (II)
——— monitors (mining), 933 (II)
——— packings, 799 (I)
——— pipe work, 785 (I)
——— pipes, thickness of, 114 (I)
——— power development, 719 (I)
——— presses, 789 (I)
——— pressure gauges, 799 (I)
——— rams, 831 (I)
——— remote controls, 795 (I)
——— sluices and lock gates, 793 (I)
——— test pressures, 114 (I)
——— transmission of power, 767 (I), 163 (II), 1110 (II)
——— valves, 777, 1273 (I)
——— variable speed-gear, 1110 (II), 1114 (II)
——— watertight bulkhead doors, 794 (I)
Hydrocarbon fuels, combustion in, 1312 (I)
Hydrodynamometer, Perrodil's, 681 (I)
Hydrogen, 1367 (I)
——— combustion of, 60 (I)
——— properties of, 56 (I)
——— sulphuretted, 637 (II), 650 (II)
——— welding, 1289, 1293 (I)
Hydrometer scales, 125 (I)
Hydro-pneumatic accumulators, 774 (I)
Hydrostatic intensifier, 776 (I)
Hydrox explosive, 951 (II)
Hygrometer, wet and dry bulb, 678 (II)
Hysteresis in fatigue, 188 (I)

ICE making, 731 (II)
Ignition, auromobile engine, 357 (II)
——— temperatures, 53 (I)
——— ——— spontaneous, 1363 (I)
Illumination, 685 (II). (*See also* Lighting)
——— calculations, 689 (II)
——— coefficients of utilization, 701 (II)
——— factory and workshop, 701 (II)
——— lighthouse, 651 (I)
——— reflectors for, 696 (II)
——— requirements for, 688 (II)
——— room index, 698 (II)
——— standards of, 691 (II)
——— terms of, 685 (II)
——— values, 691 (II)
Impact of bodies, 25 (I)
——— effect, Bridge Stress Committee's recommendations for, 502 (I)
——— on railway bridges, 500 (I)
Impeller vanes, 820 (I)
Impellers for pumps, 816 (I)
——— friction of, 819 (I)
——— proportions of, 817 (I)
Imperial wire gauge, 87, 88 (I)
Impulse turbines, steam, 1496
——— water, 835 (I)
Inch, value of fractions of an, in millimetres, 4 (I)
Inches converted into centimetres, 3 (I)
——— reduced to decimals of a foot, 2 (I)
Inclines, mining, 892 (II)
Inclining experiment on ships, 275 (II)
Income tax, 1020 (II)
——— normal depreciation allowances, 1022 (II)
Inconel, 1222 (I)
Incrustation of water pipes, 734 (I)
India, irrigation systems, 715 (I)
——— rainfall in, 662 (I), 667 (I)
——— survey department, maps, 255 (I)
Indicated horse-power of steam engines, 75 (II)
Indicator (steam engine), 79 (II), 1095 (II)
——— card, 79 (II)
——— combustion, 1193 (II)
——— Crosby, 1095 (II)
——— diagram, to interpret, 81 (II)
——— ——— approximate steam consumption from the, 81 (II)
——— diagrams of compressors, 721 (II)
Inductance, 969 (II)
Induction motors, 1489 (I)
——— cage rotor, 1491 (II)
——— synchronous, 1489 (I)
——— regulators, 1503 (I)
——— ——— and booster transformers, 1503 (I)
Industrial trucks, electric, 385 (II), 1274 (II)
Inertia, moments of, 22 (I), 26 (I), 221 (I)
——— for simple sections, 221 (I)
Inflow theory, 406 (II)
Influence values, 218 (I)
Ingot casting metal protection method, 1193 (I)
——— iron, 1140 (I)
——— 'Armco,' 1141 (I)
Injection cocks, condenser, 181 (II)
Injectors, 54 (II)
——— locomotive, 549 (II)
Inland waterways. (*See* Canals)
Inlet valves, 461 (II)

Inspection, boiler, 53 (II)
Institution of Electrical Engineers regulations for electrical equipment, 1427 (I), 1459 (II)
—— —— electrical equipment of ships, 1453 (I)
—— —— wireless installation regulations, 1445 (I)
Instrument screw threads, 92 (I)
—— transformers, 1509 (I)
Instruments, electric measuring, 1506 (I), 1194 (II)
—— —— limits of errors in, 1507 (I)
—— graphic, 1511 (I)
—— surveying, 257 (I)
Insulating materials, aluminium foil, 1443
—— —— 'Dextramite,' 1103 (II)
—— —— electrical, 1525 (I)
—— —— glass silk, 1097 (II), 1103 (II)
—— —— heat, 377 (I), 68 (II), 1102 (II)
—— —— magnesia, 1102 (II)
—— oils, 1501 (I)
—— varnishes, 989 (II)
Insulation, cold storage, 733 (II)
—— ship refrigeration, 736 (II)
—— sound, 75 (I), 310 (I), 1097 (II)
—— testing instruments, 1194 (II)
—— vibration, 76 (I)
Insulator bushings, terminal, 1502 (II)
Insulators, porcelain, 1413 (I), 1415 (I)
—— tests for, 1414 (I)
Integrating photometer, 687 (II)
Intensifiers, hydraulic, 776 (I)
—— hydrostat, 776 (I)
Interlacing, television, 961 (II)
Internal combustion engines, 187 (II), 204 (II), 1266 (II)
(*see also* Gas engines ; Oil engines ; Automobile engines ; and Petrol engines)
bearings, 249 (II)
Brotherhood-Ricardo, 1266 (II)
cams, 219 (II)
compression and expansion of gases, 188 (II)
connecting rods, 209 (II), 212 (II), 215 (II)
cooling, (198 (II), 346 (II), 1273 (II)
crank cases, 226 (II)
—— shafts, 209 (II), 227 (II), 235 (II)
—— —— torsional vibration, 235 (II)
compression ratios, 189 (II)
constant volume cycle, 194 (II)
constructional details, 204 (II)
crossheads, 206 (II)
cycles, 197 (II)
cyclic irregularity, 230 (II)
cylinder arrangements, 199 (II)
—— heads, 215 (II)
dynamic troubles, 214 (II)
efficiency of, 1302 (I), 194 (II), 195 (II)
exhaust gases, 1313 (I)
filters for, 234 (II)
firing, 200 (II)
flexible couplings, 243 (II), 1269 (II)
flywheels, 230 (II)
foundations, 248 (II)
fuel injection, 202 (II)
fuels, 200 (II)
gas laws, 188 (II)
Gleniffer, 1266 (II)
governing, 250 (II)
high and low speed, 203 (II), 213 (II)
indicators, 1095 (II)
Lloyd's rules for, 251 (II)
maintenance, 249 (II)
mountings, 248 (II)
oil pipes, 207 (II)
opposed piston, 199 (II)
piston cooling, 206 (II)
pistons, 198 (II), 207 (II), 215 (II)
pumps, water and oil, 233 (II)
scavenging, 245 (II)
silencers for exhaust, 80 (I)
size limit, 210 (II)
specific volume, 191 (II)
standard specifications, 249 (II)
starting, 207 (II)
stresses in, 204 (II), 244 (II)
supercharging, 245 (II)
timing, 216 (II)
types, 197 (II), 213 (II)
units and equivalents, 187 (II)
valve springs, 223 (II)
valves, 199 (II), 216 (II)
vibration dampers, 243 (II)
volumetric efficiency, 195 (II)
Internal stresses, girders, 215 (I)
International Combustion Ltd., air preheaters, 1226 (II)
—— —— 'Lopulco' pulverised coal system, 1188 (II)
—— —— stokers, 1222 (II)
—— —— 'Usco' air preheater, 1226 (II)
—— Gas Detectors, Ltd., 'W.R.' CO_2 or Combustion Indicator, 1193 (II)
—— standard atmosphere, 411 (II), 447 (II)
—— system metrical screw threads, 94 (I)
Invar steel, 1183 (I)
Inventions, and engineers' rights, 1027 (II)
Inverted tooth chains, 961 (I)
Involute gear teeth for spur gearing, proportions of, 970 (I)
—— —— cutters, 973 (I)
—— —— gears, 970 (I)
—— —— setting out, 970 (I)
—— wheel teeth, 969 (I)
Iron, 1140 (I)
—— Admiralty tests for, 164 (I), 167 (I)
—— analyses of typical, 1142 (I)
—— angle and tee, weight of, 136 (I)
—— 'Armco' Ingot, 1141 (I)
—— balls and bars, weight of, 127 (I)
—— Barff process for protection of, 1194 (I)
—— boiler tubes, weight of, 41 (II)
—— calorising, 1195 (I)

Iron, cast (*see* Cast Iron)
—— castings, locomotives, 560 (II)
—— chains, annealing and normalising, 176 (I)
—————— at low temperatures, 176 (I)
—————— strength of, 169 (I)
—————— testing, 175 (I)
—— columns, cast, 229 (I)
—— corrosion of, 178 (I), 1192 (I)
—— ferrite, 1140 (I)
—— Fireman, automatic stoker, 1228 (II)
—— flat rolled, weight of, 129 (I)
—— foundry pig, 1142 (I)
—— galvanised, corrugated, 233 (I), 236 (I), 238 (I)
———————— painting, 239 (I)
—— galvanising, 1193 (I)
—— hematites, 1141 (I)
—— ingot, 1140 (I)
—— nails, weight of, 155 (I)
—— nickel alloys, 1217 (I)
—— pig, 1141 (I)
—— pillars, cast, 229 (I)
—— pipe bends, 61 (II)
—— pipes, cast (*see* Pipes)
—— rivets, strength of, 232 (I)
—— rivets, weight of, 156 (I)
—— rolled, flat, weight of, 129 (I)
—— roofing, corrugated, 236 (I), 238 (I)
—— ropes, corrosion of, 178 (I)
—— rust-proof coating for, 1192 (I)
—— screw stays for combustion chambers, 35 (II)
—— sheet, weight of, 129 (I)
—— sheets, galvanised, corrugated, 236 (I), 238 (I)
—— sherardising, 1195 (I)
—— specific gravity, 121 (I)
—— stress due to temperature, 176 (I)
—— tee, weight of, 136 (I)
—— tests for, 167 (I)
—— tubes, boiler, 41 (II)
———————— charcoal, lapwelded, 42 (II)
—— weight of, 128 (I)
—————— castings, 154 (I)
—— wire, annealed and bright, sizes, weights and breaking stresses of, 162 (I)
—————— weight of, 154 (I), 162 (I)
—— wrought (*see* Wrought iron)
Ironstone, 858 (II)
—————— British Mesozoic, 859 (II)
Irregular figures, area of, 249 (I)
Irrigation, 701 (I), 714 (I)
—————— canals, critical velocity, 708 (I)
———————— water losses from, 708 (I)
—————— costs, 712 (I)
—————— evaporation losses, 401 (I)
—————— in Australia, 716 (I)
—————— India, 715 (I)
—————— New Zealand, 714 (I)
—————— North America, 716 (I)
—————— South Africa, 714 (I)
—————— United States, 716 (I)
—————— pumped water system in Australia, 712 (I)
—————— recent literature, 717 (I)
—————— rice, 708 (I)
—————— water, duty of, 710 (I)
Isothermal expansion of gas, 188 (II)
———————— or compression, 1299 (I), 757 (II)

JACKETS, steam, 77 (II)
Jet air pumps, 180 (II)
——— condensers, 167 (II)
——— propulsion, 448 (II)
Jetties for river training, 580 (I)
Jig-back aerial ropeways, 839 (II)
Jigs, screening, 912 (II)
Joggling beams, 312 (I)
Johnson patent marine water-tube boiler 298 (II)
Jointing beams, 311 (I)
——— Klinger, 1258 (II)
——— material for pipes, weight of, 157 (I)
——— Turner's asbestos fibre, 1258 (II)
Jointless floors, 305 (I)
Joints, ascension pipe, 624 (II)
——— riveted, 188 (I), 229 (I), 232 (I)
——— soldered, strength of, 177 (I)
——— steam pipe, 63 (II)
——— universal, Hardy Spicer, 1272 (II)
Joists, strutting, 314 (I)
Journal bearings, 850 (I), 303 (II)
—————— clearance for, 870 (I)
—————— methods of mounting, 855 (I)

KANTHAL alloy, 1220 (I)
Keelavite Rotary Pumps & Motors, Ltd., hydraulic drive, 1110 (II)
Keene's cement, 357 (I)
Kelvin's rope brake, 41 (I)
Kerbs, concrete, 350 (I)
Kerosene, 1354 (I)
Keys, box spanner, 105 (I)
——— keybars, keyways, 117 (I)
Keystones, 468 (I)
K factors, table of, 970 (II)
Kiel canal, 599 (I)
Kilns, 1128 (I)
——— timber drying, 291 (I)
King George V Dock, 611 (I)
King-post roofs, 304 (I)
Kirke patent boilers, 1211 (II)
Kleen-E-Ze, brushes, 1302 (II)
Klinger, R., Ltd., jointing, 1258 (II)
——— valve, 1250 (II)
——— water gauge, 1241 (II)
Knots, scaffolding, 449 (I)
Koepe shock-damping buffer, 957 (I)
Krupp ball mills, 931 (II)
Kutter's formula, velocity of water in pipes and channels, 676 (I), 756 (I)

LABOUR produced by animal power, 35 (I)
———————— man power, 35, 36 (I)
——— upon embanknent, 38 (I)
Labyrinth packings, turbine, 130 (II)
Ladder dredgers, 632, 638 (I)
Ladybower reservoir, 694 (I)
Laggan dam, 691 (I)
Lagging, notes on, 68 (II)
Lake Mentz dam, 693 (I)
Lakes, temperature of, 54 (I)
Laminated springs, 900 (I)
———————— for locomotives, 553 (II)
———————— motor cars, 367 (II)
La Mont boiler, 15 (II), 1203 (II)
Lamps, electric, (704 II)
—————— architectural, 707 (II)
—————— artificial daylight, 705 (II)

Lamps, electric, discharge, 704 (II)
——— gas-filled, lumens rating, 704 (II)
——— mercury vapour, 705 (II)
——— sodium, 705 (II)
——— spacing, 696 (II)
——— tubular fluorescent, 705 (II)
—— miner's safety, 904 (II)
Lamson Engineering Co., Ltd., pneumatic despatch tubes, 1302 (II)
Lancashire boilers, 11 (II)
Land required for railways, 499 (II)
—— treatment of sewage, 786 (II)
Landing stages, 644 (I)
Lang's Lay wire ropes, 933 (I)
Lapping gear teeth, 987 (I)
Latent heat, 62 (I)
—— of fusion of metals, 1135 (I)
Lateral canals, 585 (I)
——— cost of, 585 (I)
Lathe cans, A. C. Wells & Co., Ltd., 1151 (II)
Lathes, 1020 (I)
—— back centres, 1021 (I)
—— hydraulic feeds, 1022 (I)
—— power required, 1021 (I), 1032 (I)
—— speed of, 1020 (I)
—— tools, cutting speeds, 1074 (I)
——— high speed, making, 1074 (I)
—— wood turning, 1040 (I)
Latitude and longitude, length of a degree in, 262 (I)
—— to find, 265 (I)
Lattice girder bridges, 517 (I)
'Lautal,' 1235 (I)
Lawn mowers, motor, Ransomes, 1277 (II)
Laws relating to engineers, 1027 (II)
Layout and planning of works, 905 (I)
Lead, 1242 (I)
—— alloys, 1242, 1250 (I)
——— melting points of, 51 (I)
—— antimony alloys, 1243 (I)
—— balls, weight of, 127 (I)
—— bars, weight of, 131 (I)
—— bearing steels, 1163 (I)
—— bronzes, 1223 (I)
—— covered cables, 1388 (I)
—— ore, 859 (II)
—— paints, 981 (II), 986 (II)
—— pipes, formulæ for weight of, 141 (I)
——— strength of, 168 (I)
——— weight of, 141, 738 (I)
—— rolled, weight of, 131 (I)
—— sheet, weight of, 132 (I)
—— sheets and bars, weight of, 131 (I)
—— welding, 1282 (I)
Leadless paints, 979 (II)
Leakage, water, tracing, 731 (I), 1107 (II)
—— Capacitors, 966 (II)
Leather belting, 909 (I)
—— waterproofing grease, 996 (II)
Leathers, lubrication, 808 (I)
—— packing, hydraulic, 806 (I)
—— pump, 806 (I)
—— ram or U, 806 (I)
Leclanché primary cells, 1476 (I)
Legal notes for engineers, 1027 (II)
—— status of engineers, 1027 (II)
Legislation, boiler, 54 (II)
Length measures, 1, 5 (I)
—— of a degree in latitude and longitude, 262 (I)
Lens, apertures of, photographic, 65 (I)
—— concave, 64 (I)
—— luminous buoy, 646 (I)
Lentz valve gear, 539 (II)
Levees, 582 (I)
Level, dumpy, 258 (I)
Levelling, 259 (I)
Levers, 20 (I)
Lewis formula for gears, 976 (I)
——— modifications of, 977 (I)
—— strength factors, 977 (I)
Liability for engineer's negligence, 1030 (II)
Lifting electro-magnets, 815 (II)
—— stones, 327 (I)
—— tackle, 169 (I)
——— notes on, 175 (I)
Lifts, 817 (II)
—— canal, 590 (I)
—— electric, specification for, 817 (II)
—— wire ropes for, 939 (I)
Light, 64 (I)
—— curves (polar), 687 (II)
—— decomposition of, in the spectrum, 64 (I)
—— meters, 687 (II)
—— railways (*see* Railways)
—— signalling on railways, 595 (II)
—— velocity of, 64 (I)
Lighthouses, 649 (I)
—— Bas Serani, 653 (I)
—— cost of, 649 (I)
—— distance visible from, 651 (I)
—— Eddystone, 649 (I)
—— glazing of, 653 (I)
—— lightning conductors for, 653 (I)
—— lights, 649 (I)
—— optical apparatus, 651 (I)
—— visibility from, 651 (I)
Lighting, 685 (II)
—— acetylene, 654 (II)
—— architectural, 707 (II)
—— automobile, 1528 (I)
—— brightness, 686 (II)
—— buildings, 691 (II)
—— distribution curves, 687 (II)
—— factories and workshops, 693 (II)
—— fittings, 689 (II)
—— photometry, 687 (II)
—— systems, designs of, 689 (II)
—— terms, 685 (II)
—— units, 685 (II)
—— values, 691 (II)
Lightmeters, 687 (II)
Lightning conductors for lighthouses, 653 (I)
——— Phœnix rules, 1517 (I)
——— rules for, 1516 (I)
Lights for lighthouses, 649 (I)
Lightships, 647 (I)
Lignite, 1329 (I)
—— tars, 1358 (I)
Lime, 339 (I)
Lime and soda water softening process, 49 (II)
—— burning, 339 (I)
—— concrete, 343 (I)
—— Lias, 339 (I)
—— mortar, 337 (I)
—— quick, 340 (I)
—— specification, 340 (I)
—— sprayers, 1297 (II)
Limes, hydraulic, 339 (I)
Limit switches, 814 (II)

Linatex rubber, 1301 (II)
Line and angle bisector, 252 (I)
'Line-o-Matic' lubrication system, 1150 (II)
Liners, Atlantic, 284 (II)
Lineshaft bearings, 854 (I)
——— drive, 906 (I)
Lining, brake, 'Ferodo,' 1153 (II)
—— for furnaces, 373 (I), 1236 (II)
—— 'Linatex' rubber, 1302 (II)
Link motion, 98 (II)
———— Allen's valve gear, 100 (II)
———— Gooch's valve gear, 99 (II)
———— locomotive valve gear, 98 (II)
———— Stephenson valve gear, 98 (II)
———— Walschaerts valve gear, 538 (II)
Linseed oil, 987 (II)
Lipowitz's alloy, 52 (I)
Liptak Furnace Arches, Ltd., 'Bigelow' suspended furnace walls, 1236 (II)
—— 'Gako' gas burner, 1282 (II)
Liquid case-hardening materials, 1188 (I)
—— fuels, 1351 (I), 202 (II), 561 (II)
———— furnace firing with, 1360 (I)
———— measures, 8 (I)
———— oxygen explosives, 951 (II)
Liquid fuels, vapour tension of, 192 (II)
Liquids, lighter than water, 125 (I)
——— quenching, 1073, 1179 (I)
——— specific gravities of, 120, 122 (I)
Liquor, gas, 625 (II), 635 (II), 644 (II)
Lithium, 1241 (I)
Liverpool, floating landing stage, 656 (I)
————, Gladstone Docks, 614 (I)
Ljungström air heater, 18 (II)
Lloyd-Davies Rain Run-off Formula, 666 (I)
Lloyd's rules for boiler safety valves, 57 (II)
————— steam pipes, 62 (II)
Load on foundations, 436 (I)
——— roofs, 300 (I)
Load safe, on bearings, 181 (I)
———— bolts and nuts, 100 (I)
———— columns, 227 (I), 437 (I)
———— floors, 437 (I)
———— materials, 436 (I)
———— ordinary foundations, 436 (I)
———— piles, 440 (I)
—————— reinforced concrete, 440 (I)
———— walls, 437 (I)
Loaders, coal, 1295 (II)
Loads carried by piles, 439 (I)
—— eccentric, 229 (I)
—— methods of transport, 36 (I)
—— of carts and wagons, 432 (I)
—— on floors, 318, 381 (I)
——— highway bridges, 482, 485, 488 (I)
——— pillars, 228, 325 (I)
——— railway bridges, 481 (I)
Lock nuts, 103 (I)
Lockheed hydraulic remote controls, 795 (I)
Locks, canal design, 595 (I)
—— dock entrance, 617 (I)
—— gates, hydraulic operation of, 793 (I)
—— on canals and rivers, 590 (I)
—— twin, on German canals, 595 (I)
Locomotives, 515 (II), 1278 (II)
———— adhesion of, 531 (II)
———— American, 520 (II)
———— articulated, 523 (II)
———— axles, 550 (II)
———— bearing surfaces of, 540 (II)
———— boilers, 544 (II)
Locomotives, boilers, brass tubes for, 556 (II)
—————— copper plates for, 555 (II)
———————— tubes, 555 (II)
—————— feed water heating, 548 (II)
—————— fire boxes of, 546 (II)
—————— grate areas and heating surfaces of, 545 (II)
—————— headers, 1278 (II)
—————— heating surface calculation, 546 (II)
—————— rivets for, 557 (II)
—————— rods for, 557 (II)
—————— specification of material for, 560 (II)
—————— steel castings for, 544 (II)
———————— plates for, 557 (II)
———————— tubes for, 42 (II), 545 (II), 555 (II)
—————— superheaters, 547 (II), 1278 (II)
—————— tubes, 42 (II), 545 (II), 555 (II)
———————— life of, 545 (II)
———— brakes, 563 (II)
———— brass tubes for, 556 (II)
———— British, general types, 523 (II)
—————— recent additions to, 515 (II)
—————— tank, 518 (II), 520 (II)
—————— tender, 516 (II), 518 (II)
———— castings for, 554 (II)
———— coal consumption, 562 (II)
———— compound, 536 (II)
—————— cylinders of, 536 (II)
———— compressed air, 894 (II)
———— connecting rods, 538 (II)
———— consumption of fuel in, 562 (II)
—————— water in, 562 (II)
———— contract specification, 561 (II)
———— copper pipes, 557 (II)
—————— plates for, 555 (II)
—————— rods for, 555 (II)
—————— tubes for, 555 (II)
———— counterbalancing driving wheels, 542 (II)
———— coupling rods, 538 (II)
———— cranes, 805 (II)
———— crank axles of, 540 (II), 550 (II)
———— cylinders of, 536 (II)
—————— simple and compound of equivalent power, 536 (II)
———— design of, 524 (II)
———— developments abroad, 522 (II)
———— driving-wheels, counterbalancing, 542 (II)
———— electric, 589 (II), 894 (II), 384 (II)
———— examination of, in running sheds, 564 (II)
———— feed-water heating, 548 (II)
———— fire-boxes, 546 (II)
—————— copper plates for, 555 (II)
—————— life of, 546 (II)
—————— steel plates for, 557 (II)
———— flexible wheel base, 542 (II)

Locomotives, forgings for, 554 (II)
—— fuel consumption, 562 (II)
—— grate area and heating surface, 545 (II)
—— Great Western Railway, 518 (II), 520 (II)
—— heating surface, 545 (II)
—— horse-power of, 531 (II)
—— injectors, 549 (II)
—— iron castings for, 560 (II)
—— link motion, 98 (II)
—— liquid fuel for, 561 (II)
—— London Midland & Scottish Railway, 518 (II), 520 (II)
—— —— & North Eastern Railway, 518 (II), 520 (II)
—— lubricating oil consumption in, 562 (II)
—— mechanical stokers, 562 (II)
—— mine, 894 (II)
—— motion work of, 538 (II)
—— oil-electric, 385 (II), 565 (II)
—— oil-engined, 566 (II)
—— parts of, 536 (II)
—— piston rod packing, United Kingdom, 1281 (II)
—— —— speed, 531 (II)
—— —— valves, 537 (II)
—— power of, 531 (II)
—— —— relation to weight, 533 (II)
—— pulverised coal for, 1345 (I)
—— relation of power to weight 533 (II)
—— resistance of, 526 (II), 530 (II)
—— road, 554 (I)
—— smoke boxes, 546 (II)
—— Southern Railway, 518 (II), 520 (II)
—— speed of, 530 (II)
—— springs of, 551 (II)
—— steel castings for, 554 (II)
—— —— forgings for, 554 (II)
—— —— plates for, 557 (II)
—— —— tubes for, 42 (II), 545 (II) 555 (II)
—— storage battery, 384 (II)
—— straight, axles of, 550 (II)
—— superheaters, 547 (II), 1278 (II)
—— —— 'McLeSco,' 1278 (II)
—— tank, 518 (II), 520 (II)
—— tender, tools for, 564 (II)
—— tenders, 563 (II)
—— tractive force of, 531 (II), 536 (II)
—— tubes, 42 (II), 545 (II)
—— types, 522 (II)
—— tyres for wheels of, 534 (II), 550 (II)
—— valve gear, 538 (II)
—— —— Caprotti, 540 (II)
—— —— Lentz poppet, 539 (II)
—— —— Walschaerts, 538 (II)
—— valves of, 537 (II)
—— water consumption, 562 (II)
—— —— troughs, 564 (II)
—— weight, 533 (II)
—— —— distribution on bridges, 481 (I)
—— weight restriction in, 533 (II)
—— wheel base, 542 (II)
Locomotives, wheel base, flexible, 542 (II)
—— wheels, 541 (II)
—— —— counterbalancing, 542 (II)
Lockheed brake system, 379 (II)
Loeffler boiler, 15 (II)
Log-line, 1 (I)
Logarithmic sines, tangents, etc., 1055 (II)
Logarithms, 1074 (II)
—— hyperbolic or natural, 1090 (II)
Logs, storing, 293 (I)
Logue system of involute gear teeth, 972 (I)
Lohmanising, 1193 (I)
London Building Acts, 318 (I), 381 (I), 466 (I)
—— County Council regulations with reference to reinforced concrete, 381 (I)
—— —— regulations with reference to structural steelwork, 322 (I)
—— Midland & Scottish Railway carriages, 572 (II)
—— —— locomotives, 518 (II), 520 (II)
—— —— wagons, 574 (II)
—— & Midland Steel Scaffolding Co., Ltd. steel tubular scaffolding, 1104 (II)
—— & North Eastern Railway carriages, 572 (II)
—— —— locomotives, 518 (II), 520 (II)
—— —— wagons, 574 (II)
—— refuse, calorific value of, 1349 (I)
Longitude and latitude, length of a degree in, 262 (I)
—— to find, 268 (I)
Longitudinal force on bridges, 500 (I)
—— metacentre of ships, 276 (II)
Longwall coalcutters, 1294 (II)
Lopulco boilers and pulverised fuel system, 1188 (II)
Lorries, electric, 384 (II)
—— petrol-electric, 385 (II)
Loss of water on canals, 595 (I)
Loudness of sound, 70 (I)
Loudspeakers, matching, 972 (II)
Low pressure steam heating systems, 670 (II)
Low temperature distillation of coal, 1325 (I)
Lubricants, petroleum, 871 (I)
Lubricating oil filter, A. C. Wells & Co., Ltd., 1151 (II)
—— filtration of, 878 (I), 1151 (II)
—— flash point, 873 (I)
—— recovery of, 878 (I), 1151 (II)
—— oils, 871 (I), 995 (II)
—— —— heavy oil engine, 879 (I)
—— —— laboratory tests, 873 (I)
—— —— properties of, 872 (I)
—— —— viscosity of, 874 (I), 883 (I)
—— —— turbine, 878 (I)
Lubrication, 871 (I), 1148 (II)
—— automobile, 380 (II)
—— ball bearing, 862 (I)
—— bearing, 869 (I)
—— gearing, 1008 (I)
—— graphite, 880, (I) 1148 (II)

Lubrication, grinding, 1107 (I)
—— leather packing, 808 (I)
—— locomotives, 562 (II)
—— mechanical, with oil, 1150 (II)
—— motor car, 380 (II)
—— oil engine, 878 (I)
—— oils, standard, physical properties of, 872 (I)
—— pneumatic tools, 1081 (I)
—— pump leather, 808 (I)
—— roller bearing, 862 (I)
—— shafting, 878 (I)
—— steam turbines, 878 (I), 158 (II)
—— swing bridges, 530 (I)
—— Tecalemit system, 1150 (II)
—— wire rope, 955 (I)
Lumen unit, 685 (II)
—— value of gas-filled lamps, 704 (II)
Luminous buoys, 645 (I)
—— discharge tubes, 1442 (I)
—— paint, 995 (II)

MAAG-SULZER gear tooth grinding, 986 (I)
Machine-moulded teeth for gears, 972 (I)
—— shop drives, 907 (I)
—— —— layout, 906 (I)
Machine tools, 1019 (I), 1160 (II)
—— broaching machines, 1050 (I)
—— chucks, 1063 (I)
—— —— magnetic, 1063 (I)
—— drilling machines, high-speed, 1028 (I)
—— drills, 1055 (I)
—— forming machines, wire and strip, 1160 (II)
—— grinding machines, 1029 (I), 1107 (I)
—— horse-power required for, 1035 (I)
—— hydraulic feeds, 1022 (I)
—— lathes, 1020 (I)
—— lock nuts, 103 (I)
—— milling machines, 1025 (I)
—— planing machines, 1023 (I)
—— pneumatic, 1030, 1078 (I)
—— portable, 1030, 1078 (I)
—— sawing machines, Rapidor, 1161 (II)
—— screwing machines, 1162 (II)
—— shaping machines, 1027 (I)
—— tool steels and tools, 1063 (I)
—— tools for, 1044 (I), 1060 (I)
—— tube cutting-off, 1165 (II)
—— wood working, 1037 (I)
Machinery, depreciation of, 1017 (II)
—— purchase of, 1036 (II)
—— repairing worn parts, 1082 (I)
—— vibration, 77 (I)
—— wood working, 1037 (I)
Machines, hydraulic, 767 (I)
Machining ebonite and fibre, 1077 (I)
'Maco' adjustable template, 513 (II)
Madan, C. S., & Co. Ltd., Airhydropump, 1123 (II)
Madden reservoir, U.S.A., 693 (I)
Magnalium alloy, 1232 (I)
Magnesia, insulating, 1102 (II)
Magnesium, 1208 (I)
—— alloys, 1232 (I), 1239 (I), 416 (II)
—— properties, 1208 (I)
Magnesium, welding, 1282 (I)
Magnet steel, 1182 (I)
Magnetic chucks, 1063 (I)
—— dip, 260 (I)
—— force, horizontal and vertical, 260 (I)
—— separation, mining, 916 (II)
Magnets, electro-lifting, 815 (II)
Magnolia metal, 1243 (I)
'Magnuminium' alloy, 1240 (I)
Main line railway electrification, Ministry of Transport's Report on, 578 (II)
—— tail rope haulage, 893 (II)
Mains, air, 1078 (I)
—— —— loss in, due to friction, 757 (II)
—— gas, 624 (II)
—— hydraulic, 624 (II)
—— transformers, 967 (II)
Malay tin basis, 938 (II)
Malleable cast iron, 1149 (I)
Malm bricks, 364 (I)
Man, power of, 35 (I), 36 (I)
Manchester ship canal, 599 (I)
Manganese, 1209 (I)
—— alloys, 1226, 1232 (I)
—— Bronze & Brass Co., Ltd., self-lubricating 'Oilite' bearings, 1141 (II)
—— bronze, 1226 (I)
—— influence of, on cast iron, 1147 (I)
—— molybdenum alloy steels, 1185 (I)
—— steel, 1185 (I)
Magnanin, 1226 (I)
Manifolds, 1725 (I), 349 (II)
Manila ropes, 176 (I)
Maps of India Survey Dept., 255 (I)
—— Ordnance, scales of, 254 (I)
Marble, artificial, 358 (I)
Marine boilers, Babcock & Wilcox, 298 (II)
—— —— cylindrical, 36 (II), 293 (II)
—— —— draught of, 296 (II)
—— —— furnaces and firebars, 38 (II)
—— —— Johnson, 298 (II)
—— —— life of, 298 (II)
—— —— materials of construction, 34 (II)
—— —— standard conditions for design and construction of, 34 (II)
—— —— steel for, 163 (I)
—— —— water tube, 297 (II)
—— —— weight of, 278 (II)
—— —— Yarrow, 298 (II)
—— condensers, 167 (II)
—— engineering, 287 (II)
—— engines, 287 (II), 305 (II)
—— —— cylinder diameters, 289 (II)
—— —— diagram factor, 288 (II)
—— —— oil fuel for, 312 (II)
—— —— piston speed of, 288 (II)
—— —— recipro-turbine machinery, 289 (II)
—— —— reciprocating, 287 (II)
—— oil engines, 305 (II)
—— —— auxiliaries, 300 (II)
—— —— Diesel-electric drive, 316 (II)
—— —— fuel, 312 (II)
—— —— geared transmission, 316 (II)
—— —— high-speed, 317 (II)
—— —— in service, 306 (II)
—— —— lubrication, 878 (I)

Marine oil engines, mean indicated pressure, 307 (II)
—— Michell journal bearings 303 (II)
—— output and dimensions, 306 (II), 315 (II)
—— power, 306 (II)
—— reversing systems, 309 (II)
—— structure, 307 (II)
—— supercharging, 312 (II)
—— types of, 306 (II)
—— weights and sizes, 315 (II)
—— fuel, 312 (II)
—— propellers, 283 (II)
—— propulsion, 283 (II), 1528 (I)
—— reduction gears, 290 (II)
—— steam turbines, 290 (II)
—— turbine gears, reduction, 290 (II)
—— water-tube boilers, 297 (II)
Marl roofing tiles, 367 (I)
Marsh gas, 1367 (I)
Martensite, 1145 (I)
Martin's Cement, 357 (I)
Mash welding, 1292 (I)
Masonry, 327 (I)
—— arches, 467 (I)
—— ashlar, 330 (I)
—— block-in-course, 330 (I)
—— bonds, 331 (I)
—— chimney shafts, 464 (I)
—— dams, 686, 696 (I)
—— descriptions of, 327 (I)
—— dowelling, 331 (I)
—— measurements and valuation of, 331 (I)
—— permissible pressures on, 332 (I)
—— proportions of stones and bond 331 (I)
—— rubble, 329 (I)
—— strength of, 180 (I)
—— valuation of, 331 (I)
Mass definition, 22 (I)
Massey, B. & S., pneumatic hammers, 1210 (I)
—— power hammers, 1120 (I)
—— steam hammers, 1118 (I)
Mastic cements, 353 (I)
Materials, coefficients of friction, 21 (I)
—— hardness of, 1088 (I)
—— safe load on, 438 (I)
—— shrinkage of, 425 (I)
—— strength of, 159 (I)
—— weight of, 119 (I)
Mathematics, 1039 (II)
Mavor and Coulson Ltd., coal face machinery 1294 (II)
McKiernan Terry pile hammer, 442 (I)
McNeil Charles, Ltd., 'W' central heating boiler, 1217 (II)
Mean effective pressure, 74 (II), 82 (II), 84 (II)
—— refractions, table of, 270 (I)
Measuring instruments, electric, 1506 (I), 1194 (II)
Measurement and valuation of masonry 331 (I)
—— distance, 251 (I)
—— of light, 687 (II)
—— subtense, by tacheometer, 258 (I)
—— units of, 1 (I)
Measures, and weights, 1 (I)
—— foreign, 9, 10 (I)
—— foreign and English, 10 (I)
—— geographical and nautical, 1, 249 (I)
Mechanical draught, 1337 (I)
Mechanical efficiency of aero engines, 445 (II)
—— steam engines, 75 (II)
—— elevator, mining, 935 (II)
—— lubrication with oil, 1150 (II)
—— stokers, 1336 (I), 1220 (II)
—— Bennis, 1220 (II)
—— Crosthwaite, 1221 (II)
—— 'Iron Fireman,' 1228 (II)
—— locomotive, 562 (II)
—— Riley spreader, 1224 (II)
—— testing appliances, 191 (I)
Meg & Megger testing instruments, 1194 (II)
MeLeSco superheaters, 1229 (II), 1278 (II)
'Melloid,' 1174 (II)
Melting-point of alloys, 51, 1247 (I)
—— bismuth-lead-tin alloys, 1247 (I)
—— bricks, 371 (I)
—— bronze, 1225 (I)
—— chemical elements, 51 (I)
—— refractory materials, 371 (I), 1134 (I)
Melting-points, 50 (I)
—— on the thermo-dynamic scale, 1139 (I)
Men, power of, 35 (I)
—— work done by, 35 (I)
Mensuration, 29 (I)
Mercury-arc rectifiers, 1485 (I)
—— dilation or expansion per degree F., 62 (I)
—— freezing temperatures, 50 (I)
—— U-Tube steam meter, 202 (I)
—— vapour lamps, 704 (II)
—— rectifiers, 1468 (I), 963 (II)
—— grid control, 1486 (I)
Meridians, convergence of, 272 (I)
—— radius of curvature, 271 (I)
Mersey tunnel, 572 (I)
Metacentre, longitudinal, of ships, 276 (II)
—— transverse, of ships, 274 (II)
Metadyne control, 592 (II)
Metal, 1140 (I) (*see also* Alloys)
—— adulteration of, to discover, 121 (I)
—— airscrews, 409 (II)
—— atomic weights, 120 (I)
—— Babbitt, 1243 (I)
—— balls, weight of, 127 (I)
—— bearing, 1223 (I) 1242 (I), 462 (II)
—— bell, 1223 (I)
—— Bull's, 1174 (II)
—— coatings, protective, 1192 (I)
—— coefficients of friction, 21 (I)
—— corrosion of, 178 (I), 1192 (I)
—— cutting speeds, 1020 (I)
—— deposition on worn parts, 1082 (I)
—— Delta, 1215 (I), 1244 (I)
—— elasticity of, 159 (I)
—— extruded, 1215 (I)
—— fatigue of, 181 (I)
—— chemical action and, 188 (I)
—— limit, 183 (I)
—— temperature effect, 185 (I)
—— testing machines, 182 (I)
—— fusible, 1246 (I)
—— galvanising, 1192 (I)
—— gun, 1223 (I)
—— hardness of, 1088 (I)
—— value of, Brinell, 1089 (I)
—— Rockwell, 1093 (I)

Metal, hardness value of, scleroscope, 1089 (I)
—— heat energy required to melt, 1135 (I)
—————— treatment of, 1256 (I)
—— hexagon rod, 131 (I)
—— hysteresis in fatigue of, 188 (I)
—— mechanical treatment of, 1256 (I)
—— 'Melloid,' 1174 (II)
—— Monel, 1221 (I)
—— octagon rod, 131 (I)
—— phosphor bronze, 1224 (I)
—— physical properties of, 1253 (I), 1255 (I)
—— sawing, 1036 (I)
—— sheet, specification of, 132 (I)
—— sheets and bars, weight of, 127 (I)
—— specific gravities of, 121 (I)
—— Spence's, 1248 (I)
—— spraying, 1082 (I), 1194 (I)
—— strength of, 159 (I)
———————— at different temperatures, 160 (I)
———————— high temperatures, 1186 (I)
———————— low temperatures, 1242 (I)
—— stress due to temperature, 177 (I)
—— thermal conductivity, 63 (I), 1204 (I)
—— transverse strength of, 162 (I)
—— Tungum, 1217 (I)
—— type, 1248 (I)
—— wear of, on steel and cast iron, 1191 (I)
—— weight of square foot of, 129 (I)
—— white, Admiralty, 1243 (I)
—— working of, 1256 (I)
—— yield stress, 185 (I)
Metallic arc welding, 1287 (I)
———— packing, United States, 1260 (II)
———— paints, 993 (II)
———— piston-rod packing, United Kingdom Self - adjusting Anti - friction Metallic Packing Synd. Ltd., 1281 (II)
———— resistance materials, 1379 (I)
———— salts, 1136 (I)
Metalliferous minerals, 858 (II)
Metallurgy, 1225 (I), 1170 (II)
Metering water, air, gas, steam, and coal, 191 (I)
Meters and metering, 191 (I)
——— air, 201 (I)
——— coal, 204 (I)
——— conveyor volume, 204 (I)
——————— weight, 205 (I)
——— current, 578, 680, 756 (I)
——— electricity, 1507 (I), 1511 (I)
——————— specification for, 1512 (I)
——— gas, 198 (I), 629 (II), 648 (II)
—————— drum type, 198 (I)
—————— heat injection type, 199 (I)
—————— hot wire amemometer, 201 (I)
—————— inferential, 629 (II)
—————— orifice, 200 (I)
—————— positive displacement, 629 (II)
—————— rotary type, 200 (I)
—————— semi-positive, 629 (II)
—————— shunt type, 200 (I)
—————— Venturi type, 199 (I)
——— Hopkinson's torsion, 38 (I)
——— light, 687 (II)
——— steam 202 (I)
——————— mercury U-tube, 202 (I)
——————— orifice, 202 (I)
——————— rotary, 204 (I)
——— torsion, 37 (I)
Meters viscous flow air, 201 (I)
——— water, 193 (I), 756 (I), 1107 (II)
————————— cylindrical piston, 194 (I)
————————— disc type, 195 (I)
————————— flap type, 193 (I)
————————— nutating piston, 193 (I)
————————— piston, 193 (I)
————————— swash plate, 194 (I)
————————— venturi, 195 (I), 752 (I)
Methane (marsh gas), 1367 (I)
Metric candle, 685 (II)
——— carat, 7 (I)
——— conversion tables, 3–9 (I)
——— equivalents, 187 (II)
——— gold and silver tables, 932 (II)
——— pressures, 12 (I)
——— system, 3 (I)
Metrical screw threads, 94 (I)
———— surfaces, 6 (I)
———— weights, 7 (I)
Metropolitan Buildings Act (*see* London Building Acts)
Michell bearings, 303 (II)
Microphony, values, 963 (II)
Mild steel, gears, 981 (I)
—————— normalising, 167 (I)
—————— stanchions, 226 (I)
—————— treatment of, 166 (I)
Mill gearing, 972 (I)
Millimetres converted into inches, 4 (I)
Milling cutters, 1044 (I), 1069 (I)
——— gold, 928 (II)
————— cost of, 929 (II), 931 (II)
——— machines, 1035 (I)
—————————— power for, 1033 (I)
—————————— speeds and feeds, 1045 (I)
——— negative rake, 1047 (I)
Mills and millhouse plant for pulverised coal, 1339 (I), 1347 (I), 1186 (II)
—————————————— cost of, 1348 (I)
—— ball, 931 (II)
—— Chilian, 930 (II)
—— crushing, 933 (II)
—— gold, 928 (II)
———— cost of, 929 (II)
—— Raymond bowl, 1187 (II)
—— rolls, mining, 930 (II)
—— tube, mining, 930 (II)
Mine, legal definition of a, 847 (II)
Mineral deposits, classification of, 854 (II)
———— hardness, MOHS scale, 181 (I)
———— legal definition of, 847 (II)
———— mixtures, calculation of, 860 (II)
———— valuation, 941 (II)
Minerals, blasting, 862 (II)
———— dredging for alluvial, 936 (II)
———— geophysics for location of, 862 (II)
———— location of, 826 (II)
———— metalliferous, 858 (II)
———— non-metallic, 860 (II)
———— ownership of, 847 (II)
———— specific gravities and weights of, 860 (II)
Mines and Mining, 847 (II), 1294 (II)
——————— air regulator, 904 (II)
——————— alluvial or placer, 932 (II)
——————— blasting, 862 (II)
——————— bore-hole location for valuation of alluvial, 862 (II)
——————— boring, 861 (II)

Mines and Mining, branches of, 854 (II)
—— centrifugal pumps, 700 (II), 935 (II)
—— coal, 854 (II)
———— cleaning, 713 (II)
———— cutters, 876 (II), 1294 (II)
———— dust in, 906 (II)
———— explosives, 950 (II)
—— coffering, 867 (II)
—— compressed air locomotives, 894 (I)
—— conveyors, 895 (II)
—— core boring, 861 (II)
—— cost of, 865 (II), 868 (II), 934 (II)
—— deep shafts of, 872 (II)
—— detonators, 863 (II)
—— drainage, 899 (II)
—— dredging, 936 (II)
—— dressing of ores, 908 (II)
—— drill bits, sharpening, 887 (II)
—— drilling, 886 (II)
———— mechanical, 887 (II)
—— drills, air receivers, 889 (II)
———— required for, 888 (II)
—— electric locomotives, 894 (II)
———— signalling in, 908 (II)
—— elevators, hydraulic, 934 (II)
—— explosives, 863 (II), 950 (II)
—— fan ventilation, 903 (II)
—— fire extinction in, 908 (II)
———— damp, 901 (II)
—————— detectors in, 901 (II)
—— flotation processes and machines, 917 (II)
—— flow sheets, 918 (II)
—— François cementation process, 870 (II)
—— gas caps, 905 (II)
———— detection in, 907 (II)
—— gases in, 901 (II)
—— gold, 928 (II)
———— and silver tables, 932 (II)
———— cyanide process, 928 (II)
———— dredges, 936 (II)
—— haulage, 889 (II)
———— cost of, 890 (II)
———— endless rope and chain, 893 (II)
———— locomotives, 894 (II)
———— main and tail rope, 893 (II)
———— rails, 891 (II)
———— ropes for, 942 (I)
———— tubs, 891 (II)
—— headgears, 898 (II)
—— hydraulic, 933 (II)
—— inclines, 892 (II)
—— ironstone, 858 (II)
Mines and Mining, legal, 847 (II)
—— legislative enactments relating to, 852 (II)
—— locomotives, 894 (II)
—— magnetic separation, 916 (II)
—— masses and veins, 883 (II)
—— mechanical drilling, 887 (II)
—— minerals, classification of, 854 (II)
———— metalliferous, 858 (II)
———— ownership of, 847 (II)
—— monitors or giants, 933 (II)
—— ore dressing and coal washing, 908 (II)
—— ores, 858 (II)
—— pipe lining, 935 (II)
—— placer or alluvial, 932 (II)
—— pulleys, 898 (II)
—— pumping, 899 (II)
—— raising and winzing, 869 (II)
—— rents and royalties, 848 (II)
—— rescue apparatus, 906 (II)
—— rock drills for, 886 (II)
—— rolls, 930 (II)
—— ropes, 942 (I)
———— damping dynamic stresses in, 957 (I)
—— safety lamps, 904 (II)
—— shaft pillars, 872 (II)
———— sinking, 864 (II), 869 (II)
—————— by freezing, 869 (II)
—— shafts, 864 (II)
———— cost of sinking, 865 (II), 868 (II)
———— deep, 872 (II)
———— lining for, 866 (II)
—— rectangular, 864 (II)
—— sizing, 910 (II)
—— sluices, 933 (II)
—— spontaneous combustion, 907 (II)
—— stamps, 910 (II), 929 (II)
—— stemming, 863 (II)
—— subsidences, 875 (II)
—— tin dredges, 937 (II)
—— tubbing, 867 (II)
—— tube mills, 931 (II)
—— tubs, 891 (II)
—— valuation, 941 (II)
—— veins and masses, 883 (II)
—— ventilation, 901 (II)
—— winding, 895 (II)
———— plants, electric, 897 (II)
—— winzing and raising, 869 (II)
—— working costs of coal, 879 (II)
Miniature valves, 962 (II)
Ministry of Transport, permanent way, regulations, 493 (II)
—— railways, 493 (II)
—— regulations, overhead electric lines, 1420 (I)
—— report on main line railway electrification, 587 (II)

Ministry of Transport, standard load for highway bridges, 482 (I), 485 (I), 488 (I)
Mirrors, convex, 64 (I)
Mississippi river bank, protection of the, 580 (I)
—— dredging plant on, 640 (I)
Missouri river bank protection, 580 (I)
—— groynes on, 581 (I)
Mittel-land canal, 602 (I)
Mixers, concrete, 346 (I), 569 (I)
Mixtures, freezing, 53 (I)
—— non-freezing, 53 (I)
Modulation, frequency, 958 (II)
—— phase, 959 (II)
Modulus of elasticity of metals, 159, 1242 (I)
Mohr's theorem, 514 (I)
MOHS scale of hardness 181 (I)
Moisture changes, effects of, on building materials 359 (I)
—— content of timber, 289 (I)
Mollier diagram, 1304 (I)
Molybdenum, 1209 (I)
—— effect on cast iron, 1148 (I)
—— —— steel, 1184 (I)
—— manganese alloy steels, 1185 (I)
—— steels, 1184 (I)
Moments, bending, 215 (I), 225 (I)
—— of inertia, 22 (I), 26 (I)
—— —— for simple sections, 221 (I)
Momentum of trains, 500 (I)
Monel metal, 1220 (I)
—— mechanical properties, 1221 (I)
—— silicon, 1221 (I)
Monitors or giants, mining, 933 (II)
Monocable ropeways, 831 (II)
Monocoque construction, aircraft, 425 (II)
Monoxide in mines, 907 (II)
Monsoon rainfalls in Nagpur, 701 (I)
Morse Chain Co., Ltd., 1155 (II)
—— taper measures for twist drills, 1056 (I)
—— wire drill gauge, 1057 (I)
Mortars, 337 (I)
—— adhesive strength of, 181 (I)
—— cement, 338 (I)
—— choice of, 342 (I)
—— coloured, 342 (I)
—— crushing strength of, 181 (I)
—— lime, 337 (I)
—— mixing, 342 (I)
—— proportions for, 341 (I)
—— raw materials for, 339 (I)
—— refractory, 373 (I)
—— specifications of, 343 (I)
—— tensile strengths of, 181 (I)
—— trass, 337 (I)
—— waterproof, 341 (I)
Mortise and tenon, 313 (I)
Moscow-Volga Canal, 602 (I)
Motor-cars (*see* Automobiles)
—— steam railway, 565 (II)
—— coach trains, control equipment for, 591 (II)
—— lawn mowers, Ransomes', 1277 (II)
—— ships, 305 (II)
—— —— machinery installations, 306 (II)
—— vehicles, road regulations, 554 (I)
—— —— sizes, 554 (I)
Motors, electric, alternating current, 1489 (I)
—— cage rotor, 1491 (I)
—— carbon brushes, 1492 (I)
Moters, electric, commutator, 1492 (I)
—— control of A.C. traction, 593 (II)
—— direct current, 1487 (I)
—— —— speed regulation of, 1489
tion of, 1489 (I)
—— —— starters, 1487 (I)
—— —— traction, 593 (II)
—— efficiency of, 1481 (I)
—— fractional h.p., 1482 (I)
—— induction, 1489 (I), 1491 (I)
—— Metadyne control, 592 (II)
—— shaft driving by, 907 (I)
—— starters for, 1497 (I)
—— —— auto-transformers, 1497 (I)
—— —— rheostatic, 1497 (I)
—— —— switching, 1497 (I)
—— synchronous, 1489 (I), 1491 (I)
Moulded dimensions of ships, 271 (II)
Moulding machines, wood, 1041 (I)
—— presses for plastics, 792 (I)
—— sand, 1261 (I)
Mound breakwaters, 608 (I)
Movement of air by fans, 681 (II)
Moving coil regulator, 1504 (I)
Mowers, motor lawn, Ransomes', 1377 (II)
Muffle furnaces, 1125 (I), 1128 (I)
' Multiflow ' feed heaters, 1248 (II)
Multiple arch dams, 683 (I), 698 (I)
—— electrode welding, 1291 (I)
Mumetal, 1218 (I)
Mumford feedwater regulator, 1248 (II)
Murex Welding Processes Ltd., welding plant, 1183 (II)

NAILS, composition, weight of, 156 (I)
—— copper, weight of, 155 (I)
—— holding power in wood, 179 (I)
—— slating, 298 (I)
—— wrought-iron, weight of, 155 (I)
Naphthalene, 638 (II)
Nappe, effects of shape of, 749 (I)
Narrow-gauge, or light, railways, 499 (II)
National Grid Transmission Lines, 1410 (I)
—— porcelain insulators for, 1415 (I),
—— Physical Laboratory wind channel, 390 (II)
Natural cements, 354 (I)
—— gas, 1368 (I)
Nautical measures, 1 (I)
Naval architecture, 271 (II)
—— brass, 1216 (I)
Navigable waterways, 577 (I)
Navigation of rivers, 577 (I)
Needle bearings, 853 (I)
—— propeller shafts, 1272 (II)
Negative rake milling, 1047 (I)
Negligence, engineer's liability for, 1030 (II)
Neutral bricks, 372 (I)
New York State barge canal, 590 (I)
New Zealand, irrigation in, 714 (I)

Newman, Hender & Co., Ltd., ' Newman-Milliken ' valves, 1251 (II)
—————— steam traps, 1253 (II)
Nickel, 1206 (I), 1217 (I), 1226 (I)
—— ' Adnic ' white alloy, 1220 (I)
—— alloys, 1217 (I)
—— barium alloys, 1220 (I)
—— brass, 1218 (I)
—— chrome steel, 1164 (I)
—— chromium alloys, 1219 (I)
—— ' clothing ' 1193 (I)
—— copper alloys, 1226 (I)
—— electric resistance alloys, 1220 (I)
—— impurities in, 1206 (I)
—— influence on cast iron, 1147 (I)
—— iron accumulators, 1480 (I)
—— silver, 1218 (I)
—— steel, 1183 (I)
———— expansion of, 1184 (I)
—— welding, 1282 (I)
Nitralloy steels, 1189 (I)
Nitriding or nitrogen case-hardening, 1189 (I)
Nitro-glycerine, 950 (II)
Nitrogen case-hardening, 983 (I), 1189 (I)
Noise, 69 (I), 78 (I)
—— abatement, 74 (1), 79 (I), 1097 (II)
—— aeroplane, 80 (I)
—— electrical machinery, 81 (I)
—— gearing, 981 (I)
—— traffic, 79 (I)
—— ventilating fans, 81 (I)
Non-ferrous alloys, 1212 (I), 1249 (I), 1170 (II)
———— materials for turbine blading 152 (II)
———— properties at elevated temperatures, 1254 (I)
Non-magnetic alloys, 1248 (I)
Non-metallic inclusions in steel, 1192 (I)
———— minerals, 860 (II)
Non-resonant pinions, 981 (I)
Non-shrinking die steels, 1185 (I)
Normalising mild steel, 167 (I), 1165 (I)
Notch, method of calculating quantity of water flowing in a stream, 672 (I), 674 (I)
Notches and weirs, 746 (I)
' Novox ' resuscitating apparatus, 1300 (II)
Nozzles, 1115 (I), 713 (II), 765 (II)
—— air, 1115 (I)
—— flow of steam through, 6 (II), 107 (II)
—— steam turbine, 107 (II)
—— test purpose, 207 (I)
—— water, 752 (I)
———— turbine, 839 (I)
Nuts and bolts, 98 (I)
———— American, 99 (I)
———— British Standard, 99 (I)
———— French, 101 (I)
———— Whitworth, 98 (I), 100 (I), 102 (I)
———— bolt-heads, British standard
——— bolt-heads, British standard hexagonal, 102 (I)
——— bolt-heads, weight of, 154 (I)
—— capped, 104 (I)
—— castle, British standard, 102 (I)
—— cold punched, number per 100 lbs., 15 (I)
—— lock, 103 (I)
—— wing, 104 (I)

OBSERVATIONS, timing, 269 (I)
Octagons, geometrical and inertial properties, 31 (I)
Offshore dock, 629 (I)
Ohmmeters, 1195 (II)
Oil, 871 (I), 995 (II)
— coolers, 1265 (II)
— diesel fuel, 1355 (I)
— electric locomotives, 385 (II)
——— ship propulsion, 292 (II)
——— traction, 385 (II)
— elimination from feed water, 53 (II)
— engined locomotives, 566 (II)
——— rail cars, 565 (II)
— engines, 186 (II), 204 (II), 1266 (II) (*see also* Internal Combustion Engines)
——— aircraft, 445 (II)
——— automobile, 340 (II)
——— Brotherhood-Ricardo, 1266 (II)
——— connecting rods, 209 (II), 212 (II)
——— crank shafts, 209 (II), 212 (II) 235 (II)
——— cylinder heads, 215 (II)
——— Diesel, Admiralty specification for oil fuel, 1362 (I)
——— filters for, 234 (II)
——— flywheels of, 230 (II)
——— fuel injection, 202 (II)
————— oil specification, 1362 (I)
————— pumps, lignite, 1358 (I)
——— Gleniffer, 1266 (II)
——— high speed, 210 (II)
——— indicators for, 1095 (II)
——— lubrication, 878 (I)
——— marine, 305 (II)
—————— auxiliaries, 308 (II)
—————— fuel, 312 (II)
—————— geared transmission, 316 (II)
—————— in service, 306 (II)
—————— lubrication, 878 (I)
—————— mean indicated pressure, 307 (II)
—————— oil fuel for, 312 (II)
—————— output, dimensions, 315 (II)
—————— power of, 306 (II)
—————— reversing systems, 309 (II)
—————— size of, 306 (II)
—————— structure, 308 (II)
—————— supercharging, 312 (II)
—————— types of, 306 (II)
—————— weight of, 315 (II)
——— on motor car, [illegible] (II)
——— oil pipes, 207 (II)
——— piston cooling, 206 (II)
——— pistons, 207 (II), 210 (II)
——— starting, 207 (II)
——— stresses in, 204 (II), 244 (II)
——— tar fuel oil for, 1359 (I)
——— timing, 216 (II)
——— valves of, 216 (II)
— filters, 878 (I), 234 (II), 1151 (II)
— fired furnaces, 1129 (I), 1360 (I)
— fires, 715 (II)
— flash point, 873 (I)
——— temperature, 53 (I)
— fuel, 1351 (I)
——— British standard specification for, 1362 (I)
——— burners, 1361 (I)

Oil fuel, combustion, air required for, 1361 (I)
—— for locomotives, 561 (II)
—— marine boilers, 298 (II)
—— Navy, 1361 (I)
—— furnaces, 1360 (I)
—— properties of, 1357 (I)
— gas, 1354 (I)
— in exhaust steam, 84 (II)
— insulating, 1501 (I)
— linseed, 987 (II)
— lubricating, 871 (I)
—— flash point, 873 (I)
—— for locomotives, 562 (II)
—— mechanical, 1150 (II)
—— recovery, 878 (I), 1151 (II)
—— viscosity, 874 (I)
— properties of lubricating, 872 (I)
— pumps, 233 (II)
— residual fuel, 1354 (I)
— retaining bronze bearings, 1139 (II)
— separators, 53 (II)
— shale, 1356 (I)
— transformer, 1501 (I)
— turbine, 158 (II)
— turbulent flow, 883 (I)
— viscosity of, 874 (I), 883 (I), 997 (II)
— viscous flow of, 883 (I)
— watch, 996 (II)
— well drilling, wire ropes for, 941 (I)
Oiling rails and flanges on curves, 487 (II)
'Oilless' graphited bearings, 1139 (II)
Olefiant gas, 1368 (I)
Oleic acid fluxes, 1176 (II)
Optical apparatus, lighthouse, 651 (I)
—— pyrometers, 1138 (I)
Optics, 64 (I)
'Optimum' de-aerator, Weir, 1238 (II)
Orcutt gear grinder, 984 (I)
Ordnance scale maps, 254 (I)
—— Survey scales, 254 (I)
Ore dressing, 908 (II)
Ores, composition of the more important, 858 (II)
— copper, tin, lead, zinc, 859 (II)
— magnetic separation of, 916 (II)
Organic sulphur compounds in town gas, 639 (II)
Orifice of a culvert, discharge from the submerged, 671 (I)
Orifices, 671
—— discharge of water through, 671, 745 (I)
—— flow of air through, 756 (II)
—— steam through, 202 (I), 6 (II)
—— for test purposes, 202 (I), 207 (I)
—— gauges, 752 (I)
—— turbine nozzle, 1116 (I)
Otto Beit Bridge, 571 (I)
Outdoor transformers, National Grid, 1502 (I)
Ovens, coking, 1321 (I)
—— electric baking, 1522 (I)
Overhead electric cranes, 802 (II)
—— transmission lines, 1407 (I)
—— porcelain insulators for, 1413 (I)
—— lines, aluminium, 1411 (I)
—— copper, 1407 (I), 1409 (I)
—— Electricity Commissioners Regulations, 1425 (I)
—— Ministry of Transport Regulations, 1420 (I)
Overhead lines, porcelain insulators for, 1430 (I)
—— sags in aluminium, 1411 (I)
—— runways, 808 (II)
—— travelling cranes, 802 (II)
Oxidation, anodic, 1239 (I)
Oxy-acetylene cutting, 1283 (I)
—— welding, 1267 (I), 417 (II)
Oxychloride cements, 354 (I)
Oxygen and air required for combustion of gas, 1366 (I), 645 (II)
—— volume necessary for combustion per cubic foot of gas, 1366 (I)
—— compressor, Weir, 1291 (II)
—— cutting by hand, 1284 (I)
—— properties of, 56 (I)
—— regulators, 1273 (I)
—— supply for acetylene welding, 1269 (I)
Ozonisation of water, 733 (I)

PACKING, 1260 (II), 1281 (II)
—— hydraulic, friction of, 799 (I)
—— labyrinth, turbine, 130 (II)
—— leather, 806 (I)
—— metallic, 1260 (II), 1281 (II)
—— pump, 806 (I), 823 (I)
—— railway sleepers, 486 (II)
—— 'Seatrist' ring, 1259 (II)
—— United Kingdon, 1281 (II)
—— United States Metallic Packing, 1260 (II)
Paint, aircraft, 979 (II)
—— aluminium, 1239 (I), 993 (II)
—— bituminous, 981 (II)
—— cellulose, 988 (II)
—— colour tints, 989 (II)
—— covering capacity, 990 (II)
—— finishes, 995 (II)
—— fire-resisting, 980 (II)
—— galvanised roofing, 239 (I), 993 (II)
—— gloss, 982 (II)
—— graphite, 993 (II), 1296 (II)
—— heat resisting, 980 (II)
—— lead, 981 (II), 986 (II)
—— leadless, 979 (II)
—— luminous, 995 (II)
—— metallic, 993 (II)
—— oils, 987 (II)
—— priming, 978 (II)
—— purchasing, 977 (II)
—— red lead, 981 (II)
—— removal of, 990 (II)
—— shades, 989 (II)
—— solder, 1177 (II)
—— specific gravities, 990 (II)
—— specifications, 980 (II), 984 (II)
—— spraying equipment, 1296 (II)
—— stainers, 979 (II)
—— steelwork, 978 (II), 984 (II)
—— storage of, 977 (II)
—— wet-on-wet, 995 (II)
—— white lead, 982 (II)
—— woodwork, 978 (II), 983 (II)
Painting, 978 (II)
—— fresh cemented surfaces, 994 (II)
—— galvanised iron, 239 (I), 993 (II)
—— spray, 978 (II), 1296 (II)
—— steelwork, 978 (II), 983 (II)
—— woodwork, 978 (II), 983 (II)

Palatine Engineering Co.'s waste water detection system, 1107 (II)
—————— water meters, 1107 (II)
Palm-oil wagon grease, 996 (II)
Panama Canal, 600 (I)
—— dredging, 640 (I)
Panel bricks, 364 (I)
Paper, abrasive, 1114 (I)
—— lead-covered electric cables, 1389 (I)
Parallax, equatorial, of sun, 265 (I)
—— sun's, in altitude, 265 (I)
Parallelgram of forces, 30 (I)
Parian cements, 357 (I)
Parker protection process, 1194 (I)
Parsons steam turbine, blades and caulking pieces for, 127 (II)
—————— dummy loss in, 130 (II)
Partitions, 307 (I)
—— slab concrete, 309 (I)
—— soundproof, 309 (I)
Paste, flour, 359 (I)
Patents, 1001 (II)
Pattern composition, 1262 (I)
—— shop machinery, 1037 (I)
Patterns, weight compared with castings, 154 (I)
Paving, asphalt, 564 (I)
—— brick, 562 (I)
—— cast iron, 554 (I)
—— rubber, 553 (I)
Pearlite, 1145 (I)
Peat, 1329 (I)
Pelton turbines, 837 (I)
—— buckets, 839 (I)
—— jets, 837 (I), 934 (II)
—— regulation, 839 (I)
Pendulum, 23 (I)
—— dynamometer, 39 (I)
—— hardness tester, 1092 (I)
Percussion, centre of, 25 (I)
Permalloy, 1217 (I)
Permanent way, railway, 2006
—— lengthening life of track, 494 (II)
—— Ministry of Transport regulations for, 494 (II)
—— welding, 491 (II)
Permax alloy, 1218 (I)
Perrodil's Hydrodynamometer, 681 (I)
Peterborough canal lift lock, 592 (I)
Petrol, 1352 (I), 201 (II)
—— air gas, 654 (II)
—— aviation, 452 (II)
—— electric lorries, 385 (II)
—— engines (*see also* Automobile engines)
—— —— cooling, 198 (II)
—— —— cycles, 194 (II)
—— —— detonation in, 1353 (I)
—— —— efficiency of, 1353 (I), 194 (II)
—— —— timing, 216 (II)
—— injection, 465 (II)
Petroleum, 1351 (I)
—— lubricants, 871 (I)
—— tar mixtures, 1363 (I)
Phillips rail anchor, 485 (II)
Phœnix rules for lightning conductors, 1517 (I)
'Phons,' 71 (I)
Phosphide, 1145 (I)
Phosphor bronze, 1224 (I)
Phosphorus influence on cast-iron, 1147 (I)
Photo-electric cells and their uses, 1527 (I)
Photographic lens apertures, 65 (I)
Photometer, integrating, 687 (II)
Photometry, 687 (II)
Piers, bridge, 472 (I)
—— and walls, permissible stresses, 321 (I)
—— —— reinforced concrete, 388 (I)
Pig iron, 1141 (I)
Pile sinking with water jet, 440 (I)
Piled foundations, 437 (I)
Piles, 437 (I)
—— concrete, 438 (I)
—— —— moulded-in-place, 438 (I)
—— formulæ for safe load on, 440 (I)
—— interlocking steel, 444 (I)
—— loads carried by, 438 (I)
—— reinforced concrete, 440 (I)
—— safe load on, 440 (I)
—— screw, 443 (I)
—— sinking by water jet, 440 (I)
—— timber, 437 (I)
Piling, interlocked, 444 (I)
—— sheet steel, 444 (I)
—— timber sheet, 445 (I)
Pillars, length of, 227 (I)
—— mild steel, 325 (I)
—— mining, 872 (II)
—— shaft, 872 (II)
—— working loads on, 228 (I), 325 (I)
Pinions, non-resonant, 981 (I)
Pins, taper, 117 (I)
Pipe bends, corrugated, 61 (II)
—— —— creased, 61 (II)
—— —— unbalanced pressures on, 726 (I)
—— couplings, 582 (II)
—— —— Exactor 'Self-Sealing,' 1118 (II)
—— coverings, 68 (II)
—— fittings, dimensions of, 112 (I)
—— —— weights of, 147 (I)
—— flanges, 108 (I)
—— —— American standard, 110 (I)
—— —— boxes for, 68 (II)
—— —— British standard, 108 (I)
—— hangers, 63 (II)
—— jointing materials, weight of, 157 (I)
—— joints, 624 (II), 798 (II)
—— —— steam, 63 (II)
—— lagging, 69 (II)
—— lines (steam), drop of pressure in, 64 (II)
—— —— change-over switch, 1119 (I)
—— —— turbine (water), 726 (I), 485 (I)
—— —— —— pressure changes in, 846 (I)
—— lining for hydraulic mining, 935 (II)
—— —— 'Linatex,' 1301 (II)
—— sockets, 148 (I)
—— supports, 723 (I)
—— threads, 97 (I)
—— —— American standard, 97 (I)
—— —— British standard, 98 (I)
—— —— Whitworth standard, 91 (I)
—— work, hydraulic, 785 (I)
Pipes, aluminium, 152 (I)
—— Angus Smith's composition for iron, 994 (II)
—— asbestos cement, 353 (I)
—— brass, weight of, 141 (I), 151 (I)
—— cast-iron, 114 (I)
—— —— hydraulic power, 786 (I)

Pipes, cast-iron, water incrustation and corrosion, 734 (I)
—— —— weight of, 140 (I), 144 (I)
—— —— —— fittings, 147 (I)
—— concrete, 352 (I)
—— copper, locomotive, 557 (II)
—— —— weight of, 141 (I), 149 (I)
—— diameter of, 734 (I), 738 (I)
—— drain, 372 (I)
—— exhaust, 63 (II)
—— flow of air through, 1079 (I), 66 (II)
—— —— steam through, 64 (II), 66 (II)
—— —— water through, 675 (I), 676 (I), 735 (I), 756 (I), 760 (I), 804 (I)
—— gas, 114 (I), 116 (I), 646 (II)
—— —— British standard, 114 (I)
—— —— dimensions of, 116 (I)
—— —— weight of, 145 (I)
—— hot water heating, 669 (II)
—— hydraulic, 785 (I)
—— —— thickness of, 787 (I)
—— —— test pressure, 114) (I), 116 (I)
—— iron, corrosion of, 734 (I)
—— —— hydraulic, 786 (I)
—— lead, 738 (I)
—— —— strength of, 168 (I)
—— —— weight of, 141 (I), 738 (I)
—— leaky, tracing, 734 (I)
—— pressure in, 722 (I)
—— reinforced concrete, 405 (I), 723 (I)
—— sewage, cast-iron, 113 (I)
—— sewer, British standard, 113 (I)
—— socket and fittings, 148 (I)
—— steam, 60 (II)
—— —— bends, 61 (II)
—— —— cast-iron, 145 (I)
—— —— corrugated, 61 (II)
—— —— coverings of, 68 (II)
—— —— exhaust, 63 (II)
—— —— flow of steam through, 64 (II)
—— —— hangers, 63 (II)
—— —— high pressure and high temperature, 61 (II)
—— —— joints of, 63 (II)
—— —— Lloyd's rules for, 62 (II)
—— —— loss of heat from, 66 (II)
—— —— low pressure, 671 (II)
—— —— of boilers, 60 (II)
—— —— pressure drop, in, 64 (II)
—— —— size of, 112 (I), 63 (II)
—— —— steel bends, 61 (II)
—— —— —— corrosion of, 634 (I)
—— —— —— design of, 723 (I)
—— steel, weldless, weight of, 140 (I)
—— supports, 723 (I)
—— thicknesses of, 112 (I), 146 (I)
—— velocity of water in, 675 (I), 735 (I), 804 (I)
—— water, 112 (I), 145 (I), 734 (I)
—— —— British standard, 113 (I)
—— —— corrosion of, 734 (I)
—— —— design of, 722 (I)
—— —— diameter, 734 (I)
—— —— fittings, weights, 147 (I)
—— —— flow in, 675 (I), 735 (I)
—— —— incrustation of, 734 (I)
—— —— pressure in, 722 (I)
—— —— reinforced concrete, 723 (I)
—— —— steel, 724 (I)
—— —— weight of, 140 (I), 145 (I)
—— —— weldless steel, weight of, 140 (I)
Piping, water wooden stave, 724 (I)
—— acetylene welding, 1273 (I)
—— suction, 801 (I)
Piston cooling, 2066 (II)
—— rod packing rings, 'Seatrist,' 1259 (II)
—— —— United Kingdom metallic, 1281 (II)
—— rods, internal combustion engines 206 (II)
—— speed of marine engines, 288 (II)
—— speeds, 84 (II), 288 (II)
—— trunk, 198 (II)
—— valves, for locomotives, 537 (II)
—— —— Klinger, 1250 (II)
Pistons, internal combustion engine, 207 (II), 211 (II), 335 (II), 461 (II)
Pitch of sound, 70 (I)
Pitches for briquettes, 1331 (I)
Pilot pressure tubes, 578 (I), 680 (I), 755 (I)
Pivots, centre, for swing bridges, 528 (I)
Placer mining, 932 (II)
Plane table (survey instrument) 259 (I)
Planing machines, 1023 (I)
—— power for, 1033 (I)
—— speed table for, 1025 (I)
—— wood, 1041 (I)
Planning and layout of works, 905 (I)
Plant, depreciation of, 1017 (II)
Plaster, hardening, 357 (I)
—— of Paris, 357 (I)
Plastering and lathing, 357 (I)
Plasters, 357 (I)
—— hard, 357 (I)
Plate clutches, 931 (I)
—— laying appliances, 490 (II)
Plates, boiler, transmission of heat through, 25 (II)
—— —— caustic embrittlement of, 51 (II)
—— bulb, 244 (I)
—— shearing, force required, 1044 (I)
Platforms, railway, 493 (II)
Plating metals, 1193 (I)
Platinite, 1218 (I), 1220 (I)
Ploughing, costs, 38 (I)
—— power required for, 38 (I)
Plugs, sparking, 462 (II)
—— wander, colour code, 968 (II)
Plumber's solder, 1176 (II)
Plymouth breakwater, 608 (I)
Pneumatic ballast tamper, 491 (II)
—— caulking hammers, 1078 (I)
—— despatch tubes, 1302 (II)
—— drills, 1030 (I), 1078 (I)
—— foundations, frictional resistance of, 445 (I)
—— hammer, Massey's power, 1120 (I)
—— hammers, 1078 (I), 1120 (I), 1166 (II)
—— hose couplings, 1081 (I)
—— portable tools, 1030 (I), 1082 (I)
—— power hammers, 1120 (I), 1166 (II)
—— —— Alldays & Onions, Ltd., 1169 (II)
—— —— Massey, 1120 (I)
—— process of sinking cylinders, 441 (I)
—— receivers, 1078
—— riveting, 1030 (I), 1082 (I)
—— tools, 1030 (I), 1081 (I)
—— —— for salvage work, 1078 (I)
—— —— portable, 1030 (I), 1078 (I)
—— tubing, flexible, tools, 1081 (I)

Poetsch freezing process of shaft sinking, 869 (II)
Points, railway, 599 (II)
Poisson's Ratio, 178 (I)
Polar curves (light), 687 (II)
—— radius of earth, 263 (I)
Pole star, to find, 259 (I)
—— —— variation of compass by, 259 (I)
Poles, strength and stiffness of wood, 296 (I)
Pole's formula for flow of gas in mains, 645 (II)
Polishing, 1114 (I)
—— machine for terrazzo floors, 1100 (II)
Polygon, area of an irregular, 249 (I)
—— of forces, 30 (I)
Ponds, cooling, 181 (II), 1264 (II)
—— spray, 181 (II), 1264 (II)
Pontoon, sectional dock, 629 (I)
Poppet valve gear for locomotives, 539 (II)
Porcelain insulators for overhead lines, 1413 (I), 1415 (I)
Port Natal dredging plant, 640 (I)
Portable electric drills, 1031 (I)
—— pneumatic tools, 1030 (I), 1078 (I)
Portal bracing, 521 (I)
Portland cement, 355 (I)
—— —— blast furnace, 356 (I)
—— —— bound roads, 563 (I)
—— —— British standard, 355 (I)
—— —— mortar, tints, 342 (I)
—— —— testing, 355 (I)
Potter, F. W., & Soar, Ltd., wing fan, 1285 (II)
Pounds to tons conversion table, 7 (I)
Powder, black, 949 (II)
Power, animal, 35 (I), 37 (I)
—— —— labour produced by, 35 (I)
—— comparison of coal with water, 719 (I)
—— couplings, 1269 (II)
—— expended by force and torque, 38 (I)
—— factor improvement, electrical, 1505 (I)
—— hammers, 1120 (I), 1166 (II)
—— horse, 35 (I), 37 (I), 75 (II)
—— House Components Ltd., cable reels, 1199 (II)
—— hydraulic transmission of, 767 (I), 1110 (II)
—— mechanical, 38 (I)
—— men, 35 (I), 36 (I)
—— measurement of, 35 (I)
—— of horses, 35 (I), 37 (I)
—— —— locomotives, 531 (II)
—— —— men, 35 (I), 37 (I)
—— plant, thermal efficiencies of, 7 (II)
—— pumps, 767 (I), 804 (I)
—— required for machine tools, 1032 (I)
—— —— ploughing, 38 (I)
—— —— slewing swing bridges, 533 (I)
—— stations, condensing plant, 169 (II)
—— —— efficiency of, 7 (II)
—— —— flue gas purification, 48 (II)
—— —— fuel consumption and thermal efficiency of, 10 (II)
—— —— high pressure steam plants, 10 (II)
—— —— turbine installation in, 164 (II)
—— switchboards, 1495 (I)
—— tide, 719 (I)
Power, transmission, choice of, for factories, 906 (I)
—— —— hydraulic, 767 (I), 1110 (II)
—— —— of, by chains, 963 (I), 1155 (II)
—— —— —— leather belting, 909 (I)
—— —— —— ropes, 917 (I)
—— —— —— steel shafting, 889 (I), 906 (I)
—— water, 719 (I)
—— wind, 720 (I)
Precipitation electrostatic, 1236 (II)
Pre-heater air, 17 (II), 1226 (II)
Preloading ball and roller bearings, 864 (I)
Premier water cooling towers, 1264 (II)
Preservation of ropes, 177 (I)
—— —— timber, 285 (I)
Presses, clipping, Brett's, 1166 (II)
—— hydraulic, 789 (I)
Pressure die castings, 1179 (II)
—— drop in pipe lines, 64 (II)
—— gasholders, 631 (II)
—— gauges, hydraulic, 799 (I)
—— in hydraulic mains, 767 (I)
—— —— pipes, 722 (I)
—— mean effective, 74 (II), 80 (II), 84 (II)
—— metric conversion tables, 12 (I)
—— pipe lines, 726 (I)
—— steam conversion tables, 3 (II)
—— tanks, 889 (II)
—— tubes, Pitot, 578 (I), 680 (I)
—— vessels, welded, 33 (II)
—— water, 744 (I)
—— wind, on bridges, 497 (I)
—— —— roofs, 305 (I)
Pressures, bearing, 866 (I), 85 (II)
—— boiling points of water at various, 52 (I)
—— hydraulic test, 113 (I)
—— permissible on masonry, 332 (I)
Primary batteries, 1476 (I)
—— cells, 1476 (I)
Prismatic compass, 259 (I)
Producer gas, 1370 (I)
—— composition of, 1371 (I), 23 (II)
—— furnaces, 1129 (I)
Profiles of gear teeth, 969 (I)
Projection welding, 1292 (I)
Prony brake, 42 (I)
Prony's formula, 579 (I)
Propellants, explosive, 953 (II)
Propeller, model, experiments, law of comparison for, 286 (II)
—— propulsion, 283 (II)
—— shafts, needle bearing, Hardy Spicer, 1272 (II)
Propellers, aeroplane (*see* Airscrews)
—— noise from, 79 (I)
Properties of gases, 54 (I), 56 (I)
Props, steel, 1104 (II)
Propulsion, ship, elements of, 283 (II), 1528 (I)
'Proto' breathing apparatus, 1299 (II)
Puddle dams, 447 (I)
Pugging, 314 (I)
Pulleys, 21 (I)
—— and drums, rope, sizes of, 921 (I), 939 (I), 944 (I)
—— belt, 913 (I)
—— —— camber, 913 (I)

Pulleys, belt, velocity ratios, 914 (I)
—— mining, 898 (II)
—— rope driving, 919 (I), 921 (I), 928 (I)
—— wire rope, 936 (I)
Pulsometer Engineering Co., Ltd., centrifugal pump, 1125 (II)
—— —— high lift turbine pump, 1125 (II)
—— —— steam pump, 831 (I), 1124 (II)
—— —— 'Stereophagus' pump, 1293 (II)
—— —— water filter, 1108 (II)
Pulverised coal, 1337 (I), 1186 (II)
—— advantages of, 1346 (I)
—— at sea, 1342 (I)
—— burners, 1340 (I), 1188 (II)
—— central system, 1339 (I)
—— combustion chambers, refractory linings, 1346 (I)
—— feeders, 1340 (I)
—— fired boilers, 1188 (II)
—— for locomotives, 1345 (I)
—— fuel, 1337 (I), 1186 (II)
—— mill house plant for central system, 1339 (I)
—— mills and mill house plant, 1339 (I), 1347 (I), 1186 (II)
—— —— cost of, 1348 (I)
—— power of motors, 1341 (I)
—— pulveriser units, self-contained 1341 (I), 1186 (II)
—— systems, Clarke, Chapman, 1186 (II)
—— —— 'Lopulco,' 1188 (II)
—— transportation of, 1340 (I)
Pumping, boiler pressure rquired for, 806 (I)
—— by compressed air, 826 (I)
—— engines, 82 (II)
—— —— wind, 753 (II)
—— from wells, 808 (I)
—— in mines, 899 (II)
Pumps, 801 (I), 1110 (II), 1113 (II), 1121 (II), 1262 (II), 1293 (II)
—— air, 178 (II)
—— —— hydro, Madan's, 1123 (II)
—— —— lift, 826 (I)
—— —— —— arrangement of compressing plant, 830 (I)
—— —— —— H.P. consumed, 830 (I)
—— —— —— aize of mains, 829 (I)
—— —— —— submergences, 829 (I)
—— artesian well, 808 (I)
—— axial flow, 809 (I)
—— boiler feed and service, 815 (I), 181 (II), 1121 (II)
—— borehole, 809 (I)
—— branch proportions, 804 (I)
—— capacity of, 805 (I)
—— centrifugal, 810 (I), 1125 (II), 1130 (II)
—— —— design of, 822 (I)
Pumps, centrifugal, erection of, 823 (I)
—— —— impellers, 816 (I)
—— —— mine, 900 (II), 935 (II)
—— —— packing of, 823 (I)
—— —— starting of, 824 (I)
—— Clarke, Chapman steam, 1121 (II)
—— classification of, 803 (I)
—— compressed air, 826 (I)
—— deep well, 808 (I)
—— direct acting, 806 (I), 1121 (II), 1127 (II)
—— ends materials, 806 (I)
—— extraction, 1262 (II)
—— feed, 814 (I), 1121 (II), 1127 (II)
—— flow in pipes, 804 (I), 181 (II)
—— foot valves and strainers, 803 (I)
—— force, 181 (II)
—— fuel, 343 (II)
—— governors, Cope's, 1244 (II)
—— high-speed variable delivery, 770 (I), 1110 (II)
—— hydraulic power, 767 (I)
—— impellers, 816 (I)
—— internal combustion engine, water and oil, 233 (II)
—— jet, 180 (II)
—— Keelavite, 1110 (II)
—— Madan's airhydro pump, 1123 (II)
—— mechanical lubricating, Tecalemit, 1150 (II)
—— mining, 899 (II)
—— oil, 233 (II)
—— packings, leather, 806 (I)
—— power, 767 (I), 804 (I)
—— Pulsometer centrifugal, 1225 (II)
—— —— high-lift turbine pump, 1125 (II)
—— —— steam pump, 831 (I), 1124 (II)
—— —— 'Stereophagus,' 1293 (II)
—— ram, 768 (I)
—— reciprocating, 803 (I)
—— —— discharge from, 805 (I)
—— —— efficiency of, 805 (I)
—— rotary, 824 (I)
—— —— classification of, 826 (I)
—— steam, 831 (I), 1121 (II), 1124 (II)
—— 'Stereophagus,' 1293 (II)
—— strainers, 803 (I)
—— suction piping, 801 (I)
—— swashplate, 824 (I)
—— tests, British Standard, 747 (I)
—— treble ram, 768 (I)
—— turbine, 812 (I), 1125 (II)
—— —— boiler feed, 814 (I)
—— turbo-feed steam, Weir, 1125 (II)
—— 'VSG,' 1113 (II)
—— valves, foot, 803 (I)
—— variable delivery pressure, 770 (I), 1110 (II)
—— volute, 811 (I)
—— Weir direct-acting steam, 1127 (II)
—— —— extraction, 1262 (II)
—— —— turbo-feed steam, 1129 (II)
—— well, 808 (I)
—— Woodeson steam, 1121 (II)
Punching machines, power for, 1044 (I)
—— plates, power required for, 1044 (I)
Purchase of machinery, 1036 (II)
Purification of gas, 628 (II), 635 (II)
—— water, 729 (I), 733 (I), 1108 (II)

Purity of water, standard of, 730 (I)
Purlins, roof, 234 (I)
Pyramid diamond hardness scale, 1094 (I), 1190 (I)
Pyrometers, electric resistance, 1139 (I)
—— fusion, 1136 (I)
—— heat radiation, 1137 (I)
—— optical, 1138 (I)
—— sentinel, 1137 (I)
—— thermoelectric, 1138 (I)
Pyrometry, 1135 (I)

QUARRYING, 421 (I) (*see also* Excavation)
Quay cranes, 622 (I)
—— walls, 617 (I)
Quayside appliances, 622 (I)
—— —— burtoning, 623 (I)
—— —— coaling appliances, 623 (I)
—— —— transporters, 623 (I)
Queen-post roof, 304 (I)
Quenching baths for tempering cutting tools, 1072 (I)
—— —— liquids, 1073 (I), 1179 (I)
Quicklime, 342 (I)
Quinquennial valuation of plant, 1019 (II)

'R.R.' alloy, 1233 (I)
Races, head, 722 (I)
Racks, strainer, 722 (I)
Radians to degrees, 1047 (II)
Radiant heat transmission in boiler furnaces, 25 (II)
Radiator cleanser, Boilerine, 1239 (II)
Radiators, heating surface of, 660 (II)
Radio communication, 957 (II)
amplification, 971 (II)
amplifiers, classification of, 972 (II)
—— typical operating conditions, 971 (II)
attenuators, 968 (II)
base connexions, values, 961 (II)
caps, screening, 970 (II)
capacitors, colour code, 965 (II)
—— electrolytic, 967 (II)
—— insulation resistance, British, 966 (II)
—— leakage from, 966 (II)
cartridge fuses, colour code, 968 (II)
cathode follower, 972 (II)
colour codes, 963 (II), 965 (II), 968 (II)
electrolytic capacitors, 967 (II)
factors, 970 (II)
fixed capacitors, 965 (II)
fixed resistors, 963 (II)
frequency modulation, 959 (II)
—— —— receivers for, 959 (II)
—— —— experimental transmissions, 960 (II)
—— tables, 958 (II)
fuses, cartridge, 968 (II)
Inductance, 969 (II)
insulation resistance, capacitors, 966 (II)
interlacing, television, 961 (II)
K factors, table of, 970 (II)
leakage, capacitors, 966 (II)
loudspeaker, matching to valve, 972 (II)
mains transformers, 967 (II)
mercury vapour rectifiers, 963 (II)
microphony, 963 (II)
miniature valves, 962 (II)
modulation, frequency, 958 (II)
—— phase, 959 (II)
plugs, wander, colour code, 968 (II)
preferred values, resistors, 965 (II)
programmes, television, 961 (II)
resistors, fixed, 963 (II)
—— —— colour code, 963 (II)
—— —— preferred values, 965 (II)
rosa's correction factors, 970 (II)
screening cans, effect of, 970 (II)
—— reduction in inductance due to, 971 (II)
solenoids, short, self capacity of, 971 (II)
television, 960 (II)
—— details of system, 960 (II)
—— interlacing, 961 (II)
—— programmes, 961 (II)
—— service, extension of, 961 (II)
tone control, 969 (II)
transformers, mains, 967 (II)
values, 961 (II)
—— base connexions, 961 (II)
—— code of practice, 963 (II)
—— matching to loudspeaker, 972 (II)
—— miniature, 962 (II)
wander plugs, colour code, 958 (II)
wavelengths, 957 (II)
—— conversion table, 958 (II)
Radius, equatorial, of earth and sun, 263 (I)
—— of gyration, 22 (I), 26 (I)
Rafters, 234 (I), 305 (I)
Rail bearers of bridges, 513 (I)
—— cars, 565 (II)
—— oil engined, 566 (II)
—— motor cars and trains, 565 (II)
Rails, 469 (II)
—— anchors and keys, 484 (II)
—— axle load allowable on, 478 (II)
—— British standard, 472 (II)
—— bullhead, 472 (II)
—— —— areas of head, etc., 475 (II)
—— chairs, cast iron, 490 (II)
—— check, 480 (II)
—— conductor, 478 (II)
—— creep prevention, 484 (II)
—— dimensions of, 474 (II)
—— fishbolts and nuts, 480 (II)
—— fishplates, 749 (II)
—— flat-bottomed or flanged, 475 (II)
—— —— areas of head, etc., 476 (II)
—— heat treatment, 478 (II)
—— mechanical loading and unloading of, 491 (II)
—— mine haulage, 891 (II)
—— oiling on curves, 487 (II)
—— radii of curvature and versed sines, 501 (II)
—— short in curves, etc., 478 (II)
—— special steels for, 477 (II)
—— steel, 472 (II)
—— —— conductor, 478 (II)
—— superelevation of outer on curves, 500 (II)
—— track-laying machine, 491 (II)
—— wear of, 513 (II)
—— weight of, 474 (II), 477 (II)
—— —— minimum, 477 (II)
—— welded, 491 (II)
Railways, automatic train control, 609 (II)

Railways, axle load, maximum allowable, 478 (II)
—— —— lubricants, 995 (II)
—— axles, 550 (II)
—— ballast, 485 (II)
—— brakes, 578 (II)
—— bridges, 481 (I)
—— —— allowance for impact on, 500 (I)
—— —— live loads on, 481 (I)
—— —— locomotive weight distribution on, 481 (I)
—— —— momentum of trains on, 500 (I)
—— —— rolling loads on, 481 (I)
—— —— standard clearances, 479 (I)
—— —— —— loadings, 482 (I)
—— —— working stresses on, 493 (I)
—— building costs, 498 (II)
—— carriages, 571 (II)
—— —— axle boxes, 576 (II)
—— —— axles, 550 (II)
—— —— —— greases, 995 (II)
—— —— brakes, 578 (II)
—— —— couplings, 578 (II)
—— —— heating by steam, 578 (II)
—— —— springs, 577 (II)
—— —— wheels, 577 (II)
—— check-rails, 480 (II)
—— chemical weed killing on, 494 (II)
—— clearances on curves, 493 (II)
—— concrete sleepers, 488 (II)
—— construction, cost of, 498 (II)
—— crossings, 502 (II)
—— —— building up worn, by electric welding, 491 (II)
—— curves, 493 (II), 500 (II)
—— —— clearance on, 493 (II)
—— —— oiling rails and flanges on, 487 (II)
—— —— radii table, 501 (II)
—— —— ranging out, 509 (II)
—— —— resistance of, 529 (II)
—— —— short rails in, 478 (II)
—— —— superelevation of outer rail for, 500 (II)
—— —— transition, 510 (II)
—— cuttings, volume of earth in, 421 (I)
—— dynamometer cars, 39 (I)
—— electric, 585 (II)
—— —— conductor rails, 478 (II)
—— —— cost of, 588 (II)
—— —— systems in use, 587 (II)
—— —— traction on long distance, 585 (II)
—— —— —— suburban, 590 (II)
—— —— —— costs, 588 (II)
—— electrification, Ministry of Transport Committee's report on main line, 587 (II)
—— fishbolts and nuts, British standard for, 480 (II)
—— fishplates, 479 (II)
—— gauges, widening of, on curves, 480 (II)
—— gradients, 493 (II), 512 (II)
—— —— resistance due to, 529 (II)
—— high-speed services, 566 (II)
—— land required for, 499 (II)
—— light, cost of, 500 (II)
Railways, light, or narrow gauge, 499 (II)
—— locomotives (*see also* Locomotives), American, 520 (II)
—— —— Great Western, 518 (II) 520 (II)
—— —— London Midland and Scottish, 516 (II), 518 (II)
—— —— London and North Eastern, 516 (II), 518 (II)
—— —— Southern, 518 (II), 520 (II)
—— main line electrification, 587 (II)
—— Ministry of Transport regulations, 493 (II)
—— motor cars, steam, 565 (II)
—— narrow gauge, 499 (II)
—— packing of sleepers, 486 (II)
—— permanent way, 469 (II)
—— —— heights and widths, 493 (II)
—— —— Ministry of Transport regulations, 494 (II)
—— —— platelaying mechanical appliances, 491 (II)
—— —— welding maintenance, 492 (II)
—— platforms, 493 (II)
—— points, mechanically operated, 599 (II)
—— rails, 469 (II), 491 (II)
—— rolling stock, 571 (II)
—— —— axle boxes, 576 (II)
—— —— brakes, 578 (II)
—— —— couplings, 578 (II)
—— —— material, British standard specification, 550 (II)
—— —— roller bearings, 577 (II)
—— —— Southern Railway electrified lines, 576 (II)
—— —— springs, 577 (II)
—— —— wheels, 577 (II)
—— signalling, 595 (II)
—— signals, colour light, 606 (II)
—— sleepers, 481 (II)
—— —— concrete, 488 (II)
—— —— packing of, 486 (II)
—— —— steel, 482 (II)
—— steel sleepers, 482 (II)
—— switches and ceossings, 494 (II)
—— track circuiting, 512 (II), 601 (II)
—— —— laying machine, Bretland's, 491 (II)
—— —— maintenance of, 491 (II)
—— train acceleration, 531 (II)
—— —— control, 608 (II)
—— —— resistance, 526 (II)
—— wagons, 574 (II)
—— —— axle-boxes, 576 (II)
—— —— axles, 550 (II)
—— —— couplings, 578 (II)
—— —— springs, 577 (II)
—— —— wheels, 577 (II)
—— wear of rails, 513 (II)
—— welding in permanent way maintenance, 491 (II)

Railways, wooden sleepers, 481 (II)
—— working costs, comparative, steam and electric, 588 (II)
Rainfall, 659 (I)
—— absorption and evaporation, 663 (I)
—— and forests, 667 (I)
—— catchment, 662 (I)
—— flood discharge, 665 (I)
—— forests and, 667 (I)
—— gauges, 669 (I)
—— heavy, in various countries, 661 (I), 667 (I)
—— in British Isles, 659 (I)
—— —— foreign countries, 661 (I)
—— losses and run-off, 662 (I)
—— monsoon, Nagpur, 701 (I)
—— power from, 719 (I)
—— quantities and rates, 664 (I)
—— run-off, 662 (I), 668 (I)
Rain gauges, 669 (I)
Raising and winzing, mining, 869 (II)
Ram leathers, 806 (I)
—— pumps, 768 (I), 831 (I)
Rammed furnace linings, 374 (I)
Rams, hydraulic, 831 (I)
Ranging, 249 (I), 509 (II)
Rankine cycle, 1301 (I)
—— formula for safe load on piles, 440 (I)
Ransome and Marles Bearing Co., Ltd., ball and roller bearings, 1142 (II)
Ransomes, Sims & Jefferies, Industrial Trucks, 1274 (II)
—— —— motor lawn mowers, 1277 (II)
Ransome's artificial stone, 358 (I)
Rapid case-hardening methods, 1188 (I)
—— hardening cements, 355 (I), 357 (I)
—— machining steels, 1162 (I)
'Rapidor' sawing machines, 1161 (II)
Rating allowances, 1024 (II)
Raw water, temporary hardness of, 51 (II)
Raymond bowl mill, 1187 (II)
Reaction soldering, 1260 (I)
—— turbines, steam, 125 (II)
—— —— water, 835 (I), 840 (I)
—— —— —— regulation, 843 (I)
Rebhann's construction, 458 (I)
Receivers, air, 1078 (I)
Recipro-turbine machinery, 289 (II)
Reciprocals, squares, cubes, roots, 1062 (II)
Receiprocating pumps, 803 (I)
Recorders, CO_2, 1315 (I)
—— V-notch weir, 197 (I)
Rectangular beams, 222 (I), 401 (I)
—— slabs, reinforced concrete, 393 (I)
—— Weirs, 746 (I), 750 (I)
Rectifiers, mercury arc, electric, 1485 (I)
—— —— grid control, 1486 (I)
Red lead paints, 981 (II)
Reducing valves, 59 (II)
Reduction gear, 290 (II)
Redwood seconds conversion factors, 874 (I)
Reed signals, 656 (I)
Reels, self-winding cable, 1199 (II)
Refacers, valve, 1255 (II)
Refining steel, 1168 (I)
Reflecting power of surfaces, 686 (II)
Reflection of sound, 72 (I)
Refraction, mean, table of, 270 (I)
Refractory cements, 354 (I), 376 (I)
—— linings for pulverised fuel combustion chambers, 1346 (I)
—— materials, 370 (I)
—— —— melting points of, 371 (I) 1134 (I)
—— mortars and cement, specification for, 373 (I)
Refrigerants, 729 (II)
Refrigerating compressors, 736 (II)
—— condensers, 736 (II)
—— evaporators, 737 (II)
—— machine, the efficiency of a 1302 (I)
—— machinery, 736 (II)
—— —— ammonia compression, 724 (II)
—— —— automatic, 740 (II)
—— —— calcium brine for, 730 (II)
—— —— carbonic acid, 727 (II)
—— —— coolers, 738 (II)
—— —— for ships, 743 (II)
—— —— heat transference in, 738 (II)
—— —— indicator diagrams, 721 (II)
—— —— insulation, 733 (II)
—— —— operating, 741 (II)
—— power, 721 (II)
Refrigeration, 719 (II)
—— compression system, 736 (II)
Refuse, calorific value of, 1349 (I)
—— destructors, 1349 (I), 1191 (II)
—— —— City of Glasgow, 1350 (I)
—— —— Heenan's, 1191 (II)
Regenerative braking on tramways, 382 (II)
—— dynamometers, 1093 (II)
—— surface condenser, 1261 (II)
Regulator, air mining, 704 (II)
—— feed water, 1243 (II)
—— —— Copes, 1243 (II)
—— —— Crosby, 1245 (II)
—— —— Mumford, 1248 (II)
—— —— Thermofeed, 1247 (II)
—— —— Weir, 1247 (II)
—— induction, 1504 (I)
—— moving-coil, 1504 (I)
—— oxygen and acetylene, 1273 (I)
Rehbock formula for weirs, 747 (I)
Reinforced concrete, 343 (I), 381 (I)
—— arches, 409 (I)
—— beams, 399 (I)
—— —— deflection of, 405 (I)
—— —— inverted, 404 (I)
—— —— stresses in, 404 (I)
—— bins, 405 (I)
—— bridges, arches of, 409 (I)
—— buildings, code of pratice for, 397 (I)
—— calculation methods, 410 (I)
—— carriageways, 561 (I)
—— chimneys, 464 (I)
—— columns, 396 (I)
—— conical coverings, 408 (I)
—— construction, 381 (I), 398 (I)

Reinforced concrete, construction, Coignet system, 398 (I)
—— Considère system, 398 (I)
—— Hennebique system, 398 (I)
—— cracking, 417 (I)
—— designing, 399 (I)
—— domes, 408 (I)
—— durability, 417 (I)
—— failures, causes of, 417 (I)
—— fire resisting floors, 398 (I)
—— girders, 399 (I)
—— L.C.C. regulations, 381 (I), 393 (I)
—— piles, 440 (I)
—— pipes, 405 (I), 723 (I)
—— practical notes, 417 (I)
—— reinforcement rods, 398 (I), 418 (I)
—— reservoirs, 405 (I)
—— retaining walls, 407 (I), 461 (I)
—— roads, 561 (I)
—— shear reinforcement, 395 (I)
—— silos, 406 (I)
—— slabs, 393 (I)
—— specification for, 414 (I)
—— storage bins, 406 (I)
—— stresses in, 404 (I)
—— beams, 404 (I)
—— structures, 399 (I)
—— systems, 398 (I)
—— table of sectional area of rods, 418 (I)
—— T-beams, 402 (I)
—— tests on, 386 (I)
—— walls, 388 (I), 407 (I)
—— work, specification for, 414 (I)
—— working stresses, 389 (I)
Reinforcing rod, areas of, 418 (I)
Relay governors, 93 (II)
Relief valves, 780 (I)
Remote controls, Lockheed hydraulic, 795 (I)
Renold & Coventry Chain Co., Ltd., chain drives, 1158 (II)
Rents and royalties, mining, 848 (II)
Repairs by welding, 1180 (II)
—— to worn parts by metal deposition, 1082 (I)
Rescue apparatus for mines, 906 (II)
Reservoirs, 582 (I), 729 (I)
—— available discharge from, 721 (I)
—— circular reinforced concrete, 405 (I)
—— evaporation and absorption from, 701 (I)
—— marginal grouting of, 693 (I)
—— reinforced concrete, 405 (I)
—— silting of, 713 (I)
—— storage, 582 (I), 729 (I)
Residual fuel oils, 1354 (I)
Resin varnishes, 988 (II)
Resistance alloys, 1252 (I), 1379 (I)
—— electric materials, metalled, 1379 (I)
Resistance furnaces, 1129 (I)
—— of ships, 279 (II)
—— trains, 526 (II)
—— testing instruments, Evershed & Vignoles, Ltd., 1194 (II)
—— welding, 1290 (I)
Resistivity, electrical, of metals, 1379 (I)
Resistors, fixed, 963 (II)
—— colour code, 963 (II)
—— preferred values, 965 (II)
'Resolutor' coal pulveriser, 1186 (II)
Resuscitating apparatus, 'Novox,' 1300 (II)
Retaining-walls, 407 (I), 455 (I)
—— brick, 460 (I)
—— effect of weight of building on, 457 (I)
—— equilibrium of, 456 (I)
—— foundations of, 459 (I)
—— pressure of earth on, 458 (I)
—— water on, 455 (I)
—— reinforced concrete, 460 (I)
—— rules for, 460 (I)
—— Tautwine's rule for, 461 (I)
—— wharf, 463 (I)
Retort benches, 619 (II)
—— house foul main, 625 (II)
—— governors, 625 (II)
—— of gasworks, 619 (II)
Retorts, gas, 374 (I), 619 (II)
—— heating, 621 (II)
—— joints for ascension pipes, 624 (II)
—— vertical, 619 (II)
Reverberatory furnaces, 1125 (I), 1128 (I)
Rheostatic braking, 382 (II)
Rheostats, field, 1500 (I)
Rhone-Marseilles Canal, 587 (I)
Rice irrigation, 708 (I)
Ridged roof truss multipliers, 237 (I)
Rigby's expansion rocker bearing, 505 (I)
Riley Stoker Co., Ltd., 'Robot' automatic stoker, 1226 (II)
Rim clutches, 929 (I)
Ringlemann's smoke charts, 1336 (I)
Ringrose firedamp detector, 902 (II)
Rings, iron, strength of, 171 (I)
Rio Salado dam, 695 (I)
Ritter's method of sections, 519 (I)
River bank protection, 579 (I)
—— of the Missouri, 580 (I)
—— banks, erosion of, 578 (I)
—— canalisation, 583 (I)
—— control of flow and floods, 578 (I), 668 (I), 716 (I)
—— and diversion weirs, 699 (I)
—— dredging, 582 (I), 637 (I)
—— flow under bridges, 490 (I)
—— heightening by a weir, 674 (I)
—— regulation, cost of, 582 (I)
Rivers, cost of canalisation of, 583 (I)
—— critical velocity in, 579 (I)
—— flow and discharge of, 578 (I), 716 (I)
—— navigation of, 577 (I)
—— silt borne by, 714 (I)
—— to measure base line across, 251 (I)
—— width of, 251 (I)
—— raise the height of, by a weir, 674 (I)
—— velocity of flow in, 578 (I), 680 (I)
Riveted girder-work, 229 (I)
—— joints, 229 (I), 232 (I), 37 (II), 420 (II)

Riveted joints, fatigue of, 188 (I)
Riveting, 232 (I)
—— aircraft, 420 (II)
—— boiler, pressure required for, 33 (II)
—— bridge, 508 (I), 511 (I)
—— hammers, 1078 (I)
—— pneumatic, 1030 (I), 1078 (I), 1082 (I)
Rivets, 165 (I), 230 (I)
—— and plates, strength of, 232 (I)
—— aircraft, 420 (II)
—— copper, weight of, 156 (I)
—— iron, 230 (I)
—— —— weight of, 156 (I)
—— steel, 165 (I), 230 (I), 558 (II)
—— —— tests for, 165 (I), 232 (I)
—— strength of, 232 (I)
Roads and Highways, 543 (I)
—— acceleration and deceleration lanes, 561 (I)
—— accident statistics, 556 (I)
—— administration, 543 (I)
—— asphalt, 564 (I)
—— bearers of bridges, 513 (I)
—— binding materials, experimental, 554 (I)
—— brick, 562 (I)
—— bridges, 571 (I)
—— —— impact effect on, 501 (I)
—— —— loadings on, 482 (I), 485 (I), 488 (I), 571 (I)
—— —— standard clearances on, 479 (I)
—— —— working stresses on, 493 (I)
—— carriageway construction, 546 (I), 561 (I)
—— cast iron, 554 (I)
—— cement bound, 563 (I)
—— concrete, 561 (I)
—— construction, 545 (I)
—— cost of, 547 (I), 563 (I)
—— —— maintenance, 547 (I)
—— crusts, 565 (I)
—— design, 557 (I), 559 (I)
—— embankments, 561 (I)
—— excavating, 568 (I)
—— footpath construction, 567 (I)
—— gradients, 559 (I)
—— grouted, 565 (I)
—— high speed, 557 (I)
—— highway statistics, 544 (I)
—— kerbing, sandstone, 552 (I)
—— laboratory tests, 548 (I)
—— lanes, acceleration and deceleration, 561 (I)
—— law, 543 (I)
—— literature, 574 (I)
—— macadam, 564 (I)
—— machinery, 568 (I)
—— maintenance cost of, 547 (I), 565 (I)
—— materials, 551 (I)
—— —— experimental, 554 (I)
—— mileage in relation to population and area, 544 (I)
—— mileages per square mile, 544 (I)
—— Ministry of Transport load for bridges, 488 (I)
—— mixing machines, 569 (I)
—— National Physical Laboratory tests of roadstones, 548 (I)
—— overseas systems, 545 (I)
—— planning and design, 566 (I)
—— radius at bends of, 559 (I)
Roads regulations, 554 (I)
—— reinforced concrete, 561 (I)
—— research, 548 (I), 551 (I)
—— resistance to traction on, 324 (II)
—— road-metal rocks, 533 (I)
—— rubber, 553 (I)
—— side trees, 570 (I)
—— sideway force coefficients, 558 (I)
—— slab concrete, 561 (I)
—— specifications, 563 (I)
—— speed limit, 545 (I)
—— speedways, 556 (I)
—— stones, tests of, 548 (I), 553 (I)
—— statistics, 544 (I)
—— summit visibility, 559 (I)
—— superelevation at bends in, 559 (I)
—— surface heater, 569 (I)
—— —— rugosity, 558 (I)
—— —— tarring of, 570 (I)
—— —— treatment, 567 (I)
—— tar macadam, 564 (I)
—— testing machines, 548 (I)
—— Traffic Act, 554 (I)
—— tree planting on, 570 (I)
—— Trunk Roads Act, 543 (I)
—— tunnels, 571 (I)
—— vehicles, maximum sizes, 551 (I)
—— —— provision as to tyres, 555 (I)
—— visibility at summits, 559 (I)
—— width of, 560 (I)
'Robot' boiler feed regulator, 1247 (II)
—— stoker, 1226 (II)
Rock blasting, 641 (I)
—— boring, 861 (II)
—— breakers, or chisels, 642 (I)
—— drilling and blasting, 641 (I)
—— fill dams, 658 (I), 696 (I)
—— sub-aqueous, removal of, 541 (I)
Rocks used for road metal, 553 (I)
Rockwell hardness numbers, 1190 (I)
—— —— tester, 1093 (I)
Rods, concrete reinforcement, 418 (I)
Roe's saddle and toggle clips, 831 (II)
'Rok' roofing, 1098 (II)
Rolled copper, lead, and brass, sheet and bars, weight of, 131 (I)
—— iron, flat, weight of, 129 (I)
—— steel, flat, weight of, 131 (I)
—— —— sections, limiting dimensions, 239 (I)
Roller bearings, 849 (I), 851 (I), 1131 (II)
—— —— for rolling stock, 577 (II)
—— —— high temperatures, 862 (I)
—— —— Hoffmann, 1140 (II)
—— —— housings, 860 (I)
—— —— journal, 851 (I)
—— —— lubrication of, 862 (I)
—— —— needle, 853 (I)
—— —— preloading, 864 (I)
—— —— Ransome & Marles, 1142 (II)
—— —— Skefco, 1146 (II)
—— —— taper, 853 (I), 858 (I)
—— —— Timken taper, 1131 (II)
—— chains, 961 (I), 1155 (II)
Rollers for swing bridges, 527 (I)
Rolling-mill gears, 972 (I)
—— period of ships, 275 (II)
Rolls, gold mining, 930 (II)
—— Royce alloys, 1233 (I)
Roofing, asbestos cement slates, 299 (I)

Roofing, corrugated iron, 233 (I), 236 (I), 301 (I)
—— 'Flextile,' 1098 (II)
—— galvanised, paint and varnish, for, 239 (I)
—— lining Willesden paper, 1099 (II)
—— materials, 297 (I)
—— —— heat-insulating value of, 299 (I)
—— 'Rok,' 1098 (II)
—— 'Ruberoid,' 1099 (II)
—— self-supporting, 237 (I)
—— slates, 297 (I)
—— tiles, clay or marl, 367 (I)
—— —— concrete, 353 (I)
—— zinc, 299 (I)
Roofs, 233 (I), 297 (I)
—— angles of, 301 (I), 392 (I)
—— collar beam, 303 (I)
—— corrugated iron for, 233 (I), 236 (I), 239 (I), 301 (I)
—— coverings of, 233 (I)
—— king-post, 304 (I)
—— load on, 300 (I)
—— paint for, 239 (I)
—— queen-post, 304 (I)
—— rafters, 305 (I)
—— ridged, truss multipliers, 237 (I)
—— single-span or couple, 303 (I)
—— steel, framing and covering, 233 (I)
—— weight of components of, 301 (I)
—— —— snow on, 305 (I)
—— wind pressure on, 305 (I), 319 (I)
—— wooden, parts of, 301 (I)
Roots, squares, cubes, reciprocals, 1062 (II)
Rope and cables, lengths, 1 (I)
—— brake, Hirn's, 41 (I)
—— cotton, table of, 921 (I), 924 (I)
—— drives, design of, 920 (I), 924 (I), 1152 (II)
—— driving, 916 (I), 1152 (II)
—— —— cotton, 920 (I), 924 (I)
—— —— fluctuating loads, 926 (I)
—— —— power of, 919 (I)
—— —— pulleys for, 921 (I), 930 (I)
—— —— splicing, 927 (I)
—— —— systems of, 922 (I)
—— —— vee, 917 (I)
—— hemp, preservation of, 177 (I)
—— —— strength of, 176 (I)
—— iron corrosion of, 178 (I)
—— Lang-lay wire, 933 (I)
—— manila, 176 (I)
—— preservation of, 177 (I)
—— splicing, 927 (I)
—— strength of, 176 (I)
—— vee, 917 (I)
—— weight of, 131 (I)
—— wire (*see* Wire Ropes)
Ropeways, aerial (*see* Aerial ropeways)
Rosa's correction factors, 970 (II)
Rosin-oil soap, 996 (II)
'Rosseline' water treatment, 1239 (II)
Rotary pumps, 824 (I)
—— —— classification of, 826 (I)
—— steam flow meter, 204 (I)
—— transmitters, 795 (I)
Royal Commission on sewage, 781 (II)
Royalties, mining, 848 (II)
Rubber, cables, 1415 (I)
—— cements, 353 (I)
Rubber, Linatex, 1301 (II)
—— lined pipes, 935 (II), 1301 (II)
—— paving, 553 (I)
Rubble masonry, 329 (I)
Ruberoid Co., Ltd., roofing, 1099 (II)
Ruble elevator, mining, 935 (II)
Rudders, 282 (II)
Runabout cranes, 806 (II)
Runways, overhead, 808 (II)
Russian weights and measures, 10 (I)
Rust prevention in reinforced concrete, 349 (I)
—— preventive, 990 (II)
—— removal, 989 (II)
Rustless coating for steel, 1192 (I)
—— steel, selenium 1200 (I),
Ryburn Dam, 688 (I)
Ryland's table of iron wire, 162 (I)
—— —— steel wire, 162 (I)
Ryves' formula for flood discharge, 665 (I)

SAFE load on brick walls, 437 (I)
—— —— bridges, 482 (I), 485 (I), 488 (I)
—— —— columns, 229 (I)
—— —— floors, 437 (I)
—— —— materials, 436 (I)
—— —— ordinary foundations, 436 (I)
—— —— piles, 440 (I)
—— —— stanchions, 228 (I)
—— —— stone walls, 436 (I)
—— —— wood, 295 (I)
Safety explosives, 864 (II)
—— lamps, 904 (II)
—— valves, 1273 (I), 55 (II), 1249 (II)
—— —— areas of, 55 (II)
—— —— —— proposed new rules for, 57 (II)
—— —— Board of Trade rules, 56 (II)
—— —— Bureau Veritas rules, 57 (II)
—— —— Lloyd's rules, 57 (II)
—— —— springs of, Board of Trade rule, 56 (II)
Sale and purchase of machinery, 1256 (II)
Salinometer, Crockatt, 1256 (II)
—— Evershed & Vignoles, Ltd., 1197 (II)
Salt solutions, boiling points of, 52 (I)
—— velocity method of measuring flow, 756 (I)
Salts, metallic, melting points, 1136 (I)
—— mixtures of, 1136 (I)
Salvage work, pneumatic tools for, 1078 (I)
'Salvus' breathing apparatus, 1300 (II)
Sand, blasting, 1115 (I)
—— —— nozzle sizes, 1116 (I)
—— lime bricks, 364 (I), 366 (I)
—— moulding, 1261 (I)
—— pump dredgers, 635 (I)
Sandberg steel sleepers, 484 (II)
Sander machines, 1040 (I)
Sandstone kerbing, 552 (I)
Sanitary engineering, 769 (II), 1293 (II)
—— ware, 370 (I)
Sardis dam, 695 (I)
Saturated salt solutions, boiling points of, 52 (I)
—— steam, properties of, 3 (II), 176 (II)
Saw bench speeds, 1038 (I)
Sawing machines, Rapidor, 1161 (II)
Saws, band, 1039 (I)

Scaffolding, 418 (I), 1104 (II)
——— knots, 449 (I)
——— steel tubular, 1104 (II)
Scale in boilers, 48 (II)
Scales for surveys, 254 (I)
——— hardness, 1094 (I)
——— hydrometer, 125 (I)
——— thermometer, 49 (I)
Scarfing beams, 311 (I)
Scavenging and supercharging internal combustion engines, 245 (II)
Scimemi's formula for flow in pipes, 758 (I)
Sclerometer, 1090 (I)
——— hardness values of metals, 1190 (I)
Scotch pig iron, 1142 (I)
Screening caps, effect of, 970 (II)
——— reduction in inductance due to, 971 (II)
Screens, circulating water, 1263 (II)
——— jig, 912 (II)
——— sewage, 783 (II)
Screw, force by, 20 (I)
———heads, standard, 93 (I)
piles, 443 (I)
——— stays for combustion chambers, 35 (II)
——— threads, Acme, 93 (I)
——— ——— American, 89, 94 (I)
——— ——— brass, 92 (I)
——— ——— British Association, 90 (I), 93 (I)
——— ——— British standard Whitworth, 89 (I), 91 (I)
——— ——— Cycle Engineers' Institute standard, 93 (I)
——— ——— fine, British standard, 92 (I)
——— ——— forms of, 89 (I)
——— ——— French standard, 95 (I)
——— ——— International or metrical, 93 (I)
——— ——— on water and gas pipes, 97 (I)
——— ——— Sellers, 90 (I)
——— ——— standard, 89 (I)
——— ——— Steinlen, 89 (I)
——— ——— Swiss or Thury, 90 (I), 96 (I)
——— ——— U.S.A. National Coarse, 96 (I)
——— ——— ——— Pipe, 98 (I)
——— ——— watch and instrument, 92 (I)
——— ——— Whitworth, 89 (I), 91 (I)
Screwing machines, Winn, 1162 (II)
Screws, threads of, 89 (I)
——— weight of, 156 (I)
——— wood, 107 (I)
Scrubbers, gas, 627 (II)
'Seatrist' packing rings, 1259 (II)
Sea, distance of objects at, 263 (I)
——— fire prevention at, 715 (II)
——— pulverised coal at, 1342 (I)
——— temperature of the, 54 (I)
——— water, freezing temperature, 50 (I)
Seam welding, 1292 (I)
Seaplanes, 389 (II)
Seasoning timber, 289 (I)
——— air, 291 (I)
——— kiln, 291 (I)
——— shrinkage by, 289 (I)
Secret commissions, 1028 (II)
Secrets, disclosure of, 1028 (II)
Sectional dock, pontoon, 628 (I)
Séger cones, 1136 (I)
Selenium, 1210 (I)
——— rustless steels, 1200 (I)
'Self-Sealing' pipe coupling, 1118 (II)
Self-supporting roofing, 237 (I)
Sellers' bolts and nuts, 99 (I)
——— involute gear teeth, 972 (I)
——— screw threads, 90 (I), 96 (I)
Sennar dam, 694 (I)
Sentinel pyrometers, 1137 (I)
Separators, magnetic, 916 (II)
——— steam, 1203 (II)
——— steam and water, 69 (II)
Servo brake systems, 374 (II)
Setting, boiler, 11 (II)
——— stones, 327 (I)
Settings, boiler, leaky, 12 (II)
Severn estuary, 720 (I)
Sewage, disposal, Royal Commission on, standard of purity for sewage effluents, 781 (II)
——— filters, 787 (II)
——— hourly variation of flow of, 781 (II)
——— land treatment of, 786 (II)
——— Ministry of Health's requirements, 769 (II)
——— pipes, British standard, 114 (I) 115 (I), 116 (I)
——— pump, 'Stereophagus,' 1293 (II)
——— sludge, 785 (II)
——— ——— activated process, 788 (II)
——— tanks, 786 (II)
——— treatment of, 780 (II)
Sewerage systems, 772 (II)
Sewers, 769 (II)
——— brickwork for, 738 (I)
——— circular, 738 (I)
——— discharging capacity of, 769 (II)
——— flushing, 776 (II)
——— storm water, discharge of, 666 (I)
——— ventilation of, 776 (II)
Shaft ball and roller bearings, 851 (I)
——— couplings, 1272 (II)
——— furnaces, 1125 (I)
——— pillars, 872 (II)
——— sinking, mining, 864 (II), 869 (II)
——— ——— by freezing, 445 (I), 869 (II)
Shafting, 889 (I), 906 (I)
——— bearings, 849 (I), 866 (I)
——— bending and turning moments, 893 (I)
——— diameter of, 889 (I)
——— drives, 906 (I)
——— horse-power of, 889 (I)
——— lubrication of, 878 (I)
——— mill, 892 (I)
——— speed of, 892 (I)
——— steamship, 894 (I)
——— stiffness, 892 (I)
Shafts, 453 (I), 889 (I)
——— chimney stability of, 464 (I)
——— coned and keyed, 117 (I)
——— crane, 810 (II)
——— mine, 864 (II)
——— ——— cost of sinking, 865 (II)
——— ——— deep, 872 (II)
——— ——— lining, 866 (II)
——— ——— rectangular, 864 (II)
——— ——— round, 865 (II)
——— needle bearing propeller, 1272 (II)
——— splined, strength of, 117 (I), 362 (II)
——— timbering, 450 (I)
——— turbine, 148 (II)
Shale oil, 1356 (I)

Shaping machines, 1025 (I)
Sharpening steel mining bits, 887 (II)
Shasta dam, 697 (I)
Shearing plates, force required, 1044 (I)
——— stresses, 216 (I), 404 (I)
Sheathed explosives, 863 (II), 951 (II)
Sheet aluminium, weight of, 135 (I)
—— brass, weight of, 131 (I)
—— cast iron, 127 (I)
—— copper, weight of, 131 (I)
—— iron, galvanised corrugated, 236 (I)
———— weight of, 239 (I)
—— lead, weight of, 132 (I)
—— metal, gauges 85 (I)
—— metals, specification of, 129 (I)
————— weights of, 127 (I), 132 (I)
—— steel corrugated, 238 (I)
———— piling, 444 (I)
—— timber, piling, 445 (I)
—— zinc, trade gauge, 13 (I)
———— weight of, 132 (I)
Shell fillings, 952 (II)
Shells, boiler, 32 (II), 294 (II)
————— strength of, 32 (II)
Sherardising iron, 1195 (I)
Shing Mun dam, 694 (I)
Shipbuilding cranes, 808 (II)
Ship canals, 599 (I)
Shipping ropes, 935 (I), 942 (I)
Ships, Admiralty coefficients, 281 (II)
—— auxiliary machinery, 300 (II)
—— boilers (*see* Boilers, marine)
—— centre of buoyancy, 273 (II)
————— flotation, 274 (II)
—— coefficients of displacement, 272 (II)
—— cold storage on, 743 (II)
—— deadweight, 279 (II)
—— dimensions, 271 (II)
—————— of notable, 284 (II)
—— displacement, 272 (II)
—— draft, 271 (II)
—— electric propulsion of, 292 (II), 1528 (I)
————— transmission of power, 292 (II)
—— electrical equipment of, Institution of Electrical Engineers regulations, 1453 (I)
—— engine and boiler design of, 287 (II)
———— cylinder ratios, 287 (II)
———— oil, 305 (II) (*see* Marine oil engines)
———— piston speed, 288 (II)
———— steam, mean pressures and expansion, 287 (II)
———— turbo-reciprocating machinery, 289 (II)
———— turbines, 290 (II)
—— engines (*see* Marine engines)
—— estimate sheet of, 282 (II)
—— fire prevention on, 715 (II)
—— fuel consumption of, 282 (II), 299 (II)
——— oil for, 298 (II)
—— heating from oil engine exhaust, 308 (II)
—— horse power of, 280 (II)
—— inclining experiment, 275 (II)
—— liners, notable, 284 (II)
—— longitudinal metacentre of, 276 (II)
—— machinery, weight of, 279 (II)
—— motor, 305 (II)
—— moulded dimensions of, 271 (II)
—— overall dimensions of, 271 (II)
—— propulsion of, 283 (II)
—————— electric, 292 (II), 1528 (I)
Ships, propulsion of, electric, transmission, 292 (II)
—————— oil electric, 292 (II)
——————— engine, 305 (II)
—————— reduction gear for, 290 (II)
—— pulverised fuel on, 1342 (I)
—— recipro-turbine machinery, 289 (II)
—— refrigerating machinery for, 743 (II)
—— registered dimensions of, 271 (II)
—— resistance of, 279 (II)
—— rivets, 165 (I)
—— rolling period of, 275 (II)
—— rudders, 282 (II)
—— shafting for, 894 (I)
—— skin effective horse-power of, 280 (II)
—— stability of, 275 (II)
—— steam pressures, 287 (II)
———— turbines, 290 (II)
—————— gearing, 290 (II)
—— steel for building, 164 (I)
—— transverse metacentre of, 274 (II)
—— wave effective horse-power, 280 (II)
—— weights of, 277 (II)
————— machinery of, 279 (II)
—— wetted surface of, 274 (II)
Shore scleroscope, 1089 (I)
Shoring buildings, 450 (I)
Shorter flame-hardening process, 1189 (I)
Shouldered tenon, 313 (I)
Shovels, mechanical, 423 (I)
Shrinkage of castings, 1263 (I)
———— excavated materials, 425 (I)
———— timber, 289 (I)
Shunt gas meter, 200 (I)
Sidereal time intervals, 268 (I)
Siebe, Gorman & Co., Ltd., diving, breathing and resuscitating apparatus, 1298 (II)
Signalling, electric, in coal mines, 908 (II)
———— wireless, at sea, 656 (I)
Signals, railway, 595 (II)
————— colour light, 606 (II)
——— sea, audible, 653 (I)
———— distance audibility of, 656 (I)
———— explosive, 656 (I)
——— submarine sound, 656 (I)
Silencers, engine, 80 (I)
Silencing equipment, ventilation, 683 (II)
Silica bricks, 373 (I)
Silicate cotton or slag wool, 734 (II)
Silico-manganese steels, 1167 (I)
Silicon aluminium alloys, 1231 (I)
—— bronze, 1225 (I)
—— copper and, 1227 (I)
—— influence on cast iron, 1146 (I)
—— monel metal, 1221 (I)
Sills, ground, 582 (I)
' Silmalec,'' 1235 (I)
Silos, reinforced concrete, 406 (I)
Silting of reservoirs, 713 (I)
—— rivers and canals, 714 (I)
Silver, 1210 (I)
—— and gold tables, 932 (II)
—— nickel-, 1218 (I)
—— solder, 1258 (I), 1179 (II)
—— wire, gauge for, 86 (I)
Simple pendulum, 23 (I)
Simplex patent boiler tube cleaner, 1256 (II)
—— salinometer, 1256 (II)
—— Terazzo floor polisher, 1100 (II)
—— valve re-seater, 1255 (II)

Simson's rule for area computation, 251 (I), 421 (I)
Singapore floating dock, 628 (I)
Single-arch dams, 697 (I)
——— span roof, 303 (I)
Sines, tangents, etc., tables, 1049 (II)
——— logarithmic, 1055 (II)
Sinking cylinders, pneumatic process, 441 (I)
——— foundations by freezing process, 445 (I)
——— shaft, mining, 864 (II), 869 (II)
Sinks and water-closets, 797 (II)
'Sinuflo' Cochran boilers, 1206 (II)
——— exhaust gas boilers, 1208 (II)
——— gas fired boilers, 1214 (II)
——— waste heat boiler, 1211 (II)
Siphon weirs and spillways, 694, (I) 751 (I)
Sirens, 656 (I)
Skefko Ball Bearing Co., Ltd., ball and roller bearings, 1146 (II)
Skin effective horse-power of ships, 280 (II)
Slabs, floor, reinforced concrete, 319 (I), 393 (I)
——— partition, 309 (I)
Slag concrete, 343 (I)
Slates, asbestos-cement, 299 (I)
——— sizes of, 297 (I)
——— underlining, 422 (I)
——— weight of, 298 (I)
Slating, 297 (I)
——— clips, 298 (I)
——— nails, 298 (I)
Sleepers, concrete, 488 (II)
——— packing of, 486 (II)
——— steel, 482 (II)
——— wooden, 481 (II)
——— length of, 482 (II)
Sleeve valve aero engines, 462 (II)
Slide valve gear, 94 (II)
Slings, 172 (I)
——— strength of, 173 (I)
Slopes, angles and lengths of, 434 (I)
——— chaining on, 253 (I)
——— of excavations and embankments, 433 (I)
——— side, form of, 433 (I)
Sludge, activated, process, 788 (II)
——— sewage, 785 (II)
Sluice gates, 675 (I), 793 (I), 845 (I)
——— measurement, 675 (I)
Sluices, 675 (I), 793 (I)
——— dam, dimensions, 693 (I)
——— mining, 933 (II)
——— grade and duty of, 933 (II)
Smith alloy No. 10, 1220 (I)
Smith, Angus, composition, 994 (II)
Smoke and coal combustion, 1334 (I)
——— boxes, locomotive, 546 (II)
——— charts, 1335 (I)
——— colour of, 1335 (I)
——— helmets, Siebe, Gorman, 1298 (II)
Snercold petrol air gas plants, 654 (II)
Snow on roofs, weight of, 305 (I)
Soap rosin oil, 996 (II)
——— test for hardness of water, 49 (II)
Socket pipes and fittings, 112 (I), 146 (I), 148 (I)
Socketing wire ropes, 947 (I)
Soda, lime and, water softening, process, 49 (II)
Sodium lamps, 705 (II)
Soft soldering, 1260 (I), 1178 (II)
Softening water, 731 (I), 49 (II), 52 (II)
Soil, electric heating of, 1425 (I)
——— evaporation of water from, 706 (I)
——— pipes, 792 (II)
Soils, angles of repose of, 435 (I)
——— foundations on different, 436 (I)
——— resistance of, in ploughing, 38 (I)
——— weight of cubic yard of, 432 (I)
Solder, 1257 (I), 1175 (II)
——— aluminium, 1259 (I)
——— bismuth, 1259 (I)
——— brazing, 1258 (I), 1179 (II)
——— cold, 1260 (I)
——— creams, 1177 (II)
——— Fry's, 1175 (II)
——— fusible, 1178 (II)
——— hard, 1258 (I), 1179 (II)
——— paint, 1177 (II)
——— plumbers', 1176 (II)
——— silver, 1258 (I)
——— spelter, 1258 (I)
——— tin lead, 1259 (I), 1179 (II)
Soldered joints, strength of, 177 (I)
Soldering, 1257 (I)
——— aluminium, 1259 (I), 1177 (II)
——— cast iron, 1260 (I), 1177 (II)
——— fluxes, 1257 (I), 1176 (II)
——— reaction, 1260 (I)
——— soft, 1260 (I), 1178 (II)
Solenoids, 814 (II), 971 (II)
Solid figures, centre of gravity of, 17 (I)
——— case-hardening materials, 1187 (I)
——— fuels, 1315 (I)
Solids, specific gravity of, 122 (I)
——— velocity of sound through, 70 (I)
'Solution' heat treatment of alloys, 1231 (I)
Solutions, boiling-points of, 32 (I)
Soot blowers, 1234 (II)
Sound, 69 (I)
——— abatement, 1097 (II)
——— absorption, 73 (I)
——— Decibel scale, 71 (I)
——— distance of transmission of, 70 (I)
——— insulation, 73 (I), 75 (I), 310 (I) 1097 (II)
——— limits of audibility, 70 (I)
——— loudness, 70 (I)
——— pitch, 70 (I)
——— proof floors, 307 (I)
——— partitions, 309 (I)
——— reflection, 72 (I)
——— signals, submarine, 654 (I)
——— tone equality, 72 (I)
——— velocity of, 70 (I)
——— waves, 70 (I)
Soundings, surveying, 256 (I)
——— to reduce, to low water, 257 (I)
South, to determine, by watch, 261 (I)
South Africa, irrigation in, 714 (I)
Southampton floating dock, 627 (I)
Southern Railway carriages, 572 (II)
——— locomotives, 518 (II), 520 (II)
——— rolling stock for electrified lines, 576 (II)
——— wagons, 574 (II)
Spanners, 105 (I)
——— box, 105 (I)
——— British Standard, 105 (I)
Spark plugs, 462 (II)

Sparklets Ltd., pressure die castings, 1179 (II)
—————— Woven wire braiding, 1180 (II)
Specific gravities and weights, 119 (I)
—————————— table of, 122 (I)
—————— of chemical elements, 120 (I)
—————— earths, 125 (I)
—————— fluids, 122 (I)
—————— gases, 56 (I), 120 (I), 125 (I)
—————— granular substances, 124 (I)
—————— liquids, 120 (I)
—————— metals, 121 (I)
—————— minerals, 120 (I), 860 (II)
—————— paints, 990 (II)
—————— solids, 122 (I)
—————— stones, 122 (I)
—————— water, 119 (I)
—————— woods, 123 (I)
——— gravity, 119 (I)
——— heat, 61 (I)
————— of gases, 61 (I)
————— steam, 4 (II)
Specification, Admiralty, oil fuel, 1361 (I)
————— British Standard (*see* British Standard)
————— for reinforced concrete work, 414 (I)
Spectrum, decomposition of light in, 64 (I)
Speed gear, variable, 1110 (II)
Speedways, 556 (I)
Spelter solder, 1258 (I)
Spence's metal, 1248 (I)
Spherical excess, 256 (I)
Spikes and nails in wood, holding power of, 179 (I)
Spillway dams, 691 (I)
Spillways, automatic syphon, 694 (I), 751 (I)
——— bellmouth, 694 (I)
Spiral bevel gears, 989 (I)
——— or helical springs, 897 (I)
'Spirelmo' smoke helmet, 1298 (II)
Spirit varnishes, 985 (II)
Splicing driving ropes, 927 (I)
——— wire ropes, 955 (I)
Splined shafts, strength of, 362 (II)
Spontaneous combustion in mines, 907 (II)
————— ignition, temperatures of, 1363 (I)
Spot welding, 1291 (I), 417 (II)
Spray painting, 978 (II), 1296 (II)
——— ponds, 181 (II), 1264 (II)
'Sprayblast' oil cooler, 1265 (II)
Spraying, lime, 1297 (II)
——— metal, 1082, (I) 1194 (I)
Spring damping devices, 78 (I)
——— driving belts, 920 (I)
——— dynamometer, 34 (I)
Springs, 895 (I)
——— buffer, 578 (II)
——— clock type, 897 (I)
——— conical round, 896 (I)
——— cylindrical helical, 895 (I)
——— fatigue of, 186 (I)
——— heat treatment of, 187 (I)
——— helical 896 (I), 898 (I), 334 (II), 369 (II)
————— safe loads and stiffness, 898 (I)
——— laminated, 900 (I), 367 (II), 551 (II)
——— locomotive, 551 (II)
——— motor car, 367 (II)
————— valve, 333 (II)
Springs, railway carriages and wagons, 551 (II), 578 (II)
——— spiral power, 897 (I)
——— steel helical, round, constants for, 334 (II)
————— safe loads and stiffness, 898 (I)
——— steels for, 368 (II), 552 (II)
——— valve, 223 (II), 333 (II)
——— volute and helical, for locomotives, 553 (II)
Spur gearing (*see* Gears)
Squares and circles, area of, 1046 (II)
——— cubes, roots, reciprocals, 1062 (II)
——— geometrical and inertial properties of, 31 (I)
Stability of ships, 275 (II)
Staffordshire blue bricks, 364 (I)
Stainers, 979 (II)
Stainless steel, 1196 (I)
————— annealing, 1198 (I)
————— Austenitic, 1200 (I)
————— cold-rolling and drawing, 1198 (I)
————— etching, 1199 (I)
————— grinding, 1199 (I)
————— hardening, 1196 (I)
————— hardness of, at elevated temperatures, 1199 (I)
————— hot rolling and forging, 1198 (I)
————— normalising, 1198 (I)
————— pickling, 1198 (I)
————— properties, 1197 (I)
————— tests of tempered bars, 1199 (I)
————— welding, 1200 (I)
Stains, 994 (II)
Stairs, 314 (I)
Stamps, drop, 1120 (I), 1166 (II)
——— gold milling, 910 (II), 929 (II)
Standard (*see also* British Standards)
——— atmosphere, 412 (II)
Star, Pole, to find, 259 (I)
Stars, circumpolar, 267 (I)
——— identification of, 263 (I)
Starters, electric motor, 1497 (I)
Starting internal combustion engines, 207 (II)
——— steam turbines, 159 (II)
Static balancers, 1502 (I)
——— bending of timber, 293 (I)
——— condensers, 1505 (I)
Statics, 20 (I)
Statistics, highway, 544 (I)
Status, legal, of engineers, 1027 (II)
Staving process, 790 (I)
Stays for combustion chambers, 35 (II), 40 (II)
Steam, 3 (II)
——— accumulators, 776 (I), 183 (II)
——— adiabatic heat drop, 115 (II)
——— boilers (*see* Boilers)
——— breakdown cranes, 806 (II),
——— buildings warmed by, 670 (II)
——— condensed, measurement of, 181 (II)
——— condensers, 176 (II), 1261 (II)
——— consumption, 76 (II), 78 (II), 115 (II)
—————— theoretical, 76 (II)
—————— typical, 78 (II)
——— conversion tables, 3 (II)
——— cranes, 806 (II)
——— de-superheaters, 1249 (II)
——— diagram factor, 75 (II)
——— engines, 73 (II)

STE

Steam engines, back pressure and extraction, 82 (II)
——— bearing pressures, 85 (II)
——— bearings, 871 (II)
——— crank connecting-rod mechanism, 87 (II)
——— ——— effort, and turning moment, 89 (II)
——— cylinders, 84 (II), 87 (II), 289 (II)
——— design of, 84 (II), 85 (II)
——— diagram factors, 75 (II)
——— duty of, 79 (II)
——— efficiency of, Rankine cycle, 1301 (I), 76 (II)
——— ——— ratio of, 76 (II)
——— expansion of steam in, 73 (II), 75 (II)
——— extraction, 82 (II)
——— flywheels, 90 (II)
——— frames, 86 (II)
——— governor of, 93 (II)
——— high pressure working economy, 76 (II)
——— horse-power of, 75 (II)
——— indicator for, 81 (II), 1095 (II)
——— mechanical efficiency of, 75 (II)
——— passage and valve areas, 87 (II)
——— piston speeds, 84 (II), 288 (II)
——— pressures and expansion of steam, 287 (II)
——— reciprocating marine, cylinder ratios, 287 (II)
——— relay governors, 93 (II)
——— slide valve gear, 94 (II)
——— steam consumption, theoretical, 76 (II)
——— ——— jackets, 77 (II)
——— stresses in, 85 (II)
——— two crank, starting torque, 89 (II)
——— types of, 81 (II)
——— valve gears, 94 (II)
——— exhaust, heating by, 673 (II)
——— ——— oil in, 84 (II)
——— expansion of, 73 (II), 287 (II)
——— fittings, approximate dimensions of, 112 (I)
——— flow of, in turbines, 103 (II)
——— ——— through orifices, 6 (II), 107 (II)
——— ——— ——— pipes, 9 (II), 1201 (II)
——— generating plant, 9 (II), 1201 (II)
——— hammers, 1117 (I), 1166 (II)
——— ——— energy and velocity of a blow, 1117 (I)
——— ——— formula for, 1117 (I)
——— ——— sizes of, 1117 (I)
——— ——— steam consumption of, 1118 (I)
——— heating of buildings, 670 (II)
——— ——— railway carriages, 578 (II)
——— high pressure, 76 (II)
——— jackets, 77 (II)
——— jets, 1361 (I)
——— meters, 202 (I)
——— ——— mercury U-tube, 202 (I)
——— ——— orifice, 202 (I)
——— ——— rotary, 204 (I)

STE

Steam Mollier diagram, 1304 (I)
——— nozzles, 107 (II)
——— oil in exhaust, 84 (II)
——— passages and valves, area of, 87 (II)
——— pipe flanges, 109 (I)
——— ——— hangers, 63 (II)
——— pipes, 112 (I), 145 (I), 60 (II)
——— ——— bends for, 61 (II)
——— ——— copper, 62 (II)
——— ——— coverings for, 68 (II)
——— ——— exhaust, 63 (II)
——— ——— flange losses, 68 (II)
——— ——— flow of steam in, 64 (II), 66 (II)
——— ——— high pressure and high temperature, 61 (II)
——— ——— joints of, 63 (II)
——— ——— lagging, 69 (II)
——— ——— Lloyd's rules for, 62 (II)
——— ——— loss of heat from, 67 (II)
——— ——— low pressure, 671 (II)
——— ——— pressure drop in, 64 (II)
——— ——— size of, 112 (I), 145 (I), 63 (II)
——— ——— steel bends, 61 (II)
——— ——— ——— corrugated, 61 (II)
——— ——— thickness of, 146 (I)
——— ——— weight of, 145 (I)
——— plants, high pressure, 10 (II)
——— power stations, efficiency of, 7 (II)
——— pressure and temperature of, 3 (II)
——— ——— conversion tables, 3 (II)
——— ——— in cylinders of compound and triple-expansion engines, 288 (II)
——— ——— mean effective, 80 (II)
——— pressures corresponding to temperatures, 3 (II)
——— properties of saturated, 3 (II)
——— ——— superheated, 5 (II)
——— pumps, 831 (I), 1121 (II), 1262 (II)
——— railway motor cars, 565 (II)
——— required to heat water, 6 (II)
——— resuperheating, 139 (II)
——— saturated, properties of, 176 (II)
——— saving by superheating, 76 (II)
——— separators, 69 (II), 1203 (II)
——— specific heat of, 4 (II)
——— superheated, 4 (II), 76 (II)
——— ——— heat of, 4 (II)
——— ——— savings by, 76 (II)
——— superheaters, 16 (II), 296 (II), 547 (II), 1229 (II), 1278 (II)
——— tables, 3 (II)
——— throttling calorimeter, 6 (II)
——— traps, 69 (II), 1253 (II)
——— ——— Bradford, 1254 (II)
——— ——— Newman Carter, 1253 (II)
——— ——— practice, notes on, 70 (II)
——— turbines (*see* Turbines, steam)
——— units and conversion tables, 3 (II)
——— valves, 1249 (II)
Steamships (*see* Ships)
Steel, 1158 (I)
——— Admiralty, rests for, 164 (I)
——— age-hardening, 1179 (I)
——— aircraft, 416 (II)
——— alloy specifications, 1161 (I)
——— analysis of various, 1181 (I)
——— angle and tee, weight of, 137 (I)
——— annealing, 1168 (I), 1192 (I)
——— austenitic stainless, 1200 (I)
——— bars, forge test for, 164 (I)

STE

Steel, bars, tests for, 166 (I)
—— beams, British standard steel, 242 (I)
—— —— working stresses, 222 (I)
—— bits, mining, sharpening, 887 (II)
—— boiler, 163 (I), 32 (II)
—— —— stays, 35 (II)
—— bridgework, 163 (I), 477 (I)
—— British standard for bridges and buildings, 163 (I)
—— carbon, 1063 (I), 1158 (I), 1161 (I)
—— —— tools, 1063 (I), 1074 (I)
—— case-hardening, 1187 (I)
—— castings for locomotives, 554 (II)
—— chimneys, 466 (I)
—— chisel, 1181 (I)
—— chrome vanadium, 1167 (I)
—— chromium, 1167 (I)
—— —— in, effect of, 1183 (I)
—— coefficient of expansion at high temperatures, 1184 (I)
—— columns and struts, 227 (I)
—— conductor rails, 478 (II)
—— conduits, 1418 (I)
—— —— dimensions of, 1419 (I)
—— copper, 1202 (I)
—— cored aluminium cables, 1405 (I), 1410 (I)
—— corrosion of, 178 (I), 1192 (I)
—— creep at high temperatures, 155 (II)
—— cutting by flame, 1284 (I)
—— die, non-shrinking, 1185 (I)
—— electro-galvanising, 1193 (I)
—— expansion of, at high temperatures, 1184 (I)
—— fishplates, 479 (II)
—— flat, weight of, 131 (I)
—— forgings, 151 (II)
—— —— locomotive, 554 (II)
—— free-cutting, 1162 (I)
—— galvanising, 1192 (I)
—— girders, rolled, 239 (I)
—— —— bending moments, shearing stresses, etc., 216 (I), 225 (I)
—— grain growth and size control, 1168 (I)
—— hardening and tempering, 1168 (I), 1187 (I)
—— heat resisting, 1201 (I)
—— —— treated, test results of, 1180 (I)
—— —— treatment of, 1167 (I)
—— —— —— oil coolers for the, 1174 (II)
—— helical springs, 898 (I), 334 (II)
—— high speed tool, 1067 (I), 1160 (I)
—— —— —— hardening, 1069 (I), 1160 (I)
—— —— temperatures, effect of, on, 153 (II)
—— Invar, 1183 (I)
—— lead-bearing, 1163 (I)
—— magnet, 1182 (I)
—— manganese, 1185 (I)
—— mild treatment for, 166 (I)
—— —— normalising, 167 (I)
—— mass effect, 1170 (I)
—— molybdenum, 1163 (I), 1184 (I)
—— —— effect in, 1184 (I)
—— —— manganese alloy, 1185 (I)
—— nickel, 1164 (I), 1183 (I)
—— —— chrome, 1164 (I)
—— —— effect in, 1183 (I)
—— —— expansion of, 1184 (I)
—— 'Nitralloy,' 1189 (I)

STE

Steel, non-metallic inclusions in, 1192 (I)
—— normalising, 167 (I), 1167 (I)
—— octagon, weight of, 129 (I)
—— open hearth, 472 (II)
—— piling, 444 (I)
—— pillars, 227 (I), 327 (I)
—— pipes, design of, 723 (I)
—— —— for hydraulic power, 787 (I)
—— —— locomotive boilers, 545 (II), 556 (II)
—— —— spigot and socket, 116 (I)
—— —— water, incrustation of, 734 (I)
—— plates, detection of fine cracks in, 1407 (I)
—— —— for locomotive boilers, 557 (II)
—— precipitation hardening, 1179 (I)
—— props 1104 (II)
—— quenching liquids for, 1179 (I)
—— railway rails, 470 (II)
—— —— special, 477 (II)
—— rapid machining, 1162 (I)
—— refining, 1168 (I)
—— rivets, 165 (I)
—— rod reinforcing bars, 398 (I)
—— rolled, 239 (I)
—— —— flat, weight of, 131 (I)
—— —— limiting dimensions of, 239 (I)
—— roofs, 233 (I)
—— ropes, 176 (I)
—— round, weight of, 129 (I)
—— rustless, 1200 (I)
—— rust proof coating for, 1192 (I)
—— scaffolding, 1104 (II)
—— sections, rolled, British standard, 239 (I)
—— —— limiting dimensions of, 239 (I)
—— selenium rustless, 1210 (I)
—— sheet piling, 444 (I)
—— sheets, corrugated, 238 (I)
—— shipbuilding, 164 (I)
—— silico manganese, 1167 (I)
—— sleepers, 482 (II)
—— specific gravity, 121 (I)
—— spring, 368 (II), 551 (II)
—— springs, 898 (I)
—— —— constants for round wire, 334 (II)
—— square, weight of, 129 (I)
—— stainless, 1196 (I) (*see* Stainless steel)
—— stanchions, 227 (I)
—— steam pipe bends, 61 (II)
—— strain-ageing, 1179 (I)
—— strength and elasticity of, 159 (I)
—— —— at high temperatures, 153 (II)
—— stress due to temperature, 177 (I)
—— strip, hardness of cold rolled, 1191 (I)
—— structural, 163 (I), 227 (I), 496 (I)
—— —— L.C.C. rules for use of, 322 (I)
—— —— working stresses in, 495 (I)
—— tee and angle, weight of, 137 (I)
—— tempered, colours of, 1072 (I)
—— tempering and hardening, 1069 (I), 1168 (I)
—— tests for, Admiralty, 164 (I), 166 (I)
—— tool, carbon, 1063 (I)
—— —— high speed, 1074 (I), 1160 (I)
—— —— —— hardening, 1069 (I), 1160 (I)
—— —— lathe, cutting speeds, 1074 (I)
—— troughing, 506 (I)

Steel, tubes, collapsing, strength of, 169 (I)
— — for locomotive boilers, 42 (II), 556 (II)
— — heavy weight, dimensions and weights of, 111 (I)
— — light weight, dimensions and weights of, 111 (I)
— — weight of weldless, 140 (I)
— vanadium, 1167 (I)
— wear of metals on, 1191 (I)
— weight of, 111 (I), 128 (I), 137 (I)
— welding, 1278 (I)
— white metal, molten effect on, 1203 (I)
— wire, weight of, 131 (I), 154 (I)
— — gauze, 87 (I)
— — ropes (*see* Wire Ropes)
— — springs, constants for round, 334 (II)
— — strength, 162 (I)
Steelwork for buildings, 322 (I)
— paints, 984 (II)
Steering gear, automobile, 371 (II)
'Stefco Wildish' steam separator, 1203 (II)
Steinlen screw thread, 89 (I)
Stemming, 863 (II)
Stephenson's link motion valve gear, 98 (II)
'Stereophagus' pump, 1293 (II)
Sterilisation of water, 733 (I)
Stevenson's Creek dam, California, 693 (I)
'Stirling' through bolt header, 1278 (II)
Stokers, automatic 'Iron Fireman,' 1228 (II)
— — Robot, 1226 (II)
— mechanical, 1336 (I), 1220 (II)
— — Bennis air-draught, 1220 (II)
— — Crosthwaite, 1221 (II)
— — International Combustion, Ltd., 1222 (II)
— — locomotive, 562 (II)
— — Riley Stoker Co., Ltd., 1224 (II)
Stone, artificial, 358 (I)
— coefficients of friction, 21 (I)
— Ransome's, 358 (I)
— road, 553 (I)
— stairs, 315 (I)
— Victoria, 358 (I)
— walls, safe load on, 436 (I)
— weather effect on, 334 (I)
— weight of, 333 (I)
Stones, for masonry, 327 (I)
— lifting, 327 (I)
— road, tests of, 548 (I)
— setting, 327 (I)
— specific gravity of, 122 (I)
— weight of, 333 (I)
Stoneware, 370 (I)
Stop valves, 779 (I)
Storage bins, reinforced concrete, 406 (I)
— capacity of gas holders, 630 (II)
— coal, 1319 (I), 632 (II)
— — loss of thermal value by, 1320 (I)
— coke, 633 (II)
— cold, 731 (II)
— gas, 630 (II)
— paint, 977 (II)
— reservoirs, 582 (I), 720 (I)
Storm-water discharge of sewers, 666 (I), 782 (II)
Stoving Japans, 988 (II)
Strain-ageing of steel, 1179 (I)
Strain gauge dynamometer, 39 (I)
Strainer racks, 722 (I), 845 (I)
Strainers and foot valves, 803
Straining beams, joints of, 313 (I)
Streams, fire, 713 (II)
— measuring quantity of water flowing in, 674 (I)
— — velocity of water in, 679 (I)
Strength, ashesive, of mortars, 181 (I)
— collapsing of steel tubes, 169 (I)
— crushing, of bricks, 180 (I), 365 (I)
— fatigue, of wires and springs, 186 (I)
— of animals, 35 (I), 37 (I)
— — beams and poles, 222 (I)
— — bricks, 180 (I), 365 (I)
— — brickwork, 180 (I)
— — building materials, 180 (I)
— — cement, 180 (I)
— — chains, 169 (I)
— — columns, 229 (I), 396 (I)
— — concrete, 180 (I), 347 (I)
— — cylinders, thick hollow, 168 (I)
— — double helical gears, 994 (I)
— — glue, 180 (I)
— — iron chains, 169 (I)
— — lead pipes, 168 (I)
— — materials, 159 (I)
— — — transverse, 159 (I)
— — men, 37 (I)
— — metals, 159 (I), 1242 (I)
— — — at various temperatures, 160 (I), 1186 (I)
— — mortars, adhesive, 181 (I)
— — poles, 296 (I)
— — rings, 171 (I)
— — rivets, 232 (I)
— — ropes, 176 (I)
— — slings, 172 (I)
— — soldered joints, 177 (I)
— — steel, 160 (I)
— — — tubes, 169 (I)
— — swivels, 172 (I)
— — timbers, 179 (I), 293 (I)
— — tubes, 168 (I)
— — welds, 1278 (I)
— — wire, 162 (I)
— — woods, 179 (I), 293 (I)
— tensile, of Bull's metal, 1174 (II)
— — materials, 159 (I)
— — metals and alloys, 159 (I)
— transverse, of materials, 159 (I)
— — metals, 162 (I)
Stress, breaking, concentrations, 187 (I)
— — wire and wire ropes, 162 (I)
— concentrations, 187 (I)
— due to temperature, 177 (I)
Stresses in arches, 410 (I), 533 (I)
— on beams and girders, 211 (I), 216 (I), 222 (I)
— — reinforced concrete, 404 (I)
— — dock gates, 619 (I)
— — — walls, 615 (I)
— — engines, 85 (II)
— — safe working on structures, 222 (I), 229 (I)
— — turbines, 145 (II)
— — walls and piers, 321 (I)
— — working, on bridges, 493 (I), 517 (I)
— — — reinforced concrete, 404 (I)

Strip metal forming machines, 1160 (II)
Stromeyer testing machine, 183 (I)
Structural vibration and noise, 76 (I)
——— steel, working stresses in, 222 (I), 227 (I), 495 (I)
——— steelwork for buildings, 163 (I), 322 (I)
Structures, crane, 809 (II)
——— reinforced concrete, 399 (I)
——— safe working stress, 222 (I), 229 (I)
Struts, 227 (I)
——— aeroplane, 421 (II)
——— between houses, 450 (I)
Strutting, 314 (I)
Stubs' iron and steel wire gauges, 87 (I)
Sturtevant Engineering Co., Ltd., electrostatic precipitation, 1236 (II)
Subaqueous rock removal, 641 (I)
Submarine bells, 656 (I)
Subsidences, mining, 875 (II)
Subtense measurements, tacheometer, 258 (I)
Suburban railways, electric traction, 590 (II)
Suction cutter dredgers, 636 (I)
——— dredgers, 635 (I)
——— pump, 801 (I)
——— piping, 801 (I)
Suez Canal, 600 (I)
Sulphate of ammonia, 644 (II)
Sulphur influence on cast iron, 1147 (I)
——— removal of, from flue gases, 48 (II)
Sulphuretted hydrogen removal from gas, 637 (II)
Sundial, horizontal, 272 (I)
——— time, variation from clock time, 261 (I)
Sun, the, 263 (I)
——— data regarding the earth and, 263 (I)
——— to find variation of compass by single altitude of, 259 (I)
Sun's parallax in altitude, 265 (I)
——— radius and distance from earth, 263 (I)
Supercharging, 245 (II)
——— aero engines, 457 (II)
——— automobile engines, 347 (II)
——— internal combustion engines, 245 (II)
——— marine oil engines, 312 (II)
Superevelation of outer rail on curves, 500 (II)
Superheated steam, 4 (II), 76 (II)
——— degree of, 77 (II)
——— economy of, 76 (II)
——— heat of, 4 (II)
Superheater Co.'s superheaters, 1299 (II), 1278 (II),
——— header, multiple valve regulator 1280 (II)
Superheaters, 16 (II), 296 (II), 547 (II), 1229 (II), 1278 (II)
——— for cylindrical boilers, 296 (II)
——— locomotive, 547 (II), 1278 (II)
——— MeLeSo, 1229 (II), 1278 (II)
——— tubes for water tube boilers and, 43 (II)
Superheating, steam saving by, 76 (II)
Supports, pipe, 723 (I)
Surcharged wall, 457 (I)
Surface condensers, 168 (II), 1261 (II)
——— heat transmission in, 170 (II)
——— grinding, 1110 (I)
——— hardening (Shorter process), 1189 (I)
Surface measures, 1 (I)
——— wetted, of ships, 274 (II)
Surfaces, reflecting power of, 686 (II)
Surveying, 249 (I)
——— chain, 1 (I), 249 (I)
——— instruments, 257 (I)
——— rate of, 251 (I)
——— soundings, 256 (I)
Surveys, scales for, 254 (I)
Suspension bridges, 537 (I)
——— self-anchored, 539 (I)
——— motor-car, 365 (II)
Swashplate pump, 824 (I)
Swing bridges, 525 (I)
——— lubrication of, 530 (I)
——— pivots, 528 (I)
——— rollers, 527 (I)
——— slewing power required for, 531 (I)
——— some experiments on friction in, 529 (I)
——— stresses on, 527 (I)
Swiss screw thread, 90 (I), 96 (I)
Switch and transformer oil, 1501 (I)
——— boards, 1495 (I)
——— crane, 814 (II)
——— gear, British standard, 1495 (I)
Switches and crossings on railways, 494 (II)
——— pipe-line change-over, 1119 (II)
Swivels, strength of, 172 (I)
Synchronous clocks, 1529 (I)
——— condensers, 1505 (I)
——— motors, 1489 (I)
——— induction type, 1491 (I)
Synthetic resin varnishes, 988 (II)
Syphon weirs, 694 (I), 751 (I)
——— tests of, 694 (I)

T BARS, 245 (I)
— beams, concrete, 402 (I)
— joint, 312 (I)
Tacheometer, subtense measurements, 258 (I)
Tal-y-bont dam, 695 (I)
Tampers, mechanical, 491 (II)
Tangents, sines, etc. tables, 1049 (II)
——— logarithmic, 1055 (II)
Tanks, gasholder, 630 (II)
——— pressure, 889 (II)
——— sewage, 783 (II)
Tantalum, 1210 (I)
Taper pins, 117 (I)
——— roller bearings, 853 (I), 1131 (II)
Tar, 1323 (I)
——— distillates, 1356 (I)
——— distillation, 641 (II)
——— extraction, 627 (II)
——— fuels, 1358 (I)
——— lignite, 1358 (I)
——— macadam, 564 (I), 570 (I)
——— oil engines, 1358 (I)
——— fuel for Diesel engines, 1356 (I), 1358 (I)
——— properties of, 1356 (I)
Tarring roads, 567 (I)
Tecalemit, Ltd., mechanical lubrication, 1150 (II)
Tee iron, weight of, 136 (I)
— steel, weight of, 137 (I)
Teeth (*see* Gears and Gearing)

Television, 960 (II)
——— details of system, 960 (II)
——— interlacing, 961 (II)
——— programmes, 961 (II)
——— service extension of, 961 (II)
Tellurium, 1211 (I)
Temperature, absolute zero of, 50 (I)
——— colour scales, 50 (I)
——— control of furnaces, 1135 (I)
——— conversion factors, 50 (I)
——— ——— tables, 49 (I)
——— effect of, on bridges, 503 (I)
——— ——— on fatigue limit, 185 (I)
——— entropy diagram, 1303 (I)
——— melting points, 50 (I)
——— of atmosphere, 54 (I)
——— ——— earth, 54 (I)
——— ——— lakes and seas, 54 (I)
——— scales, 49 (I)
——— stress due to, 177 (I), 503 (I)
——— tables, 49 (I)
Temperatures, colour of different, 50 (I)
——— contract, 663 (II)
——— flash, 53 (I)
——— freezing, 50 (I)
——— gas works, 623 (II)
——— ignition, 53 (I)
——— spontaneous ignition, 1363 (I)
Tempered steel, colours of, 1072 (I)
Tempering and hardening steel, 1069 (I), 1168 (I), 1187 (I)
——— cutting tools, quenching baths for, 1072 (I)
——— steels, 1069 (I), 1168 (I)
Template, 'Maco,' adjustable, 513 (II)
Tenders, locomotive, 563 (II)
Tenon and mortise, 313 (I)
——— tusk or shouldered, 313 (I)
Tensile strength of cast iron, 1150 (I)
——— ——— materials, 181 (I), 1191 (I)
——— test pieces, 35 (II)
Tensioning devices for belts, 915 (I)
Terminal insulator bushings, 1502 (II)
Terra-cotta, 369 (I)
Terrazzo floor polisher, 1100 (II)
Testing chains, 175 (I)
——— electrical installations, 1449 (I), 1194 (II)
——— instruments, electrical, Evershed & Vignoles Ltd., 1194 (II)
——— machines, fatigue, 182 (I)
——— manilla rope, 177 (I)
——— methods for fatigue in metals, 182 (I)
——— plant, aero-engine, 'Froude' wind tunnel, 1094 (II)
——— Portland cement, 355 (I)
——— water, 731 (I), 1197 (II)
——— welds, 1283 (I)
Tests, Admiralty, for iron, 164 (I)
——— ——— steel, 164 (I)
——— for bricks, 366 (I)
——— ——— concrete, 386 (I)
——— ——— gas, 649 (II)
——— ——— steel bars, Admiralty, 166 (I)
——— ——— ——— rivets, Admiralty, 165 (I)
Theodolite, adjustment of, 257 (I)
——— ranging out railway curves with, 509 (II)
——— the Transit, 264 (I)
Thermal conductivity, 63 (I), 310 (I), 1204 (I)
——— expansion, 62 (I)
Thermal treatment of case-hardened materials, 1188 (I)
——— units, British, 54 (I)
Thermit welding, 1294 (I)
Thermodynamics, 1299 (I)
——— formulæ relating to perfect gases, 1299 (I)
Thermo-electric pyrometers, 1138 (I)
'Thermofeed' regulator, 1247 (II)
Thermometer scale conversion chart, 49 (I)
——— scales, 49 (I)
Thermometers, expansion type, 1136 (I)
Therms, 54 (I)
Thicknesses and weights of tubes, 146 (I)
Thimble splices for wire ropes, 948 (I)
Thread screws, 89 (I)
Thring-Hopkinson torsionmeter, 39 (I)
Throttling calorimeter, 6 (II)
Thrust bearings, 854 (I), 858 (I), 1135 (II)
——— and torque of airscrews, 407 (II)
Thury's screw thread, 90 (I), 96 (I)
Tide power, 719 (I)
Tie-beams of roof, 305 (I)
Tilbury floating landing stages, 644 (I)
Tiles, 299 (I), 367 (I)
——— concrete, 350 (I), 353 (I)
——— paving, 316 (I)
——— roofing, 299 (I), 367 (I)
——— ——— clay or marl, 367 (I)
Tilting furnaces, 1129 (I)
Timber, 179 (I), 277 (I)
——— creosoting, 285 (I)
——— cubic feet in one ton, 283 (I)
——— drying, apparatus for, 291 (I)
——— durability of, 284 (I)
——— elasticity of, 179 (I)
——— fire resistance of, 285 (I)
——— handling appliances, 625 (I)
——— kilns, 291 (I)
——— measures, 277 (I)
——— moisture content, 289 (I)
——— piles, 437 (I)
——— poles, strength of, 296 (I)
——— preservation of, 285 (I)
——— resistance to fungal and other attack, 284 (I)
——— ——— torsional rupture, 179 (I)
——— safe working stress, 295 (I)
——— seasoning, 289 (I), 295 (I)
——— sheet piling, 445 (I)
——— shrinkage of, 289 (I)
——— specific gravity of, 123 (I)
——— steaming, 293 (I)
——— storage, 293 (I)
——— strength and elasticity of, 179 (I), 293 (I)
——— terms, 277 (I)
——— types of, 278 (I)
——— weights of green and seasoned, 283 (I)
——— working stress, safe, 295 (I)
Timbering tranches, 450 (I)
Time, graphic method of finding from altitude of sun or star, 271 (I)
——— equation of, 261 (I)
——— to determine, 266 (I), 271 (I)
Timing observations, 269 (I)
Timken tapered roller bearings, 1131 (II)
Tin, 1205 (I)
——— alloy melting points, 51 (I)
——— antimony alloys, 1243 (I)
——— base alloys, 1243 (I), 1250 (I)

Tin, dredging, 937 (II)
—— lead solders, 1259 (I), 1175 (II)
—— ore, 859 (II)
—— weights, Malay, 938 (II)
Tinplates, weight of, 153 (I)
Tinplating, 1193 (I)
Titanium, 1211 (I)
Tip wagons, 432 (I)
Tocco hardening process, 1189 (I)
Tone control, 969 (II)
Tool steels, carbon, 1063 (I)
———— high speed, 1067 (I), 1160 (I)
———————— hardening, 1069 (I), 1160 (I)
Tools, 1019 (*see also* Machine Tools)
—— carbon steel, 1063 (I)
———————— cutting speeds, 1071 (I)
—— cemented carbide, 1060 (I)
—— chip breakers, 1061 (I)
—— chromium plated, 1161 (I)
—— cutting angles, 1062 (I), 1070 (I)
—— grinding, 1061 (I)
—— machine, 1019 (I)
—— portable pneumatic, 1078 (I)
—— quenching baths for tempering cutting, 1072 (I)
—— roughing, 1073 (I)
—— shape of, 1073 (I)
—— small, for use on machine tools 1044, (I)
Tooth gearing (*see* Gears)
Torque, 38 (I), 39 (I)
—— airscrew, 407 (II)
—— curve, harmonic components of, 239 (II)
—— measurement by dynamometers, 39 (I)
Torsiographs, 244 (II)
Torsion meter, Hopkinson-Thring, 39 (I)
—— meters, 39 (I)
Torsional vibrations of crankshafts, 235 (II)
Towers, water-cooling, 182 (II), 1264 (II)
Town gas in storage, contamination of 641 (II)
Tracing water leakages, 734 (I), 1107 (II)
Track circuiting on railways, 512 (II), 601 (II)
—— laying machines (Bretland's), 491 (II)
—— wagons, 432 (I)
Traction, canal, 597 (I)
—— electric, 381 (II) 585 (II),
———————— Board of Trade regulations for, 382 (II)
———— motors, control of A.C., 593 (II)
———— resistance on a level railroad, 528 (II)
Tractive force of locomotives, 532 (II)
—— resistance on tramways, 382 (II)
Tractor boats on canals, 598 (I)
Trade-marks, 1010 (II)
Train acceleration, 531 (II)
—— control, automatic, 609 (II)
—— high speed, 566 (II)
—— resistance, 526 (II)
Training welders, 1282 (I)
Trains, electric motor coach, 591 (II)
—— momentum of, 500 (I)
—— resistance of air to, 526 (II)
—— table of speed of, 530 (II)
Tramways, electric, 581 (II)
———————— Board of Trade regulations, 382 (II)
———————— brakes for, 382 (II)
———————— braking regenerative, 382 (II)
Tramways, electric, braking, rheostatic, 382 (II)
———————— compared with buses, 384 (II)
———————— tractive resistance on, 382 (II)
Transandine tunnel, 572 (I)
Transformers, electric, 1500 (I), 967 (II)
———————— booster, 1503 (I)
———————— bushings for, 1502 (I)
———————— instrument, 1509 (I)
———————— oil for, 1501 (I)
———————— on the National 'Grid' system, 1502 (I)
Transit theodolite, 264 (I)
Transition curves, 510 (II)
Transitions, setting out, by theodolite, 509 (II)
Transmission lines, 1410 (I)
———— of power, hydraulic, 767 (I), 163 (II)
———————— wave, 772 (I)
Transport, costs and capacities of, 37 (I)
Transporter bridges, 527 (I)
Transporters, quay-side, 623 (I)
Transporting earth, 425 (I)
Transverse metacentre of ships, 274 (II)
———— strength of materials, 159 (I)
———————— metals, 162 (I)
Trapezium, area of, 249 (I)
Trapezoid, area of, 249 (I)
Trapezoidal weirs, 747 (I)
Traps, steam, 69 (II), 1253 (II)
———— general notes on, 69 (II)
Trass mortar, 337 (I)
Trautwine rule for retaining walls, 461 (I)
Travelling cranes, 802 (II)
Tree planting on roads, 570 (I)
Trenches, timbering, 450 (I)
Trent Canal, Canada, 592 (I)
Triangle of forces, 20 (I), 30 (I)
Triangulation, 250 (I)
Trigonometry, 1047 (II)
Triple helical gears, 996 (I)
Trist, Ronald, & Co., Ltd., 'Seatrist' packing rings, 1259 (II)
———————— 'Thermofed' regulator, 1247 (II)
Trolley buses, 374 (II)
———— motors and controls, 384 (II)
Trolhatta Canal, Sweden, 587 (I)
Troughs, locomotive water, 564 (II)
———— steel for bridge flooring, 507 (I)
Troy weight, 6 (I)
'Tru-Lay' wire ropes, 941 (I)
Trucks, industrial electric, 385 (II), 1274 (II)
Trunk pistons, 198 (II)
—— Roads Act, 543 (I)
Tubbing of mines, 867 (II)
Tube, aluminium, sizes and weight of, 152 (I)
—— boiler, 25 (II), 31 (II)
———— charcoal iron lapwelded, 42 (II)
———— flue, 33 (II)
———— locomotive, 42 (II), 545 (II)
———— surface of, 42 (II)
———— transmission of heat through, 25 (II), 31 (II)
———— weight of iron, 41 (II)
—— brass, seamless, weight of, 151 (I)
—— cleaners, 1239 (II)
—— condenser, 169 (II)
———— cleaning, 169 (II)

Tube, copper, weight of, 149 (I)
—— couplings, 1081 (I)
—— cutting-off machines, 1168 (II)
—— fittings, 111 (I)
—— for water-tube boilers and superheaters, 43 (II)
—— locomotive boiler, 42 (II), 545 (II)
—— mills, 93 (II)
—— pitot, 578 (I), 680 (I), 755 (I)
—— pneumatic despatch, 1302 (II)
—— pressure, Pitot, 578 (I), 680 (I), 755 (I)
—— screwed and socketed, dimensions of, 112 (I)
—— —— thicknesses and weights of, 146 (I)
—— steel, cold drawn weldless, 140 (I)
—— —— collapsing strength of, 169 (I)
—— —— heavy weight, dimensions and weights, 111 (I)
—— —— light weight, dimensions and weights, 111 (I)
—— —— scaffolding, 169 (I)
—— —— strength of, 168 (I)
—— superheater, 1147 (I)
—— thin walled metal in compression and torsion, 426 (II)
—— Venturi, 853 (I)
—— wrought iron, weight of, 142 (I)
Tubing, aluminium, weight of, 152 (I)
—— metallic flexible, 1081 (I), 1273 (I)
Tubs, colliery, 891 (II)
Tubular air cooler, 1198 (II)
—— heaters, 28 (II)
Tummel development, 694 (I)
Tungsten, 1211 (I)
—— carbide tipped tools, 1060 (I)
Tunnels, Antwerp, River Scheldt, 572 (I)
—— Dartford-Purfleet, 571 (I)
—— Mersey, 572 (I)
—— Transandine, 572 (I)
—— wind, 390 (II), 1094 (II)
Turbine gears, 996 (I)
—— double helical, 998 (I)
—— head races, 722 (I)
—— pumps, 812 (I), 1125 (II)
—— —— for, boiler feeding, 814 (I), 1125 (II)
—— —— impellers, 816 (I)
—— —— Pulsometer, 1125 (II)
Turbines, 253 (II)
(Gas)
—— aircraft, 265 (II)
—— auxiliary, as, 266 (II)
—— coal burning, 265 (II)
—— combustion gas engines, 254 (II)
—— cooling, 258 (II)
—— fuel, 265 (II)
—— future scope, 267 (II)
—— heat recuperation, 258 (II)
—— installations, 259 (II)
—— jet propulsion unit, 265 (II)
—— locomotives, 264 (II)
—— marine units, 261 (II)
—— regulation methods, 266 (II)
—— reheat, 258 (II)
—— starting, 266 (II)
—— types and operation, 253 (II)
(Steam), 103 (II), 290 (II)
—— back pressure, 141 (II)
—— bearings of, 133 (II)
—— blade vibration, 149 (II)
Turbines.
(Steam), blading of, 114 (II), 124 (II)
—— —— materials, 152 (II)
—— British standard specification for, 159 (II)
—— carry-over energy, 123 (II)
—— combination impulse, 123 (II)
—— consumption of steam in, 115 (II)
—— corrections for steam conditions, 142 (II)
—— critical speeds of, 148 (II)
—— design of, 113 (II)
—— diaphragm gland loss, 129 (II)
—— disc drum type, 129 (II)
—— —— centrifugal stresses in, 145 (II)
—— —— friction in, 131 (II)
—— double helical gear for, 160 (II)
—— dummy loss in, 130 (II)
—— efficiency of, 114 (II)
—— —— variation in, with steam pressures, 138 (II)
—— flow of steam, 103 (II)
—— forgings for, 151 (II)
—— gear, 160 (II)
—— geared, installations of, 996 (I), 166 (II), 290 (II)
—— glands, 131 (II)
—— high-pressure, 140 (II)
—— hydraulic transmission, 163 (II)
—— impulse, 120 (II)
—— labyrinth packings of, 130 (II)
—— losses in, 129 (II)
—— low pressure, 140 (II)
—— lubrication of, 158 (II)
—— marine, 290 (II)
—— materials for construction of, 151 (II)
—— mixed pressure, 141 (II)
—— multi-pressure stage, 122 (II)
—— nozzles of, 107 (II)
—— oils, 158 (II)
—— packing, labyrinth, 130 (II)
—— Parsons, steam blades and caulking pieces for, 127 (II)
—— pitch of impulse blades, 125 (II)
—— power station installations, 164 (II)
—— reaction, 125 (II)
—— —— design, 128 (II)
—— recent developments in steam conditions and modifications in operating cycle, 134 (II)
—— recipro-, combined, 289 (II)
—— reducing, 141 (II)
—— reduction gears, 160 (II), 290 (II)
—— regenerative feed heating, 137 (II)
—— re-superheating, 139 (II)
—— shafts, 148 (II), 150 (II)
—— ship, 290 (II)
—— single pressure stage, 120 (II)
—— speeds, critical, 148 (II)
—— starting large, 159 (II)
—— steam consumption in, 115 (II)
—— —— pressure increased, 136 (II)
—— —— temperature increased, 136 (II)
—— steels, 152 (II)
—— stresses in, 145 (II)
—— tests of, corrections, 142 (II)
—— tip clearance loss of, 132 (II)
—— types of, 140 (II)
—— variation in efficiency with steam pressure, 138 (II)
—— vibration, wheel and blade, 148 (II)

Turbines.
(Steam), water glands of, 131 (II)
——— windage, loss in, 131 (II)
(Water), 835 (I)
——— choice of type of, 836 (I)
——— governor power, 844 (I)
——— governors of, 844 (I)
——— impulse, 835 (I)
——— Pelton, 837 (I)
——— ——— buckets, 839 (I)
——— ——— nozzles, 839 (I)
——— ——— regulation, 839 (I)
——— pipe lines, 845 (I)
——— pressure changes in pipe line, 846 (I)
——— reaction, 835 (I), 840 (I)
——— ——— general proportions of, 843 (I)
——— ——— regulation of, 843 (I)
——— relations between power speed, head, etc., 835 (I)
——— settings, 840 (I)
——— sluice gates, 845 (I)
——— specific speed, 836 (I)
——— speed changes, momentary, 845 (I)
——— strainer racks, 845 (I)
——— ——— valves, 846 (I)
Turbo-electric ship propulsion, 292 (II)
——— pumps, 1129 (II)
——— reciprocating machinery, 289 (II)
Turbulent flow, 883 (I)
Turner Bros. Asbestos Co., C.A.F. jointing, 1258 (II)
Turpentine for paints, 987 (II)
Turplat steel rail key, 484 (II)
Tusk tenon, 313 (I)
Twist drills, 1055 (I)
——— drilling notes on, 1057 (I)
——— grinding of, 1055 (I)
——— high speed, 1059 (I)
——— Morse wire gauge, 1057 (I)
——— taper measures for, 1056 (I)
Two-stroke cycle, 197 (II), 218 (II)
Tygart dam, 697 (I)
Type metal, 1248 (I)
Tyres, carriages and wagon railway, 551 (II)
——— for locomotive wheels, 541 (II), 550 (II)
——— railway-wagon on wheels, 551 (II)
——— widths of, 555 (I)

'U' LEATHERS, 806 (I)
Utralumin, 1230 (I)
Underground haulage, cost of, 889 (II)
Unit, British Thermal, 54 (I)
United Kingdom Self-adjusting Anti-friction Metallic Packing Synd., Ltd., metallic piston-rod packing, 1281 (II)
——— States Metallic Packing Co.'s Bradford steam trap, 1254 (II)
——— ——— Packing, 1260 (II)
——— ——— river regulation works, 584 (I)
Universal joints, 1272 (II)
Unwin's formula for flow of gas in mains, 645 (II)
——— ——— weight of girders, 492 (I)
——— rule for rivets, 233 (I)
Urinals, 1795 (II)
'Usco' air preheater, 1226 (II)

'V' BELTING, 916 (I)
'V' driving ropes, 917 (I)
'V' notches, 748 (I)
——— method of calculating quantity of water flowing in a stream, 674 (I)
——— weir recorder, 197 (I)
——— Vaalbank dam, 697 (I)
Vacuum brake, 583 (II)
——— correction for steam turbines, 144 (II)
——— heating system, 671 (II)
Valuation of masonry, 331 (I)
——— minerals, 941 (II)
——— quinquennial, 1019 (II)
Valve gear, 94 (II), 538 (II)
——— Allens, 100 (II)
——— Caprotti, 540 (II)
——— Lentz, 539 (II)
——— locomotive, 98 (II), 538 (II)
——— slide, 94 (II)
——— Walschaert, 538 (II)
——— Zeuner diagram, 94 (II)
——— re-facers, 1255 (II)
——— springs, motor car, 223 (II), 333 (II)
Valves, brake, 579 (II)
——— bypass, 782 (I)
——— Crosby, 1249 (II)
——— cushioning, 782 (I)
——— cylindrical, 537 (II)
——— hydraulic control, 777 (I)
——— inlet, aero-engine, 461 (II)
——— internal combustion engine, 199 (II), 216 (II)
——— internally cooled exhaust, 459 (II)
——— Klinger, 1250 (II)
——— locomotive, 537 (II)
——— manipulating, 777 (I)
——— 'Newman Milliken' glandless lubricated plug valves, 1251 (II)
——— pump, 803 (I)
——— radio, 961 (II)
——— ——— base connexions, 961 (II)
——— ——— code of practice, 963 (II)
——— ——— matching to loudspeaker, 972 (II)
——— ——— miniature, 962 (II)
——— reducing, 59 (II)
——— reflux, 765 (I)
——— relief, 780 (I)
——— safety, 55 (II), 1249 (II)
——— ——— areas, 55 (II)
——— ——— ——— proposed new rules for, 57 (II)
——— ——— Board of Trade rules for, 56 (II)
——— ——— Bureau Veritas rules for, 57 (II)
——— ——— hydraulic, 1273 (I)
——— ——— Lloyd's rules for, 57 (II)
——— ——— nozzle type, 1249 (II)
——— sluice, 764 (I)
——— steam, 1249 (II)
——— stop, 779 (I)
Vanadium, 1211 (I)
——— steels, chrome, 1167 (I)
Vanes, impeller, 820 (I)
Vapour, lock at high altitude, 454 (II)
——— specific gravity, 125 (I)
——— tension in liquid fuels, 192 (II)
——— ——— of water, 192 (II)
Variable delivery pressure pumps, 770 (I), 1113 (II)
——— density channel, 392 (II)
——— pitch airscrew, 409 (II)

Variable Speed Gear pump, 1113 (II)
——— 'VSG,' hydraulic variable speed gear, 1114 (II)
Variation of compass, annual, 259 (I)
——— to find, 259 (I)
Varnish for galvanised iron, 239 (I)
Varnishes, 985 (II)
——— insulating, 989 (II)
——— stoving, 988 (II)
——— synthetic resin, 988 (II)
Vectors, 29 (I)
Vee engines, 243 (II)
Vehicles, electric battery, 384 (II)
——— charging, 385 (II)
——— road, regulations, 554 (I)
——— working costs, comparative, 384 (II)
Velocity in hydraulic mains, 788 (I)
——— light, 64 (I)
——— of rivers and canals, 578 (I), 680 (I), 708 (I)
——— water in pipes, 735 (I)
——— sound, 69 (I)
Ventilation, air cooling, 677 (II)
——— distribution of, 675 (II)
——— filtering, washing and conditioning, 676 (II), 1282 (II)
——— flow and velocity, 679 (II)
——— humidity, 678 (II)
——— necessary for, 675 (II)
——— fans, 681 (II), 903 (II), 1285 (II)
——— furnace, mining, 903 (II)
——— of hydraulic mains, 788 (I)
——— mines, 901 (II)
——— sewers, 776 (II)
——— silencing equipment, 683 (II)
Venturi flumes, 198 (I)
——— gas meter, 199 (I)
——— tubes, 207 (I), 753 (I)
——— water meter, 195 (I), 752 (I)
Versed sines for rails, table of, 501 (II)
Vertical boilers, 13 (II)
Vessels, welded pressure, 33 (II)
Vibration, 69 (I)
——— damping capacity of cast iron, 1154 (I)
——— devices, 77 (I)
——— insulation against, 76 (I), 1097 (II)
——— machinery, 77 (I)
——— torsional, of crankshafts, 235 (II)
——— wheel and blade, in steam turbines, 149 (II)
Vickers-Armstrongs Ltd., 'VSG' Variable Delivery pressure pump, 1113 (II)
——— 'VSG' variable speed gear, 1114 (II)
——— hardness scale, 1190 (I)
Victoria stone, 358 (I)
Villon's rust proofing process, 1194 (I)
V.I.R. cables, 1384 (II)
Visco Engineering Co., Ltd., air filter, 1282 (II)
——— dust collectors, 1283 (II)
——— water and oil coolers, 1264 (II)
Viscometers for testing bituminous materials, 550 (I)
Viscosity index, 883 (I)
——— of oils, 874 (I), 883 (I), 997 (II)
——— water, 744 (I)
Viscosity tests for bituminous materials, 549 (I)
Viscous flow air meter, 201 (I)
——— of oils, 883 (I)
'Voidmeter" for packing sleepers, 486 (II)
Voltage transformers, 1510 (I)
Voltages, standard, 1379 (I)
Voltmeters, 1507 (I)
Volume meters, 204 (I)
Volumetric efficiency of aero-engines, 450 (II)
Volute pumps, 811 (I)
'VSG' variable delivery pump, 1113 (II)
——— speed gear, 1114 (II)
Vulcanising presses, 791 (I)

'W' central heating boiler, Charles McNeil, Ltd., 1217 (II)
Waddel's formula, 500 (I)
Wagons and carts, loads of, 432 (I)
——— railway, 574 (II)
——— tip, 432 (I)
Wall-breakwaters, 609 (I)
Walls, 319 (I)
——— breakwater, 609 (I)
——— brick, safe loads on, 437 (I)
——— dampness of, to remedy, 359 (I)
——— dock, 615 (I)
——— foundation of, 326 (I), 459 (I)
——— furnace, 1236 (II)
——— London Building Act rules, 319 (I), 388 (I)
——— quays, cost of, 617 (I)
——— reinforced concrete, 388 (I), 407 (I)
——— retaining-, 407 (I)
——— brick, 460 (I)
——— equilibrium of, 456 (I)
——— foundations of, 459 (I)
——— pressure of earth, 458 (I)
——— water on, 455 (I)
——— reinforced concrete, 407 (I), 464 (I)
——— rules for, 460 (I)
——— Trautwine's rule for, 461 (I)
——— weight of buildings on, 457 (I)
——— rules for building, 319 (I), 388 (I)
——— stone and brick, loads on, 436 (I)
——— stresses, permissible, 321 (I)
——— surcharged, 457 (I)
——— warehouse, London Building Act rules, 320 (I)
Walschaerts valve gear, 538 (II)
Wander plugs, colour code, 968 (II)
Wanner Photometric pyrometer, 1138 (I)
Ware, sanitary, 370 (I)
Warehouse walls, 320 (I)
Warrington wire-gauge, 89 (I)
Washers, air, 676 (II)
——— and scrubbers, gas, 627 (II)
——— weight of, 155 (I)
Waste heat boilers, 623 (II), 1211 (II)
——— recovery of, 1372 (I)
——— water detection, 1107 (II)
——— meters, 1107 (II)
Waste weir capacities, 712 (I)
Watch, determination of south by means of, 269 (I)
——— screw threads, 92 (I)
Watches, oil for, 996 (II)

Water, 729 (I)
——— bearing formations, 729 (I)
——— boiler, treatment of, 48 (II), 49 (II)
——— boiling-points of, 52 (I)
——— chemical purification of, 733 (I)
——— chlorinating, 733 (I)
——— circulating screens, 1263 (II)
——— closets, 797 (II)
——— compressibility of, 744 (I)
——— consumption of, in locomotives, 562 (II)
——— cooled condensers, 626 (II)
——— cooler, Heenan, 1273 (II), 1287 (II), 1292 (II)
—————— Visco, 1264 (II)
——— cooling for internal combustion engines, 1273 (II)
—————— ponds, 181 (II), 1264 (II)
—————— towers, 182 (II), 1264 (II)
——— de-aeration, 1238 (II)
——— discharge of, through orifices, 671 (I), 745 (I)
————————— pipes, 675 (I), 735 (I)
————————— sewers, 666 (I)
————————— weirs, 750 (I)
——— dynamometer, Froude, 43 (I)
——— evaporation from, 701 (I)
——— expansion of, 743 (I)
——— feed, 48 (II)
—————— conditioning of, 52 (II)
—————— heating, 1248 (II)
—————— for locomotives, 548 (II)
—————— regulators, 1243 (II)
—————— softeners, 49 (II)
—————— treatment of, 48 (II), 49 (II)
——— filtration, 729 (I), 1105 (II)
——— flow of, through orifices, 671 (I)
————————— flumes and channels, 675 (I), 756 (I)
————————— pipes, 675 (I), 735 (I), 756 (I), 760 (I)
——— flowing in streams, to determine quantity of, 674 (I), 679 (I)
—————— over weirs, table for, 672 (I), 750 (I)
—————— through pipes, head required for, 720 (I)
—————— under bridges, 490 (I)
——— gas, 642 (II)
——— gathering grounds, 729 (I)
——— gauges (boiler), 57 (II)
—————— 'Klinger,' 1241 (II)
——— glands, steam turbine, 131 (II)
——— glass cements, 353 (I)
——— hardness of, 731 (I)
——— head losses, 762 (I)
——— horse-power of, 720 (I)
——— hot, heating systems, 668 (II)
—————— boiler, 1217 (II)
—————— supply of, 672 (II)
——— irrigation, duty of, 710 (I)
——— leakage detection, 734 (I), 1107 (II)
——— losses from canals, 595 (I), 708 (I)
—————— of head, 762 (I)
——— main pipes, cast-iron, weight of, 140 (I)
——— mains, deterioration of, 759 (I)
——— metering, 191 (I), 756 (I), 1107 (II)
——— meters, 193 (I), 756 (I), 1107 (II)
——— meters, cylindrical piston, 194 (I)
—————— disc type, 195 (I)
Water, meters, flap type, 193 (I)
—————— nutating piston, 194 (I)
—————— piston, 193 (I)
—————— venturi, 195 (I), 752 (I)
——— ozonisation, 733 (I)
——— paints, 986 (II)
——— pipe bends, unbalanced pressure on, 726 (I)
—————— fittings, weight of, 147 (I)
—————— screw threads, 146 (I)
——— pipes, British standard, cast iron, 114 (I), 115 (I)
—————— design of, 722 (I)
—————— incrustation and corrosion of, 734 (I)
—————— iron, weight of, 145 (I)
—————— lead, weight of, 738 (I)
—————— main, cast-iron, weight of, 140 (I)
—————— steel, 116 (I), 723 (I)
—————— thickness of, 146 (I)
—————— threads of, 146 (I)
—————— wood stave, 724 (I)
——— power, available, 719 (I)
—————— discharge, 721 (I)
—————— compared with coal power, 719 (I)
—————— development of, 719 (I)
—————— to compute, 719 (I)
——— pressure, 744 (I)
—————— in hydraulic mains, 767 (I)
—————— on retaining walls, 455 (I)
——— pump for automobiles, 233 (II)
——— purification of, 730 (I), 1105 (II)
—————— for boilers, 48 (II)
—————— standard of, 730 (I)
——— rain, 659 (I)
——— raising by compressed air, 826 (I)
——— raw, temporary hardness of, 51 (II)
——— screens, circulating, 1263 (II)
——— soap test for hardness of, 49 (II)
——— softening, 731 (I), 49 (II)
—————— cost of, 732 (I)
—————— lime and soda process, 49 (II)
—————— Zeolite process, 52 (II)
——— specific gravity of, 119 (I)
——— standard of purity of, 730 (I)
——— steam required to heat, 6 (II)
——— sterilisation, 733 (I)
——— storage (*see* Reservoirs)
——— supply, irrigation, 701 (I), 729 (I)
——— surface, evaporation from, 701 (I)
——— testing, 731 (I), 1197 (II)
——— towers, 1264 (II)
——— troughs, locomotive, 564 (II)
——— tube boiler superheaters, 1229 (II)
—————— tubes, 42 (II)
—————— boilers for land service, 9 (II), 1201 (II)
————————— marine service, 293 (II), 297 (II)
————————— stokers, 1220 (II)
——— turbines (*see* Turbines, water)
——— vapour tension of, 192 (II)
——— velocity of, in pipes and channels, 578 (I), 673 (I), 708 (I), 735 (I)
——— viscosity of, 744 (I)
——— volume and weight relation, 119 (I)
——— waste, detection of, 1107 (II)
——— weighers, 196 (I)

WAT

Water, weight of, 119 (I), 145 (I), 743 (I)
Waterless gasholders, 631 (II)
Waterproof cements, 357 (I)
——— glues, 354 (I)
Waterproofing concrete, 349 (I)
Watertight bulkhead doors, hydraulic, 794 (I)
Waterway of bridges, 490 (I)
Waterways, German, 587 (I)
——— navigable, 577 (I)
Watson dynaometer, 39 (I)
Wattmeters, 1507 (I)
Wave action on breakwater, 607 (I)
——— effective horse-power of ships, 280 (II)
——— transmission of energy, 772 (I)
Wavelengths, 957 (II)
——— conversion tables, 958 (II)
'Wayne' cable reel, 1199 (II)
Weather, effect of, on stone, 334 (I)
Wedges, 313 (I)
Weed killing, chemical, on railways, 494 (II)
Weighing machines, coal, 206 (I)
——— water, 196 (I)
Weight, definition of, 22 (I)
——— of aero-engines, 457 (II)
——— aluminium bar and strip, 133 (I)
——— sheet, 135 (I)
——— tubes, 152 (I)
——— angle iron, 136 (I)
——— steel, 137 (I)
——— ball cast iron, 127 (I)
——— balls of various metals, 127 (I)
——— bar cast iron, 127 (I)
——— boiler tubes, 41 (II)
——— bolt heads, 154 (I)
——— brass bars, 131 (I)
——— pipes, 141 (I)
——— square, 129 (I)
——— tubes, seamless, 151 (I)
——— wire, 154 (I)
——— braziers' copper rivets, 156 (I)
——— buildings, effect of, on retaining walls, 457 (I)
——— cast-iron balls, 127 (I)
——— gas pipes, 145 (I)
——— pipes, 140 (I), 144 (I)
——— water pipes, 140 (I), 145 (I)
——— castings from wood patterns, 154 (I)
——— coal, 857 (II)
——— composition nails, 156 (I)
——— concrete, 347 (I)
——— copper bars, 131 (I)
——— nails, 155 (I)
——— pipes, 141 (I)
——— rivets, 156 (I)
——— square, 129 (I)
——— tubes, seamless, 149 (I)
——— wire, 154 (I)
——— crowds, 39 (I)
——— ebonite, 139 (I)
——— galvanised corrugated iron sheets, 239 (I)
——— steel sheets, 238 (I)
——— gas-pipes, 145 (I)
——— hexagon rod metal, 131 (I)
——— hoop-iron, 153 (I)
——— iron, angle and tee, 136 (I)
——— boiler tubes, 41 (II)
——— flat rolled, 129 (I)
——— rivets, 156 (I)
——— wire, 154 (I)

WEI

Weight of iron, wrought, 128 (I)
——— lead bars, 131 (I)
——— pipes, 141 (I), 738 (I)
——— materials, 119 (I)
——— metal sheets and bars, 127 (I)
——— metals, 127 (I)
——— meters, conveyor, 205 (I)
——— minerals, 860 (II)
——— nails, 155 (I)
——— nut and bolt heads, 154 (I)
——— nuts, cold-punched, 154 (I)
——— octagon rod metal, 131 (I)
——— pipe-jointing material, 157 (I)
——— pipes and tubes, 140 (I)
——— railway rails, 474 (II), 477 (II)
——— rolled flat iron, 129 (I)
——— lead, copper, and brass, sheet and bars, 131 (I)
——— roofs, 301 (I)
——— ropes, steel, 131 (I)
——— screws, 156 (I)
——— sheet, bar, and ball metals, 127 (I)
——— cast iron, 127 (I)
——— lead, 132 (I)
——— metals, 129 (I)
——— zinc, 132 (I)
——— ship machinery, 279 (II)
——— ships, 277 (II)
——— slates, 298 (I)
——— snow on roofs, 305 (I)
——— socket pipes and fittings, 146 (I), 148 (I)
——— soils, 432 (I)
——— square steel, 131 (I)
——— steam pipes, 145 (I)
——— steel, 129 (I), 131 (I)
——— flat, 131 (I)
——— octagonal, 129 (I)
——— ropes, 131 (I)
——— round, 129 (I)
——— square, 129 (I)
——— tubes, 140 (I)
——— wire ropes, 131 (I)
——— wrought, 128 (I)
——— stones, 333 (I)
——— tee iron, 136 (I)
——— steel, 137 (I)
——— timber, 283 (I)
——— tinplates, 153 (I)
——— tubes, 140 (I), 146 (I)
——— washers, 155 (I)
——— water, 119 (I), 743 (I)
——— pipe fittings, 147 (I)
——— wire, 154 (I), 162 (I)
——— wrought-copper nails, 155 (I)
——— iron and steel, 128 (I)
——— nails, 155 (I)
——— pipes, 142 (I)
Weights, 6 (I)
——— and measures, 1 (I)
——— foreign, 10 (I)
——— specific gravities, 120 (I)
——— atomic, 120 (I)
——— bridgework, 492 (I)
——— metrical, 7 (I)
——— of girders, 492 (I)
——— motor cars, 369 (II)
Weir, G. & J., air compressors, 1290 (II)
——— pumps, 1262 (II)
——— boiler feed regulator, 1247 (II)
——— de-aerator, 1238 (II)

Weir, G. & J., direct-acting steam feed pump, 1127 (II)
—— ejector air pump, 1262 (II)
—— 'Electrofeeder,' 1130 (II)
—— evaporators, 1238 (II)
—— extraction pump, 1262 (II)
—— feed water heaters, 1248 (II)
—— —— regulators, 1247 (II)
—— multi-stage centrifugal pump, 1130 (II)
—— surface condenser, 1261 (II)
—— turbo-feed pump, 1129 (II)
Weir measurement, water, 674 (I)
—— recorders, 197 (I)
—— waste, capacities, 712 (I)
Weirs, and notches, 672 (I), 746 (I)
—— bellmouth, 751 (I)
—— Cippoletti, 748 (I)
—— effects of shape of Nappe, 749 (I)
—— formulæ for, 673 (I), 746 (I)
—— rectangular, 746 (I)
—— river control and diversion, 699 (I)
—— sector near Bremen, 583 (I)
—— syphon, 694 (I), 751 (I)
—— to raise a river a certain height by, 674 (I)
—— trapezoidal, 747 (I)
—— water flowing over, 672 (I), 674 (I)
Weld tests, 1283 (I)
Welded pressure vessels, 33 (II)
—— rails, 491 (II)
Welders, training, 1282 (I)
Welding and cutting, 1267 (I), 1180 (II)
—— acetylene, 1267 (I)
—— aeroplanes, 416 (II)
—— aluminium, 1280 (I)
—— arc, 1286 (I)
—— atomic hydrogen system, 1289 (I)
—— Barimar, 1180 (II)
—— blow pipes, 1274 (I)
—— brass, 1281 (I)
—— bridge, 1291 (I)
—— bronze, 1281 (I)
—— butt, 1290 (I)
—— button and disc depression, 1291 (I)
—— carbon arc, 1286 (I)
—— —— steels, 1279 (I)
—— cast iron, 1279 (I)
—— copper, 1281 (I)
—— —— hydrogen electric, 1293 (I)
—— electric, 1286 (I), 1180 (II)
—— —— arc, 1286 (I), 1183 (II)
—— —— Murex, 1183 (II)
—— electro-percussive, 1292 (I)
—— expansion and contraction, 1277 (I)
—— filling materials for, 1276 (I)
—— fluxes, 1277 (I)
—— goggles, 1276 (I)
—— hydrogen arc, 1289 (I)
—— lead, 1282 (I)
—— machines, 1282 (I)
—— magnesium, 1282 (I)
—— mash, 1292 (I)
—— metallic arc, 1287 (I), 1288 (I)
—— multiple electrode, 1291 (I)
—— Murex, 1183 (II)
—— nickel, 1282 (I)
—— oxy-acetylene, 1267 (I), 417 (II)
—— —— acetylene supply 1271 (I)
—— —— blowpipe for, 1274 (I)
Welding oxy-acetylene cutting, 1283 (I)
—— —— equipment, 1267 (I)
—— —— flame characteristics, 1275 (I)
—— —— oxygen supply, 1269 (I)
—— —— piping, 1273 (I)
—— —— regulators, 1273 (I)
—— —— safety valves, 1273 (I)
—— preheating, 1277 (I)
—— preparation of work, 1277 (I)
—— projection, 1292 (I)
—— protective equipment, 1276 (I), 1288 (I)
—— rails, 491 (II)
—— regulators, 1273 (I)
—— repairs, 1180 (II)
—— resistance, 1290 (I)
—— seam, 1292 (I), 417 (II)
—— speed and cost of, 1289 (I)
—— spot, 1291 (I), 417 (II)
—— steel, 1200 (I), 1278 (I)
—— Thermit, 1294 (I)
Weldless steel pipes for water, gas and sewage, 114 (I)
—— tubes for locomotive boilers, 42 (II), 556 (II)
Welds, fatigue of, 187 (I)
—— testing of, 1283 (I)
Welland Canal, Canada, 602 (I)
Wellington formula for safe load on piles, 440 (I)
Wellman Bibby Co., Ltd., flexible couplings, 1269 (II)
Wells, 738 (I)
—— A. C. & Co., Ltd., lathe cans, 1151 (II)
—— —— lime spraying plant, 1297 (II)
—— —— oil filters, 1151 (II)
—— —— spray painting equipment, 1297 (II)
—— artesian, pumps for, 808 (I)
—— brickwork of, 738 (I)
—— pumping from, 808 (I)
—— —— by compressed air, 826 (I)
West Coast iron, 1141 (I)
West, true, to determine, 261 (I)
Weston crane brake, 811 (II)
Westinghouse automatic brake, 579 (II)
Wetted surface of ships, 274 (II)
Wharf cranes, 806 (II)
—— walls, 463 (I)
Wheel base of locomotives, 542 (II)
—— vibration in steam turbines, 148 (II)
Wheels, abrasive, 1099 (I) (*see also* Grinding wheels)
—— chain, 966 (I)
—— —— pitch diameter, 964 (I)
—— grinding, 1098 (I) (*see also* Grinding wheels)
—— locomotive, counterbalancing, 542 (II)
—— —— tyres of, 541 (II)
—— railway wagon and carriage, 577 (II)
—— teeth in chain, 962 (I)
Whistling buoys, 645 (I)
White lead paints, 982 (II)
—— metal alloy, 1243 (I)
—— —— molten effect on steel, 1203 (I)
—— nickel alloy, 'Adnic,' 1220 (I)

White spirits, 987 (II)
Whitecross Co., Ltd., 'Tru-Lay' and 'Contra-Lock' wire ropes, 1154 (II)
Whitworth castle nuts, 102 (I)
—— nuts and bolts, 98 (I), 200 (I)
—— screw threads, 89 (I), 91 (I)
—— stanadrd screws for watch and instrument makers, 92 (I)
—— wire gauge, 89 (I)
Width of river, to measure, 251 (I)
Wilkinson Rubber Linatex Ltd., Linatex rubber, 1301 (II)
Williams drop-forged fishplate, 512 (II)
Wind channels, 390 (II)
—— engines, 753 (II)
—— power, 720 (I)
—— pressure and weight of snow on roofs, 305 (I)
—— —— on bridges, 497 (I)
—— —— on buildings, 319 (I)
—— tunnels, 390 (II), 1094 (II)
Windage loss in steam turbines, 131 (II)
—— electric, 897 (II)
—— engines, mining, 895 (II)
—— —— steam consumption of, 79 (II)
Wing fan, F. W. Potter & Soar, Ltd., 1285 (II)
—— flaps, aeroplane, 395 (II)
—— nuts, 104 (I)
Winn, C. & Co., Ltd., screwing machines, 1162 (II)
—— —— tube cutting-off machines, 1165 (II)
Winzing and raising in mining, 869 (II)
Wire, annealed and bright iron, Rylands', 162 (I)
—— braiding, woven, Sparklets, Ltd., 1180 (II)
—— breaking strength, 162 (I)
—— copper, breaking strain of, 163 (I)
—— —— enamelled, 1412 (I)
—— cut bricks, 364 (I)
—— drill gauge, 1057 (I)
—— fatigue strength of, 186 (I)
—— forming machines, 1160 (II)
—— gauge standards used in the United States, 88 (I)
—— gauges, 85 (I)
—— hard-drawn copper, 163 (I)
—— iron, breaking strains of, 162 (I)
—— —— weight of, 154 (I), 162 (I)
—— ropes, 177 (I), 933 (I), 1154 (II)
—— —— aerial carrying, 950 (I)
—— —— barrels and pulleys, 939 (I)
—— —— breaking stresses, 933 (I)
—— —— capping, 946 (I)
—— —— clamps, 948 (I)
—— —— coiling on drums, 958 (I)
—— —— crane, 936 (I)
—— —— deterioration, 953 (I)
—— —— durability of, 940 (I)
—— —— dynamic stresses in, 957 (I)
—— —— engineering, 934 (I)
—— —— factors of safety, 946 (I)
—— —— flattened strand, 941 (I), 958 (I)
—— —— guides and rubbing ropes, 950 (I)
—— —— haulage, 951 (I)
—— —— hooks for, 937 (I)
—— —— Lang's Lay, 933 (I)
—— —— lift and hoist, 939 (I)
—— —— lubrication, 955 (I)
Wire, ropes, mechanical properties, 937 (I)
—— —— mining, 942 (I)
—— —— non-rotating, 942 (I)
—— —— oil well, 941 (I)
—— —— pulleys and drums, 936 (I), 939 (I), 941 (I), 944 (I)
—— —— selection of, 952 (I)
—— —— shackles, 959 (I)
—— —— shipping, 935 (I), 942 (I)
—— —— slings, 959 (I)
—— —— socketing, 947 (I)
—— —— —— club end method, 947 (I)
—— —— splicing, 955 (I)
—— —— stresses in, 945 (I), 957 (I)
—— —— thimble splices, 948 (I)
—— —— Tru-lay, 941 (I)
—— —— Whitecross Contra-Lock, 1154 (II)
—— Rylands', 162 (I)
—— springs, safe loads for, 898 (I)
—— —— steel, 334 (II)
—— strength of, 162 (I)
—— weight of, 131 (I), 154 (I)
Wireless (*see* Radio communication)
—— signalling in fog, 656 (I)
Wires and springs, fatigue strength of, 186 (I)
Wiring, crane, 814 (II)
—— electric-light systems, Institution of Electrical Engineers' rules for, 1427 (I), 1459 (I)
Witness, engineer as, 1029 (II)
Wöhler testing machine, 182 (I)
—— tests for stresses, 521 (I)
Wonham rail joint shim, 479 (II)
Wood, 179 (I), 278 (I) (*see also* Timber)
—— airscrews, 408 (II)
—— beams, strength, 222 (I)
—— charcoal fuel, 1330 (I)
—— creosoting of, 285 (I)
—— cubic feet per ton, 283 (I)
—— drying and seasoning of, 289 (I)
—— floors, 305 (I)
—— fuel, 1330 (I)
—— holding power of nails in, 179 (I)
—— measures, 277 (I)
—— moisture content, 289 (I)
—— poles, strength of, 296 (I)
—— preservative processes, 285 (I)
—— preservatives and stains, 285 (I), 994 (II)
—— resistance to reversal of stress, 179 (I)
—— roofs, 301 (I)
—— safe working stresses, 295 (I)
—— screws, 107 (I)
—— seasoning, 289 (I)
—— —— increase of strength by, 295 (I)
—— shrinkage of, 289 (I)
—— sleepers, 481 (II)
—— specific gravities, 123 (I), 124 (I)
—— stairs, 316 (I)
—— stave pipes, 724 (I)
—— storage of, 293 (I)
—— strength and elasticity of, 179 (I), 293 (I)
—— turning lathes, 1040 (I)
—— types of, 278 (I)
—— weight per cubic foot, 283 (I)
—— working machinery, 1037 (I)
—— —— speeds of, 1037 (I)
Woodeson feed pumps, 1121 (II)
Wood's alloy, 52 (I)

Woranorah Dam, 688 (I)
Work, 25 (I)
—— done by men and animals, 35 (I)
—— hardening test, 1092 (I)
Workington steel sleeper, 483 (II)
Works contract, powers of an engineer, 1030 (II)
—— electrical regulations, 1459 (I)
—— planning and layout, 905 (I)
Worm gear applications, 1007 (I)
—— calculations, 1004 (I)
—— location, 1007 (I)
—— gearing, 1001 (I)
—— —— horse-power formulæ, 1007 (I)
—— —— Hindley or globoidal, 1001 (I)
—— —— nomenclature, 1003 (I)
Worn parts, machinery, saving of, 1082 (I)
Woven belts, 909 (I)
'W.R.' CO_2 Indicator, 1193 (II)
Wrought iron, 128 (I), 1141 (I)
—— chains, annealing, 176 (I)
—— nails, weight of, 155 (I)
—— pipes, 142 (I)
—— tubes, weight of, 142 (I)
—— weight of, 128 (I), 142 (I)
—— welding, 1278 (I)

X-RAYS in industry, 1525 (I)

'Y' ALLOY, 1236 (I)
Yarrow boilers, land type, 1217 (II)
—— —— marine type, 298 (II)
Young's modulus of elasticity, 158 (II)
—— —— timber in tension, 179 (I)

ZEOLITE process of water softening, 52 (II)
Zeuner valve gear, 94 (II)
Zinc, 1205 (I), 1223 (I)
—— aluminium alloys, 1228 (I)
—— and copper, alloys of, 1212 (I)
—— castings, weight of, 154 (I)
—— chloride solution, 1257 (I)
—— impurities in, 1205 (I)
—— ore, 859 (II)
—— roofing, 299 (I)
—— sheet, trade gauge, 86 (I)
—— —— weight of, 132 (I)
—— tin alloys, 1245 (I)
Zirconium, 1211 (I)

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Brushware (Industrial).
Kleen-e-Ze Brush Co., Ltd.

Cabinets.
Steel Equipment Co., Ltd.
A. C. Wells & Co., Ltd.

Cables, Electric.
Craigpark Electric Cable Co., Ltd.
Scottish Cables, Ltd.

Canvas Belting.
George Angus & Co., Ltd.
Bell's Asbestos and Engineering, Limited.

Capstan Lathes.
Alfred Herbert, Ltd.
Norman E. Potts, Ltd.
Charles Taylor (Birmingham), Ltd.
Timbrell & Wright, Ltd.
A. C. Wickman, Ltd.

Capstans.
Sir William Arrol & Co., Ltd.
Clarke, Chapman & Co., Ltd.
M. B. Wild & Co., Ltd.

Carbide Tipped Tools.
English Steel Corporation, Ltd.
A. C. Wickman, Ltd.

Carbonising Plant.
West's Gas Improvement Co., Ltd.

Case Hardening Compounds.
Amalgams Co., Ltd.

Casting Machines.
Constructional Engineering Co., Ltd.

Castings and Forgings.
Bull's Metal & Melloid Co., Ltd.
Clyde Alloy Steel Co., Ltd.
English Steel Corporation, Ltd.
International Meehanite Metal Co., Ltd.
Manganese Bronze & Brass Co., Ltd.
W. L. Marrison, Ltd.
Railway Signal Co., Ltd.
S. Russell & Sons, Ltd.
Stanton Ironworks Co., Ltd.

Castings (Non-Ferrous).
B.K.L. Alloys, Ltd. (Aluminium)
John Broadfoot & Sons, Ltd.
C. S. Madan & Co., Ltd.
Manganese Bronze & Brass Co., Ltd.

Castings (Steel).
F. W. Brackett & Co., Ltd.
Clyde Alloy Steel Co., Ltd.
English Steel Corporation, Ltd.

Centres (for Lathes).
C. S. Madan & Co., Ltd.

Centrifugal Pumps. *See* Pumps.

Chain Cases.
Morse Chain Co., Ltd.

Chain Couplings.
Morse Chain Co., Ltd.
Renold & Coventry Chain Co., Ltd.

Chain Wheels.
Morse Chain Co., Ltd.
Renold & Coventry Chain Co., Ltd.

Chains.
Morse Chain Co., Ltd.
Renold & Coventry Chain Co., Ltd.

Chemical Plant.
Blairs, Ltd.
John Thompson (Wolverhampton), Ltd.

Chemical Pumps. *See* Pumps.

Chimney Stacks.
F. Braby & Co., Ltd.
John Thompson (Wolverhampton), Ltd.

Chlorinating Equipment.
Filtrators, Ltd.

Chucks
Findlay & Co.
J. H. Humphreys & Sons.
Charles Taylor (Birmingham), Ltd.
Timbrell & Wright, Ltd.

Circulating Water Screens.
F. W. Brackett & Co., Ltd.

Cisterns.
Arkinstall Brothers, Ltd.
F. Braby & Co., Ltd.
A. C. Wells & Co., Ltd.

Clutch Linings.
George Angus & Co., Ltd.
Ferodo, Ltd.

Clutches
Renold & Coventry Chain Co., Ltd.

Coal and Ash Handling Plant.
Bennis Combustion, Ltd.
Butters Brothers & Co., Ltd.
Crosthwaite Furnaces & Scriven Machine Tools, Ltd.
Mavor & Coulson, Ltd.
Ritchie-Atlas Engineering Co., Ltd.
West's Gas Improvement Co., Ltd.
Wilkinson Rubber Linatex, Ltd.

For Advertisers' Addresses and positions of their announcements, consult the following Alphabetical List, pages 1403 to 1408.

Coal Cutters.
Mavor & Coulson, Ltd.

Coal Pulverisers.
Clarke, Chapman & Co., Ltd.
Alfred Herbert, Ltd.
International Combustion, Ltd.

Cocks.
Bell's Asbestos and Engineering, Limited.
R. Klinger, Ltd.
Newman, Hender & Co., Ltd.
Rotherham & Sons, Ltd. (Brass)
John Thompson (Wolverhampton), Ltd.
Wilkinson Rubber Linatex, Ltd.

Coils, Steel Tube.
Le Bas Tube Co., Ltd.

Coke Screening and Grading Plant.
Ritchie-Atlas Engineering Co., Ltd.
West's Gas Improvement Co., Ltd.

Collars, Shafting.
Thomas A. Ashton, Ltd.
H. Fordsmith, Ltd.

Colleges and Schools, Engineering.
Glasgow, Royal Technical College.
Heriot-Watt College, Edinburgh.
King's College, University of London.
Manchester, College of Technology.
Technological Institute of Great Britain.

Colliery Plant.
International Gas Detectors, Ltd.
Mavor & Coulson, Ltd.
Robey & Co., Ltd.

Compressors.
F. W. Brackett & Co., Ltd. (Air).
British Thomson-Houston Co., Ltd.
Peter Brotherhood, Ltd. (Air).
Robey & Co., Ltd.
Siebe, Gorman & Co., Ltd. (Air).
Tecalemit, Ltd. (Air).
G. & J. Weir, Ltd. (Air and Gas).

Concreting Papers.
Ruberoid Co., Ltd.

Condenser Pressure Cleaning Units.
Kleen-e-Zee Brush Co., Ltd.

Condensers.
Richardsons Westgarth & Co., Ltd.
G. & J. Weir, Ltd.

Constructional Work, Steel.
Acme Welding and Constructional Engineering Co., Ltd.
Allott Brothers & Leigh, Ltd.
Arkinstall Brothers, Ltd.
Wm. Bain & Co., Ltd.
F. Braby & Co., Ltd.
Hall Brothers (West Bromwich), Ltd.
Lamson Engineering Co., Ltd.
London and Midland Steel Scaffolding Co. Ltd.
Railway Signal Co., Ltd.
John Thompson (Wolverhampton), Ltd.
Thos. W. Ward, Ltd.

Control Gear.
British Thomson-Houston Co., Ltd. (Electric).
Butters Brothers & Co., Ltd. (Electric).

Controls.
British Thermostat Co., Ltd. (Automatic).

Conveyors.
Acme Welding and Constructional Engineering Co., Ltd.
George Angus & Co., Ltd.
Bennis Combustion, Ltd.
R. & J. Dick, Ltd.
Lamson Engineering Co., Ltd.
Herbert Morris, Ltd.
Mavor & Coulson, Ltd.
Ritchie-Atlas Engineering Co., Ltd.
John Thompson (Wolverhampton), Ltd.
West's Gas Improvement Co., Ltd.
M. B. Wild & Co., Ltd.

Coolers.
Premier Cooler & Engineering Co., Ltd. (Water, Air and Oil).
Superheater Company, Ltd.

Cooling Towers.
Peter Brotherhood, Ltd.
Premier Cooler & Engineering Co., Ltd.

Copper Rods and Sections.
James Booth & Co., Ltd.
Delta Metal Co., Ltd.
Manganese Bronze & Brass Co., Ltd.
McKechnie Brothers, Ltd. (Extruded).

Cork Packing.
Cork Manufacturing Co., Ltd.

Corrugated Sheeting.
F. Braby & Co., Ltd.
Hall Brothers (West Bromwich), Ltd.

Cotton Ropes and Belting.
George Angus & Co., Ltd.
J. H. Fenner & Co., Ltd.
Thomas Hart, Ltd.
William Kenyon & Sons, Ltd.

Counting Instruments.
Findlay & Co.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT THE FOLLOWING ALPHABETICAL LIST, PAGES 1403 TO 1408.

Couplings, Flexible.
Renold & Coventry Chain Co., Ltd.
Wellman, Bibby Co., Ltd.

Cranes.
Sir William Arrol & Co., Ltd.
Butters Brothers & Co., Ltd.
Clarke, Chapman & Co., Ltd.
Lamson Engineering Co., Ltd.
Herbert Morris, Ltd.
Ransomes, Sims & Jefferies, Ltd. (Electric Runabout).
Thos. W. Ward, Ltd.

Crushing Machinery.
International Combustion, Ltd.

Cupro-Nickel Ingots.
McKechnie Brothers, Ltd.

Cutting-off Machines.
Charles Winn & Co., Ltd.

Cutting Tools.
Findlay & Co.
A. C. Wickman, Ltd.

Cycle Parks.
Constructors, Ltd.

Dampcourses.
Ruberoid Co., Ltd.

De-Aerating Equipment.
G. & J. Weir, Ltd.

Demagnetisers.
J. H. Humphreys & Sons.

Demineralising Plant.
Filtrators, Ltd.
Permutit, Ltd.

De-Superheaters.
Crosby Valve & Engineering Co., Ltd.
Superheater Company, Ltd.

Die Castings.
Sparklets, Ltd.

Diesel Engines.
Peter Brotherhood, Ltd.
Gleniffer Engines, Ltd.
Russell Newbery & Co., Ltd.
Swan, Hunter & Wigham Richardson, Ltd.
Wallsend Slipway & Engineering Co., Ltd.

Distilling Plant.
Blairs, Ltd.
G. & J. Weir, Ltd.

Diving Apparatus.
Siebe, Gorman & Co., Ltd.

Dock Equipment.
Sir William Arrol & Co., Ltd.

Docks, Floating.
Swan, Hunter & Wigham Richardson, Ltd.

Doors, Fireproof.
Constructors, Ltd.

Drawing Office Equipment.
Mavitta Drafting Machines, Ltd.

Drill Chucks.
English Steel Corporation, Ltd.
Frederick Pollard, Ltd.

Drillers (Well).
Alexander Munro & Co. (Johnstone), Ltd.

Drilling Machines.
James Archdale & Co., Ltd.
B.S.A. Tools, Ltd.
Charles Churchill & Co., Ltd.
Alfred Herbert, Ltd.
John Macdonald & Co. (Pneumatic Tools), Ltd.
Frederick Pollard, Ltd.
Norman E. Potts, Ltd.

Drills, Electric Portable.
B.S.A. Tools, Ltd.
Frederick Pollard, Ltd.

Drills, Twist.
Richd. W. Carr & Co., Ltd.
John Cooper & Son, Ltd.
English Steel Corporation, Ltd.
W. L. Marrison, Ltd.
Sanderson Brothers & Newbould, Ltd.

Driving Chains.
Morse Chain Co., Ltd.
Renold & Coventry Chain Co., Ltd.

Driving Ropes and Belting.
R. & J. Dick, Ltd.
Thomas Hart, Ltd.
William Kenyon & Sons, Ltd.

Drop Stamps.
Brett's Patent Lifter Co., Ltd.
Walters & Dobson, Ltd.

Drums (Oil).
F. Braby & Co., Ltd.

Drying Apparatus.
Blairs, Ltd.
John Mathison, Ltd. (Re-circulating type driers)

For Advertisers' Addresses and positions of their announcements, consult the following Alphabetical List, pages 1403 to 1408.

Duct Work.
F. Braby & Co., Ltd.
Premier Cooler & Engineering Co., Ltd.

Duralumin.
James Booth & Co., Ltd.

Dust Collecting Plant.
Dallow Lambert & Co., Ltd.
James Howden & Co., Ltd.

Dynamometers.
British Thomson-Houston Co., Ltd.
Peter Brotherhood, Ltd.
Heenan & Froude, Ltd.

Dynamos.
Ransomes, Sims & Jefferies, Ltd.

Ejectors.
James Beresford & Son, Ltd.
Green & Boulding, Ltd.
C. S. Madan & Co., Ltd.
Newman, Hender & Co., Ltd.

Electric Cables. *See* CABLES.

Electric Generating Sets.
British Thomson-Houston Co., Ltd.
Richardsons Westgarth & Co., Ltd.

Electric Motors.
James Beresford & Son, Ltd.
British Thomson-Houston Co., Ltd.
Butters Brothers & Co., Ltd.

Electric Overhead Line Equipment.
Poles, Ltd.

Electric Railway Equipment.
British Thomson-Houston Co., Ltd.
Craigpark Electric Cable Co., Ltd.

Electric Trucks.
Ransomes, Sims & Jefferies, Ltd.

Electric Welding.
Acme Welding & Construction Co., Ltd.
Barimar, Ltd.
Murex Welding Processes, Ltd. (Equipment).

Electrical Accessories.
James Beresford & Son, Ltd.
British Thomson-Houston Co., Ltd
Craigpark Electric Cable Co., Ltd.

Electrical Instruments.
Evershed & Vignoles, Ltd.

Electrical Resistances.
British Thomson-Houston Co., Ltd.

Electrical Testing and Repair Equipment
Evershed & Vignoles, Ltd.

Electrodes.
Murex Welding Processes, Ltd.

Electro-Mechanical Brakes.
British Thomson-Houston Co., Ltd.
Elliston, Evans & Jackson, Ltd.

Electro Plating Brushware.
Kleen-e-Ze Brush Co., Ltd.

Elevators.
Bennis Combustion, Ltd.
Elliston, Evans & Jackson, Ltd.
Lamson Engineering Co., Ltd. (Goods).
Mavor & Coulson, Ltd.
Ritchie-Atlas Engineering Co., Ltd.
M. B. Wild & Co., Ltd.

Enamels and Paints.
British Paints, Ltd.

Engine and Boiler Equipments.
Clyde Blowers, Ltd.

Engine Packing.
George Angus & Co., Ltd.
Bell's Asbestos and Engineering, Limited.
Cork Manufacturing Co., Ltd.
Ronald Trist & Co., Ltd.
Turner Brothers Asbestos Co., Ltd.
U.K. Anti-Friction Metallic Packing (Railway Signal Co., Ltd.)
United States Metallic Packing Co., Ltd.

Engineers' Tools.
B.S.A. Tools, Ltd.
Charles Churchill & Co., Ltd.
English Steel Corporation, Ltd.
Timbrell & Wright, Ltd.
A. C. Wickman, Ltd.

Engines, Gas.
Peter Brotherhood, Ltd.

Engines, Marine.
Gleniffer Engines, Ltd.
Parsons Marine Steam Turbine Co., Ltd.
Richardsons Westgarth & Co., Ltd.
Russell Newbery & Co., Ltd.
Swan, Hunter & Wigham Richardson, Ltd.
Wallsend Slipway & Engineering Co., Ltd.
Yarrow & Co., Ltd.

Engines, Oil.
Peter Brotherhood, Ltd.
Gleniffer Engines, Ltd.
Robey & Co., Ltd.
Russell Newbery & Co., Ltd.

Engines, Petrol.
Peter Brotherhood, Ltd.

Engines, Steam.
Peter Brotherhood, Ltd.
James Howden & Co. (Land), Ltd.
Marshall Sons & Co., Ltd.
Ransomes, Sims & Jefferies, Ltd.
Robey & Co., Ltd.

Engravers and Printers.
John Swain & Son, Ltd.

Etching Stamps and Fluid.
E. M. Richford, Ltd. (for Steel).

Evaporators.
Blairs, Ltd.
Davie & Horne, Ltd. (Sea-water).
G. & J. Weir, Ltd.

Exhaust Heat Boilers.
Cochran & Co., Annan, Ltd.
Marshall Sons & Co., Ltd.

Factory Equipment.
Constructors, Ltd.

Fans.
Alldays & Onions, Ltd.
Clyde Blowers, Ltd.
James Howden & Co. (Land), Ltd.
F. W. Potter & Soar, Ltd.

Feed Water Filters.
W. Crockatt & Sons, Ltd.
Davie & Horne, Ltd.
G. & J. Weir, Ltd.

Feed Water Heaters.
Davie & Horne, Ltd.
G. & J. Weir, Ltd.

Feed Water Regulators.
Copes Regulators. Ltd.
Ronald Trist & Co., Ltd.
G. & J. Weir, Ltd.

Feed Water Treatment.
Albright & Wilson, Ltd.
Clarke, Chapman & Co., Ltd.
Filtrators, Ltd.
Permutit Co., Ltd.

Felts, Roofing *See* Roofing.

File Making Machinery.
Fred Cam (Engineers), Ltd.
Walters and Dobson, Ltd.

File Manufacturers.
Richd. W. Carr & Co., Ltd.
English Steel Corporation, Ltd.
Sanderson Brothers & Newbould, Ltd.

Filters and Filtering Apparatus.
Bell's Asbestos and Engineering, Limited (Filter Mediums for Chemicals).
F. W. Brackett & Co., Ltd.
W. Crockatt & Sons, Ltd. (Oil).
Dallow Lambert & Co., Ltd. (Dust).
Filtrators, Ltd.
Pulsometer Engineering Co., Ltd.
Tecalemit, Ltd. (Oil).
Vokes, Ltd. (Oil).
A. C. Wells & Co., Ltd. (Oil).
Zwicky, Ltd. (Oil).

Fire Appliances.
George Angus & Co., Ltd.
Glenfield & Kennedy, Ltd.
Charles Winn & Co., Ltd.

Fire Clay Goods.
General Refractories, Ltd.
John G. Stein & Co., Ltd.

Firedamp Alarms.
International Gas Detectors, Ltd.

Flagstaffs.
Poles, Ltd.

Flame Dampers and Silencers.
Vokes, Ltd.

Flanges.
Barr, Thomson & Co., Ltd.
London & Midland Steel Scaffolding Co., Ltd.

Flexible Couplings. *See* Couplings.

Floor Polishing Machines.
W. Crockatt & Sons, Ltd.

Flooring, Iron and Steel.
Wm. Bain & Co., Ltd.
Bettles & Sons, Ltd.
F. Braby & Co., Ltd.
Cementation Co., Ltd. (Steel).
Constructors, Ltd.
Allan Kennedy & Co., Ltd. (Steel).
Stanton Ironworks Co., Ltd. (Iron).

Fluxes, Soldering.
Cementation Co., Ltd.
Fry's Metal Foundries, Ltd.

Forced and Induced Draught Apparatus.
Clyde Fuel Systems, Ltd.
Crosthwaite Furnaces & Scriven Machine Tools, Ltd.
James Howden & Co. (Land), Ltd.

Forging Equipment.
Brett's Patent Lifter Co., Ltd.
Fred Cam (Engineers), Ltd.
Charles Churchill & Co., Ltd.
Walters & Dobson, Ltd.

Forging Hammers. *See* HAMMERS.

Forgings.
Bull's Metal & Melloid Co., Ltd.
English Steel Corporation, Ltd.
Kirkstall Forge, Ltd.
Light Metal Forgings, Ltd. (Aluminium Alloy).
London & Midland Steel Scaffolding Co., Ltd.
Manganese Bronze & Brass Co., Ltd.
Charles McNeil, Ltd.

Foundations.
Cementation Co., Ltd.
Alexander Munro & Co. (Johnstone), Ltd. (Testing).

Foundry Plant.
Alldays & Onions, Ltd.
Constructional Engineering Co., Ltd.
Kleen-e-Ze Brush Co., Ltd. (Brushes)
Charles McNeil, Ltd.

Friction Linings.
George Angus & Co., Ltd.
Ferodo, Ltd.

Fuel Pulverisers.
International Combustion, Ltd.

Fuel Saving Devices.
International Gas Detectors, Ltd.
John Mathison, Ltd.

Furnace Arches,
Liptak Furnace Arches, Ltd.

Furnace Linings.
John G. Stein & Co., Ltd.

Furnace Stokers.
Ashwell & Nesbit, Ltd.
Bennis Combustion, Ltd.

Furnaces.
Alldays & Onions, Ltd.
Cementation Co., Ltd.
James Howden & Co. (Land), Ltd. (Oil).
Incandescent Heat Co., Ltd.
John Mathison, Ltd.

Furniture.
Constructors, Ltd. (Steel).
H. Morris & Co., Ltd.

Galvanised Sheets.
F. Braby & Co., Ltd.
Hall Brothers (West Bromwich), Ltd.

Galvanisers.
F. Braby & Co., Ltd.
General Galvanizers, Ltd.

Gas Analysis.
International Gas Detectors, Ltd.

Gas Burners.
Liptak Furnace Arches, Ltd.

Gas Engines. *See* ENGINES.

Gas Masks.
Siebe, Gorman & Co., Ltd.

Gas Retorts.
West's Gas Improvement Co., Ltd.

Gaskets.
George Angus & Co., Ltd.
Cork Manufacturing Co., Ltd.
Wilkinson Rubber Linatex, Ltd.

Gasworks Plant.
Allan Kennedy & Co., Ltd.
West's Gas Improvements Co., Ltd.

Gauges.
James Chesterman, Ltd.
Drayton Regulator and Instrument Co., Ltd. (Draft)
Findlay & Co.
John Thompson (Wolverhampton), Ltd. (Steam and Water).

Gear Steels.
Clyde Alloy Steel Co., Ltd.

Gearing.
George Angus & Co., Ltd.
Butters Brothers & Co., Ltd.
Parsons Marine Steam Turbine Co., Ltd.

Gears (Hydraulic, Oil).
Keelavite Rotary Pumps & Motors, Ltd.
Vickers-Armstrongs, Ltd. (Variable Speed Gear Dept.).

Graphite.
Graphite Products, Ltd.

Grease Guns.
Tecalemit, Ltd.
Walters & Dobson, Ltd.

Grinders.
John Macdonald & Co. (Pneumatic Tools), Ltd.

Grinding Machines.
B.S.A. Tools, Ltd.
Fred Cam (Engineers), Ltd
A. C. Wickman, Ltd.

Gunmetal Ingots.
Manganese Bronze & Brass Co., Ltd.
McKechnie Brothers, Ltd.

Guns, Pressure.
Kleen-e-Ze Brush Co., Ltd. (tube cleaning)

Hacksaw Blades.
Richd. W. Carr & Co., Ltd.
English Steel Corporation, Ltd.
Edward G. Herbert, Ltd.
Frederick Pollard, Ltd.
Sanderson Brothers & Newbould, Ltd.

Hacksaws.
Sanderson Brothers & Newbould, Ltd.

Hair Belting.
George Angus & Co., Ltd.
Bell's Asbestos and Engineering, Limited.
J. H. Fenner & Co., Ltd.

Hammers, Electric Portable.
B.S.A. Tools, Ltd.

Hammers, Forging.
Alldays & Onions, Ltd.
Brett's Patent Lifter Co., Ltd.
Fred Cam (Engineers), Ltd.
Walters & Dobson, Ltd.

Hammers, Pneumatic.
Alldays & Onions, Ltd.
Brett's Patent Lifter Co., Ltd.
Fred Cam (Engineers), Ltd.
John Macdonald & Co. (Pneumatic Tools), Ltd.

Haulage Brakes.
M. B. Wild & Co., Ltd.

Heat Insulating Materials. *See* Insulating Materials.

Heat Treatment (Furnaces).
John Mathison, Ltd.

Heating and Ventilating Machinery.
Lamson Engineering Co., Ltd.
Charles McNeil, Ltd.

Hoists.
Tecalemit, Ltd.
United States Metallic Packing Co., Ltd.

Hose Pipes and Fittings.
George Angus & Co., Ltd.
Bell's Asbestos and Engineering, Limited.
Wilkinson Rubber Linatex, Ltd.

Hydraulic Leathers.
Thomas A. Ashton, Ltd.

Hydraulic Machine Tools. *See* Machine Tools.

Hydraulic Packing.
Ronald Trist & Co., Ltd.

Hydraulic Pumps. *See* Pumps.

Hydrogen Plant.
International Electrolytic Plant Co., Ltd.

India Rubber, Gutta Percha & Bitumen.
George Angus & Co., Ltd.

Indicators, Gas.
International Gas Detectors, Ltd.

Indicators for Steam, Gas and Oil Engines.
John Thompson (Wolverhampton), Ltd.

Ingot Metals,
Manganese Bronze & Brass Co., Ltd.
McKechnie Brothers, Ltd. (Non-Ferrous).

Injectors.
Green & Boulding, Ltd.
Newman, Hender & Co., Ltd.

Insulating Materials (Electrical).
Bakelite, Ltd.
Craigpark Electric Cable Co., Ltd.

Insulating Materials (Heat).
Bell's Asbestos and Engineering, Limited.
Cementation Co., Ltd.
Chemical & Insulating Co., Ltd.
Fibreglass, Ltd.
General Refractories, Ltd.
William Kenyon & Sons, Ltd.
John G. Stein & Co., Ltd.

Insulating Materials (Sound).
Bell's Asbestos and Engineering, Limited
Cementation Co., Ltd.
Fibreglass, Ltd.

Iron and Steel (Bars, Sheets and Sections).
F. Braby & Co., Ltd. (Sheets).
Hall Brothers (West Bromwich), Ltd.

For Advertisers' Addresses and positions of their announcements, consult the following Alphabetical List, pages 1403 to 1408.

Iron Castings.
International Meehanite Metal Co., Ltd.
Marshall Sons & Co., Ltd
S. Russell & Sons, Ltd.
Stanton Ironworks Co., Ltd.

Iron and Steel Manufacturers.
Allan Kennedy & Co., Ltd.
Stanton Ironworks Co., Ltd. (Pig and Refined Iron)

Irrigation Equipment.
Glenfield & Kennedy, Ltd.

Jigs.
John Cooper & Son, Ltd.
Charles Taylor (Birmingham), Ltd.

Jointing Compounds.
George Angus & Co., Ltd.
Bell's Asbestos and Engineering, Limited.
Graphite Products, Ltd.

Jointing, Expansion.
George Angus & Co., Ltd.
Ruberoid Co., Ltd.

Jointing Materials.
George Angus & Co., Ltd.
Bell's Asbestos and Engineering, Limited.
Cork Manufacturing Co., Ltd.
R. Klinger, Ltd.
Turner Brothers Asbestos Co., Ltd
Wilkinson Rubber Linatex, Ltd.

Joints.
Cork Manufacturing Co., Ltd.
Stanton Ironworks Co., Ltd. (Flexible)
Ronald Trist & Co., Ltd.

Keys.
H. Fordsmith, Ltd. (Wheel).

Laminated Belting.
George Angus & Co., Ltd.
James Hendry, Ltd.

Lamps.
British Thomson-Houston Co., Ltd. (Electric).
A. C. Wells & Co., Ltd. (Industrial).

Lathes.
B.S.A. Tools, Ltd.
Charles Churchill & Co., Ltd.
Alfred Herbert, Ltd.
Martin Bros. (Machinery), Ltd.
Charles Taylor (Birmingham), Ltd.
A. C. Wickman, Ltd.

Lead Base Alloys.
McKechnie Brothers, Ltd.

Leather Belting.
George Angus & Co., Ltd.
Thomas A Ashton, Ltd.
Bell's Asbestos and Engineering, Limited.
J. H. Fenner & Co., Ltd.
James Hendry, Ltd.

Lifting and Hoisting Tackle.
Herbert Morris, Ltd.
Ritchie-Atlas Engineering Co., Ltd.

Lifts.
Elliston, Evans & Jackson, Ltd.
Herbert Morris, Ltd.
Tecalemit, Ltd.

Light Railway Equipment.
Wm. Bain & Co., Ltd.
Robert Hudson, Ltd.

Lighting Standards.
Poles, Ltd.
Stanton Ironworks Co., Ltd. (Spun Concrete).

Lime Sprayers.
A. C. Wells & Co., Ltd.

Lining (Rubber).
George Angus & Co., Ltd.
Wilkinson Rubber Linatex, Ltd.

Lockers.
F. Braby & Co., Ltd.
Constructors, Ltd. (Steel).
Steel Equipment Co., Ltd.

Locomotive Packing. *See* PACKING.

Locomotive Superheaters. *See* SUPERHEATERS.

Locomotives.
Robert Hudson, Ltd. (Diesel and Steam).

Lubricants.
Bell's Asbestos and Engineering, Limited.
Graphite Products, Ltd.
W. B. Harrison.
Rotherham & Sons, Ltd.
Tecalemit, Ltd.

Lubricating Equipment.
Rotherham & Sons, Ltd.
Tecalemit, Ltd.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT THE FOLLOWING ALPHABETICAL LIST, PAGES 1403 TO 1408.

Machine Tools.
James Archdale & Co., Ltd.
Charles Churchill & Co., Ltd.
Crosthwaite Furnaces & Scriven Machine Tools, Ltd. (Hydraulic).
Findlay & Co
Alfred Herbert, Ltd.
Edward G. Herbert, Ltd.
Martin Bros. (Machinery), Ltd.
John Mathison, Ltd. (Charging machines).
Frederick Pollard, Ltd.
Norman E. Potts, Ltd.
S. Russell & Sons, Ltd.
Charles Taylor (Birmingham), Ltd.
Timbrell & Wright, Ltd.
Walters & Dobson, Ltd.
Thos. W. Ward, Ltd.
A. C. Wickman, Ltd.

Machined Components and Assemblies.
C. S. Madan & Co., Ltd.

Machinery Guards.
F. Braby & Co., Ltd.
F. W. Potter & Soar, Ltd.
Wm. Riddell, Cousland & Co., Ltd.

Machinery Merchants.
Thomas A. Ashton, Ltd.
B.S.A. Tools, Ltd.
Martin Bros. (Machinery), Ltd.
Thos. W. Ward, Ltd.

Magnetic Separators. *See* Separators.

Magnets.
Electromagnets, Ltd.
English Steel Corporation, Ltd.
W. L. Marrison, Ltd.

Mallets, Raw Hide.
George Angus & Co., Ltd.

Manganese Bronze.
Bull's Metal & Melloid Co., Ltd.
Manganese Bronze & Brass Co., Ltd.
McKechnie Brothers, Ltd. (Bars, Sections and Ingots).

Marine Boilers. *See* Boilers.

Marine Engines. *See* Engines.

Measuring Instruments.
Palatine Engineering Co., Ltd.
Tecalemit, Ltd.
A. C. Wickman, Ltd.

Measuring Tapes.
James Chesterman & Co., Ltd.

Mechanical Stokers. *See* Stokers.

Meehanite Metals.
International Meehanite Metal Co., Ltd.

Metal Pressings.
McKechnie Brothers, Ltd. (Non-Ferrous).

Metal Sawing and Cutting Machines.
Edward G. Herbert, Ltd.
Martin Bros. (Machinery), Ltd.
Charles Taylor (Birmingham), Ltd.

Metal Windows.
F. Braby & Co., Ltd.
John Thompson (Wolverhampton), Ltd. (Sashes and Casements).

Metallic Packing.
George Angus & Co., Ltd.
Green & Boulding, Ltd.
U.K. Anti-Friction Metallic Packing (Railway Signal Co., Ltd.)
United States Metallic Packing Co., Ltd.

Metals, Non-Ferrous.
James Booth & Co., Ltd.
British Aluminium Co., Ltd.
Delta Metal Co., Ltd.
Fry's Metal Foundries, Ltd.
Manganese Bronze & Brass Co., Ltd.
McKechnie Brothers, Ltd.

Metals, Powdered.
Bound Brook Bearings (G.B.), Ltd.

Meters.
Glenfield & Kennedy, Ltd. (Water).
International Gas Detectors, Ltd.
Palatine Engineering Co., Ltd. (Water).
Tecalemit, Ltd.

Micrometers.
Moore & Wright (Sheffield), Ltd.

Milling Cutters.
Richd. W. Carr & Co., Ltd.
English Steel Corporation, Ltd.

Milling Machines.
Fred Cam (Engineers), Ltd.
Martin Bros. (Machinery), Ltd.
Norman E. Potts, Ltd.
A. C. Wickman, Ltd.

Mills, Ball and Pebble.
Wilkinson Rubber Linatex, Ltd.

Mineral Borers & Well Sinkers.
Alexander Munro & Co. (Johnstone), Ltd.

Motor Car Accessories.
George Angus & Co., Ltd.
British Thermostat Co., Ltd. (Thermostats).
Rotherham & Sons, Ltd.
Skefko Ball Bearing Co., Ltd.
Tecalemit, Ltd.
Walters & Dobson, Ltd.

Motors, Electric.
James Beresford & Son, Ltd.
Ransomes, Sims & Jefferies, Ltd.

Moulding Machines.
Constructional Engineering Co., Ltd.

Name Plates.
Eyre & Baxter.

Noise Abatement Equipment.
Cementation Co., Ltd.
Fibreglass, Ltd.

Numbering Machines.
E. M. Richford, Ltd. (Automatic).

Oil Burners.
Clyde Fuel Systems, Ltd.
Richardsons Westgarth & Co., Ltd.
Wallsend Slipway & Engineering Co., Ltd.

Oil Coolers.
Heenan & Froude, Ltd.
Premier Cooler & Engineering Co., Ltd.

Oil Eliminators.
Filtrators, Ltd.
Permutit Co., Ltd.

Oil Engines. *See* ENGINES.

Oil Filters. *See* FILTERS.

Oil Furnaces. *See* FURNACES.

Oil Pipe Lines.
Stanton Ironworks Co., Ltd.

Oil Pumps. *See* PUMPS.

Oil Reclaiming.
Vokes, Ltd.

Oil Seals.
Ronald Trist & Co., Ltd.

Oilcans.
A. E. Westwood, Ltd.

Oils, Lubricating.
W. B. Harrison.

Oxy-Acetylene Welding and Cutting.
Acme Welding and Constructional Engineering Co., Ltd.
Arkinstall Brothers, Ltd.
Barimar, Ltd.
Ritchie-Atlas Engineering Co., Ltd.

Oxygen Resuscitating Apparatus.
Siebe, Gorman & Co., Ltd.
Sparklets, Ltd.

Packings.
George Angus & Co., Ltd.
Cork Manufacturing Co., Ltd.
Graphite Products, Ltd.
Ronald Trist & Co., Ltd. (Hydraulic and Steam).
Turner Brothers Asbestos Co., Ltd.
U.K. Anti-Friction Metallic Packing (Railway Signal Co., Ltd.)
United States Metallic Packing Co., Ltd.

Paints.
British Paints, Ltd.
Graphite Products, Ltd.
W. B. Harrison.

Papers (Waterproof).
Ruberoid Co., Ltd.

Partitions.
Constructors, Ltd. (Steel).

Perforated Metals.
F. Braby & Co., Ltd.
Robert Riley, Ltd.

Petrol Engines. *See* ENGINES.

Phosphor Bronze.
John Broadfoot & Sons, Ltd.
Manganese Bronze & Brass Co., Ltd.
McKechnie Brothers, Ltd. (Bars and Ingots).

Piles and Piling.
Cementation Co., Ltd.

Pinions.
George Angus & Co., Ltd. (Raw Hide).

Pins.
H. Fordsmith, Ltd. (Taper).

Pipe Coverings.
Bell's Asbestos and Engineering, Limited.
Chemical & Insulating Co., Ltd.
Fibreglass, Ltd.

Pipes and Fittings.
Thomas A. Ashton, Ltd.
British Aluminium Co., Ltd.
Le Bas Tube Co., Ltd.
Stanton Ironworks Co., Ltd. (Cast Iron, Spun Iron and Spun Concrete).

Pipework.
Brightside Foundry & Engineering Co., Ltd.
Le Bas Tube Co., Ltd.
John Thompson (Wolverhampton), Ltd.

Piston Packing.
George Angus & Co., Ltd.
Bell's Asbestos and Engineering, Limited.

Plastic Materials.
Bakelite, Ltd.
Wilkinson Rubber Linatex, Ltd.

Plywood.
H. Morris & Co., Ltd.

Pneumatic Conveyors.
Lamson Engineering Co., Ltd.

Pneumatic Dispatch Tubes.
Lamson Engineering Co., Ltd.

Pneumatic Tools.
John Macdonald & Co. (Pneumatic Tools), Ltd.

Points and Crossings.
Robert Hudson, Ltd.
Jas. Young (Contractors), Ltd.

Poles.
Poles, Ltd. (Steel Transmission, Telegraph).

Polishing Machines.
Walters and Dobson, Ltd.

Powdered Metal Products.
Bound Brook Bearings (G.B.), Ltd.

Power Presses.
Brett's Patent Lifter Co., Ltd.
Norman E. Potts, Ltd.
Walters & Dobson, Ltd.

Power Transmission Equipment.
George Angus & Co., Ltd.
Thomas A. Ashton, Ltd.
Hoffmann Manufacturing Co., Ltd.
William Kenyon & Sons, Ltd.
Ransome & Marles Bearing Co., Ltd.
Skefko Ball Bearing Co., Ltd.
Vickers-Armstrongs, Ltd. (Variable Speed Gear Dept.).

Precision Tools.
Charles Churchill & Co., Ltd.
Moore & Wright (Sheffield), Ltd.

Press Tools.
John Cooper & Son, Ltd.

Presses.
Joshua Bigwood & Son, Ltd. (Friction Screw)
Brett's Patent Lifter Co., Ltd.
Brightside Foundry & Engineering Co., Ltd.
Walters & Dobson, Ltd. (Friction Screw)

Pressings. Steel.
Barr. Thomson & Co., Ltd.
Charles McNeil, Ltd.

Pressure Controls.
British Thermostat Co., Ltd.
Crosby Valve & Engineering Co., Ltd.

Presswork.
F. S. Ratcliffe (Rochdale), Ltd.

Producer Gas Plants.
Incandescent Heat Co., Ltd.

Propellers.
Bull's Metal & Melloid Co., Ltd. (Ships).
Manganese Bronze & Brass Co., Ltd.
H. Morris & Co., Ltd.
Palatine Engineering Co., Ltd.

Protective Clothing.
George Angus & Co., Ltd.
Bell's Asbestos and Engineering, Limited.

Publications.
"The Chemist & Druggist."
"The Chemist and Druggist Export Review."
"Electronic Engineering."
"The Engineer."
"The Engineer" Directory.
"Historical Research."
"The Ironmonger."
Morgan Brothers (Publishers), Ltd.

Pulleys.
J. H. Fenner & Co., Ltd.
Herbert Morris, Ltd.

Pulverised Fuel Equipment.
International Combustion, Ltd.

Pulverisers.
Clarke, Chapman & Co., Ltd.

Pump Governors.
Copes Regulators, Ltd.

Pump Lining (Rubber).
George Angus & Co., Ltd.

Pump Oilcans.
A. E. Westwood, Ltd.

For Advertisers' Addresses and positions of their announcements, consult the following Alphabetical List, pages 1403 to 1408.

Pumps and Pumping Machinery.
James Beresford & Son, Ltd.
F. W. Brackett & Co., Ltd.
Clarke, Chapman & Co., Ltd.
Davie & Horne, Ltd. (Circulating).
Glenfield & Kennedy, Ltd.
Keelavite Rotary Pumps & Motors, Ltd.
Pulsometer Engineering Co., Ltd.
Towler Brothers (Patents), Ltd.
G. & J. Weir, Ltd.
M. B. Wild & Co., Ltd.
Wilkinson Rubber Linatex, Ltd.
Zwicky, Ltd.

Pumps, Air.
Blairs, Ltd.
Pulsometer Engineering Co., Ltd.
Rotherham & Sons, Ltd.
G. & J. Weir, Ltd.

Pumps, Centrifugal.
Pulsometer Engineering Co., Ltd.
G. & J. Weir, Ltd.
Wilkinson Rubber Linatex, Ltd.

Pumps, Chemical.
Pulsometer Engineering Co., Ltd.
Wilkinson Rubber Linatex, Ltd.

Pumps, Hydraulic.
James Beresford & Son., Ltd.
Brightside Foundry & Engineering Co., Ltd.
Glenfield & Kennedy. Ltd.
Keelavite Rotary Pumps & Motors Ltd.
C. S. Madan & Co., Ltd.
Pulsometer Engineering Co., Ltd.
Towler Brothers (Patents), Ltd.
Vickers-Armstrongs, Ltd. (Variable Speed Gear Dept.).
G. & J. Weir, Ltd.

Pumps, Oil.
James Beresford & Son, Ltd.
Clarke, Chapman & Co., Ltd.
Keelavite Rotary Pumps & Motors, Ltd.
Pulsometer Engineering Co., Ltd.
Rotherham & Sons, Ltd.
Tecalemit, Ltd.
Vickers-Armstrongs, Ltd. (Variable Speed Gear Dept.).
G. & J. Weir, Ltd.
A. E. Westwood, Ltd.
Charles Winn & Co., Ltd.
Zwicky, Ltd.

Pumps, Petrol.
James Beresford & Son, Ltd.
Pulsometer Engineering Co., Ltd.
Tecalemit, Ltd.
Zwicky, Ltd.

Pumps, Sludge.
James Beresford & Son, Ltd.
F. W. Brackett & Co., Ltd.
Clarke, Chapman & Co., Ltd.
Glenfield & Kennedy, Ltd.
Pulsometer Engineering Co., Ltd.
M. B. Wild & Co., Ltd.
Wilkinson Rubber Linatex, Ltd.

Pumps, Submersible.
J. Beresford & Son, Ltd.
Pulsometer Engineering Co., Ltd.

Pumps, Turbine.
James Beresford & Son, Ltd.
Pulsometer Engineering C
G. & J. Weir, Ltd.

Punches.
E. M. Richford, Ltd. (St

Pyrometers.
Amalgams Co., Ltd.

Railings (Tubular).
Bettles & Sons, Ltd.
Le Bas Tube Co., Ltd.
London & Midland Steel Scaffolding Co., Ltd.

Railway and Tramway Equipment.
James Beresford & Son, Ltd.
Craigpark Electric Cable Co., Ltd.
Robert Hudson, Ltd.
Walker & Wilson, Ltd.
Jas. Young (Contractors), Ltd. (Sidings)

Railway Signal Equipment.
Railway Signal Co., Ltd.

Raw Hide Hammers, Mallets and Pinions.
George Angus & Co., Ltd.

Reamers.
Richd. W. Carr & Co., Ltd.
John Macdonald & Co., (Pneumatic Tools) Ltd.

Recording Instruments.
Evershed & Vignoles, Ltd.
International Gas Detectors, Ltd.
Rotherham & Sons, Ltd.

Reducing Valves. *See* VALVES.

Reduction Gears.
George Angus & Co., Ltd.
Sanderson Brothers & Newbould, Ltd.

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT THE FOLLOWING ALPHABETICAL LIST, PAGES 1403 TO 1408.

Refractories.
General Refractories, Ltd.
John G. Stein & Co., Ltd. (Chrome and Magnesite).
West's Gas Improvements Co., Ltd.

Refractory Cement.
General Refractories, Ltd.
John G. Stein & Co., Ltd.

Refrigerating Machinery.
British Thermostat Co., Ltd. (Controls).
Peter Brotherhood, Ltd.
Pulsometer Engineering Co., Ltd. (Pumps).
G. & J. Weir, Ltd.

Refuse Disposal Equipment.
Heenan & Froude, Ltd.

Regulators (Automatic).
Copes Regulators, Ltd.
Drayton Regulator & Instrument Co., Ltd.
Ronald Trist & Co., Ltd.
G. & J. Weir, Ltd.

Riveting Machines.
Sir William Arrol & Co., Ltd.
John Macdonald & Co. (Pneumatic Tools), Ltd.
A. C. Wickman, Ltd.

Rivets.
British Aluminium Co., Ltd.

Roads (Iron).
Stanton Ironworks Co., Ltd.

Roller Chains.
Morse Chain Co., Ltd.

Roller Stamps.
E. M. Richford, Ltd.

Rolling Mills.
Brightside Foundry & Engineering Co., Ltd.

Roofing.
D. Anderson & Son, Ltd.
F. Braby & Co., Ltd.

Roofing Felts.
D. Anderson & Son, Ltd.
W. B. Harrison.
Ruberoid Co., Ltd.

Rope Dressing.
W. B. Harrison.
Thomas Hart, Ltd.

Rope Drives.
George Angus & Co., Ltd.
R. & J. Dick, Ltd.
Thomas Hart, Ltd.

Ropes (Steel Wire).
Latch & Batchelor, Ltd.
Wright's Ropes, Ltd.

Rubber.
George Angus & Co., Ltd.
Wilkinson Rubber Linatex, Ltd.

Rubber Stamps and Etching Equipment.
Eyre & Baxter.

Rubber Stamps.
E. M. Richford, Ltd.

Rules and Tapes.
James Chesterman & Co., Ltd.

Rust Preventives.
Bell's Asbestos and Engineering, Limited.

Salinometers (Electric).
W. Crockatt & Sons, Ltd.
Evershed & Vignoles, Ltd.

Saw Sharpeners.
Fred Cam (Engineers), Ltd.
S. Russell & Sons, Ltd.

Sawing and Cutting Machines (Metal).
S. Russell & Sons, Ltd.

Scaffolding (Steel).
London & Midland Steel Scaffolding Co., Ltd.

Schools and Colleges (Engineering). *See* COLLEGES.

Screwing Machines.
Charles Winn & Co., Ltd.

Screws.
Unbrako Socket Screw Co., Ltd.

Seals and Sealing Section.
British Thermostat Co., Ltd. (Packless shaft).
Wilkinson Rubber Linatex, Ltd.

Separators (Magnetic).
Electromagnets, Ltd.

Separators (Steam).
Green & Boulding, Ltd.
Superheater Company, Ltd.
United States Metallic Packing Co., Ltd

Shears.
Joshua Bigwood & Son, Ltd. (Guillotine).

FOR ADVERTISERS' ADDRESSES AND POSITIONS OF THEIR ANNOUNCEMENTS, CONSULT THE FOLLOWING ALPHABETICAL LIST, PAGES 1403 TO 1408.

Sheet Metal Work.
Acme Welding and Constructional Engineering Co., Ltd.
Arkinstall Brothers, Ltd.
F. Braby & Co., Ltd.
Lamson Engineering Co., Ltd.
F. W. Potter & Soar, Ltd.
F. S. Ratcliffe (Rochdale), Ltd.
Steel Equipment Co., Ltd.

Sheet Metal Working Machinery.
Keeton Sons & Co., Ltd.
A. C. Wickman. Ltd.

Sheets and Sections, Aluminium.
James Booth & Co., Ltd.
British Aluminium Co., Ltd.
Reynolds Tube Co., Ltd.

Sheets and Sections, Iron and Steel.
F. Braby & Co., Ltd.
Hall Brothers (West Bromwich), Ltd.

Shelving.
F. Braby & Co., Ltd.
Constructors, Ltd. (Steel).
Steel Equipment Co., Ltd.

Shims.
Willford & Co., Ltd. (Rail Joint).

Shipbuilders and Repairers.
Swan, Hunter & Wigham Richardson, Ltd.
Yarrow & Co., Ltd.

Silencers.
Vokes, Ltd.

Silica Bricks.
General Refractories, Ltd.
John G. Stein & Co., Ltd.

Skylights.
F. Braby & Co., Ltd.

Slag.
Stanton Ironworks Co., Ltd.

Sleepers.
Stanton Ironworks Co., Ltd. (Concrete).

Slings (for Lifting).
Thomas Hart, Ltd.

Sludge Pumps. *See* PUMPS.

Sluice Valves. *See* VALVES.

Smoke Helmets.
Siebe, Gorman & Co., Ltd.

Socket Screws.
Unbrako Socket Screw Co., Ltd.

Soldering Fluxes.
Fry's Metal Foundries, Ltd.

Solders.
Fry's Metal Foundries, Ltd.

Soot Blowers.
Clyde Blowers, Ltd.

Sound Insulation. *See* INSULATION.

Spring Making Machinery.
Fred Cam (Engineers), Ltd.

Spring Steel.
Effingham Steel Works, Ltd.

Springs.
Cockburns, Ltd.
English Steel Corporation, Ltd.
F. S. Ratcliffe (Rochdale), Ltd.
Robert Riley, Ltd.
Willford & Co., Ltd. (Laminated and Coil).

Stainless Steel.
Sanderson Brothers & Newbould, Ltd
Stainless Steel Wire Co., Ltd. (Wire)
Wardlows, Ltd.

Stair Treads.
George Angus & Co., Ltd.
British Aluminium Co., Ltd.
Ferodo, Ltd.
Allan Kennedy & Co., Ltd.
McKechnie Brothers, Ltd. (Brass).

Staircases (Steel).
Bettles & Sons, Ltd.
F. Braby & Co., Ltd.
Allan Kennedy & Co., Ltd.

Stampings (Steel).
English Steel Corporation, Ltd.
Kirkstall Forge, Ltd.
F. S. Ratcliffe (Rochdale), Ltd.

Stamps.
Brett's Patent Lifter Co., Ltd. (Friction and Steam Drop).
Eyre & Baxter (Letters, Figures and Name) (Steel).
Walters & Dobson, Ltd.

Standpipes.
Barr, Thomson & Co., Ltd.

Steam Accumulators.
Cochran & Co., Annan, Ltd.
Richardsons Westgarth & Co., Ltd.

Steam Engines. *See* ENGINES.

Steam Fittings.
Bell's Asbestos and Engineering, Limited
Cockburns, Ltd.
Newman, Hender & Co., Ltd.
John Thompson (Wolverhampton), Ltd.

Steam Gauges.
Bell's Asbestos and Engineering, Limited.
Newman, Hender & Co., Ltd.
John Thompson (Wolverhampton), Ltd.

Steam Packing.
George Angus & Co., Ltd.
Bell's Asbestos and Engineering, Limited.
British Metallic Packing Co. (1933), Ltd.
Ronald Trist & Co., Ltd.
United States Metallic Packing Co., Ltd.

Steam Pipes.
Le Bas Tube Co., Ltd.
John Thompson (Wolverhampton), Ltd.
Yarrow & Co., Ltd.

Steam Separators. *See* Separators.

Steam Traps.
Bell's Asbestos and Engineering, Limited.
Drayton Regulator & Instrument Co., Ltd.
Newman, Hender & Co., Ltd.
John Thompson (Wolverhampton), Ltd.
United States Metallic Packing Co., Ltd.

Steam Turbines.
British Thomson-Houston Co., Ltd.
Peter Brotherhood, Ltd.
Wallsend Slipway & Engineering Co., Ltd.
G. & J. Weir, Ltd.
Yarrow & Co., Ltd.

Steel Annealing.
Annealers, Ltd.

Steel Bars, Sheets, and Sections.
Hall Brothers (West Bromwich), Ltd.
Kirkstall Forge, Ltd. (Bars and Sections).
W. Wesson & Co., Ltd. (Bright drawn).

Steel Constructional Work.
Sir William Arrol & Co., Ltd.
Wm. Bain & Co., Ltd.
Bettles & Sons, Ltd.
F. Braby & Co., Ltd.
Allan Kennedy & Co., Ltd.
London & Midland Steel Scaffolding Co., Ltd.
S. Russell & Sons, Ltd.
Steel Equipment Co., Ltd. (Shelving).

Steel Makers.
Richd. W. Carr & Co., Ltd.
Clyde Alloy Steel Co., Ltd.
Dunford & Elliott (Sheffield), Ltd.
Effingham Steel Works, Ltd.
English Steel Corporation, Ltd.
W. L. Marrison, Ltd.
Wardlows, Ltd.

Steel Rollers.
English Steel Corporation, Ltd.

Steel Ropes.
Latch & Batchelor, Ltd.
Wrights' Ropes, Ltd.

Steel Sections (Cold Drawn).
Kirkstall Forge, Ltd.

Steel Stamps.
E. M. Richford, Ltd.

Steel Strip.
Effingham Steel Works, Ltd.
W. Wesson & Co., Ltd. (Hot Rolled and Bright Cold Rolled).

Steel Tubes and Fittings.
Le Bas Tube Co., Ltd.
Reynolds Tube Co., Ltd.

Steels, Heat Resisting.
Clyde Alloy Steel Co., Ltd.

Steels, High-Speed.
J. Beardshaw & Son, Ltd.
Richd. W. Carr & Co., Ltd.
Dunford & Elliott (Sheffield), Ltd.
English Steel Corporation, Ltd.
W. L. Marrison, Ltd.
Wardlows, Ltd.

Stencil Plates.
Eyre & Baxter.
E. M. Richford, Ltd.

Stokers, Coal.
Clyde Fuel Systems, Ltd.

Stokers, Mechanical.
Ashwell & Nesbit, Ltd.
Bennis Combustion, Ltd.
Crosthwaite Furnaces & Scriven Machine Tools, Ltd.
International Combustion, Ltd.
John Thompson (Wolverhampton), Ltd.

Storage Equipment.
F. Braby & Co., Ltd.
Constructors, Ltd. (Steel).
Steel Equipment Co., Ltd. (Bins).

Street Lighting Columns.
Poles, Ltd.

Stretchers, Hydraulic.
Joshua Bigwood & Son, Ltd. (Sheet and Bar).

Sugar Machinery.
Blairs, Ltd.

Superheaters.
Superheater Company, Ltd.
John Thompson (Wolverhampton), Ltd.
Yarrow & Co., Ltd.

For Advertisers' Addresses and positions of their announcements, consult the following Alphabetical List, pages 1403 to 1408.

Surveyor's Chains and Verniers.
James Chesterman & Co., Ltd.

Swarf Collecting Plant (Pneumatic).
Dallow Lambert & Co., Ltd.

Switches.
Drayton Regulator & Instrument Co., Ltd. (Electric).

Switchgear.
British Thomson-Houston Co., Ltd.

Tanks and Cisterns.
Arkinstall Brothers, Ltd.
F. Braby & Co., Ltd.

Telegraph Engineers.
Craigpark Electric Cable Co., Ltd.

Temperature Controls and Regulators.
British Thermostat Co., Ltd.
Crosby Valve & Engineering Co., Ltd.
Drayton Regulator & Instrument Co., Ltd.

Testing Machines.
Edward G. Herbert, Ltd.

Thermometers.
British Thermostat Co., Ltd.
Drayton Regulator & Instrument Co., Ltd.

Thermostatic Valves.
British Thermostat Co., Ltd.
Drayton Regulator & Instrument Co., Ltd.

Thermostats.
British Thermostat Co., Ltd.
Drayton Regulator & Instrument Co., Ltd.

Thrust Bearings.
Hoffman Manufacturing Co., Ltd.
Pollard Bearings, Ltd.
Ransome & Marles Bearing Co., Ltd.
Skefko Ball Bearing Co., Ltd.

Tipping Wagons.
Robert Hudson, Ltd. (Rail).

Tool Makers.
English Steel Corporation, Ltd.

Tool Steel.
Richd. W. Carr & Co., Ltd.
Clyde Alloy Steel Co., Ltd.
English Steel Corporation, Ltd.

Tools and Fittings.
Findlay & Co.
Timbrell & Wright, Ltd.

Tractors.
Marshall Sons & Co., Ltd. (Shunting).

Transformers, Electric.
British Thomson-Houston Co. Ltd.

Transmission Equipment.
George Angus & Co., Ltd.
Thomas A. Ashton, Ltd.
Thomas Hart, Ltd.
Pollard Bearings, Ltd. (Ball, Roller and Flexible).

Transmission (Hydraulic, Oil).
Keelavite Rotary Pumps & Motors, Ltd
Towler Brothers (Patents), Ltd.
Vickers-Armstrongs, Ltd. (Variable Speed Gear Dept.).

Traps, Steam. *See* STEAM TRAPS.

Trucks.
Brett's Patent Lifter Co., Ltd.
Herbert Morris, Ltd.
Ransomes, Sims & Jefferies, Ltd.

Tube Burnishers.
Kleen-e-Ze Brush Co., Ltd.

Tube Cleaners.
Clyde Blowers, Ltd.
W. Crockatt & Sons, Ltd.
Kleen-e-Ze Brush Co., Ltd.
Wm. Riddell, Cousland & Co., Ltd.

Tubes.
James Booth & Co., Ltd.
British Aluminium Co., Ltd.
Manganese Bronze & Brass Co., Ltd.
Reynolds Tube Co., Ltd.

Tubular Railings. *See* RAILINGS.

Tunnel Segments.
Stanton Ironworks, Ltd.

Turbine Pumps. *See* PUMPS.

Turbines.
British Thomson-Houston Co., Ltd.
Peter Brotherhood, Ltd.
Parsons Marine Steam Turbine Co., Ltd.
Wallsend Slipway & Engineering Co., Ltd.
G. & J. Weir, Ltd.
Yarrow & Co., Ltd.

Turbo-Generators.
British Thomson-Houston Co., Ltd.

Turntables.
Robert Hudson, Ltd.

Twist Drills. *See* DRILLS.

'V' Rope Drive.
George Angus & Co., Ltd.
Bell's Asbestos and Engineering, Limited.
R. & J. Dick, Ltd.
J. H. Fenner & Co., Ltd.

Vacuum Cleaning Plant.
Lamson Engineering Co., Ltd.

Valve Reseaters.
W. Crockatt & Sons, Ltd.
Charles Winn & Co., Ltd.

Valves and Fittings.
Barr, Thomson & Co., Ltd.
Bell's Asbestos and Engineering, Limited.
British Thermostat Co., Ltd. (Motorised).
John Broadfoot & Sons, Ltd.
Cockburns, Ltd.
Glenfield & Kennedy, Ltd.
R. Klinger, Ltd.
Newman, Hender & Co., Ltd.
Wilkinson Rubber Linatex, Ltd.

Valves, Reducing.
Glenfield & Kennedy, Ltd.
Newman, Hender & Co., Ltd.
Palatine Engineering Co., Ltd.
G. & J. Weir, Ltd.

Valves, Safety.
Glenfield & Kennedy, Ltd.
Newman, Hender & Co., Ltd.

Valves, Sluice.
Bell's Asbestos and Engineering, Limited.
Glenfield & Kennedy, Ltd.
Newman, Hender & Co., Ltd.
Charles Winn & Co., Ltd.

Variable Speed Gears.
George Angus & Co., Ltd.
Keelavite Rotary Pumps & Motors, Ltd.
Towler Bros. (Patents), Ltd.
Vickers-Armstrongs, Ltd. (Variable Speed Gear Dept.).

Varnishes.
Bakelite, Ltd.
W. B. Harrison.

Ventilating and Heating Engineers.
James Howden & Co., Ltd.
Lamson Engineering Co., Ltd.

Vices.
Charles Taylor (Birmingham), Ltd.

Voltmeters.
Evershed & Vignoles, Ltd.

Wagons.
Robert Hudson, Ltd. (Rail Tipping).

Washers.
Thomas A. Ashton, Ltd. (Leather and Fibre).
Positive Lock Washer Co., Ltd. (Lock and Spring).

Washing Plant (Car).
Tecalemit, Ltd.

Water Alarms (High and Low).
Ronald Trist & Co., Ltd.
G. & J. Weir, Ltd.

Water Coolers.
Peter Brotherhood, Ltd.
Heenan & Froude, Ltd.
Premier Cooler & Engineering Co., Ltd.

Water Demineralisation Apparatus.
Permutit Co., Ltd.

Water Distilling Apparatus.
G. & J. Weir, Ltd.

Water Ejectors.
James Beresford & Son, Ltd.

Water Level Indicators.
Evershed & Vignoles, Ltd.
R. Klinger, Ltd.
Newman, Hender & Co., Ltd.
Palatine Engineering Co., Ltd.
Charles Winn & Co., Ltd.

Water Screening Equipment.
F. W. Brackett & Co., Ltd.
Glenfield & Kennedy, Ltd.

Water Softening and Purifying Equipment.
Filtrators. Ltd.
Permutit Co., Ltd.
John Thompson (Wolverhampton), Ltd.

Water Treatment.
Albright & Wilson, Ltd.
Filtrators, Ltd.
Permutit Co., Ltd.

Water Tube Boilers.
Bennis Combustion, Ltd.
Clarke, Chapman & Co., Ltd.
Simon-Carves, Ltd.
Wallsend Slipway & Engineering Co., Ltd.

Waterworks Plant.
James Beresford & Son, Ltd.
Filtrators, Ltd.
Glenfield & Kennedy, Ltd.
Permutit Co., Ltd.

For Advertisers' Addresses and positions of their announcements, consult the following Alphabetical List, pages 1403 to 1408.

Welding.
Acme Welding and Constructional Engineering Co., Ltd.
Arkinstall Brothers, Ltd.
Barimar, Ltd. (and Welding Repairs).
Cementation Co., Ltd.
Murex Welding Processes, Ltd. (Equipment).

Wheels and Axles.
Robert Hudson, Ltd.

Winches.
Butters Brothers & Co., Ltd.
Clarke, Chapman & Co., Ltd.
M. B. Wild & Co., Ltd.

Winding and Hauling Equipment.
British Thomson-Houston Co., Ltd. (Electric).
Clarke, Chapman & Co., Ltd.
Robey & Co., Ltd.
M. B. Wild & Co., Ltd.
Wrights' Ropes, Ltd. (Ropes).

Wire Brushes.
Kleen-e-Ze Brush Co., Ltd.

Wire Drawing.
Wm. Bain & Co., Ltd.
Latch & Batchelor, Ltd.

Wire and Strip Forming Machines.
Heenan & Froude, Ltd.

Wire Manufacturers.
British Aluminium Co., Ltd.
Craigpark Electric Cable Co., Ltd. (Insulated).
Latch & Batchelor, Ltd.
F. W. Potter & Soar, Ltd. (Woven).
Wm. Riddell, Cousland & Co., Ltd. (Woven and Gauze).
Sparklets, Ltd. (Woven).
Stainless Steel Wire Co., Ltd.
Whitecross Co., Ltd.

Wire Ropes.
Latch & Batchelor, Ltd.
Whitecross Co., Ltd.
Wrights' Ropes, Ltd.

Wireless Masts and Towers.
Poles, Ltd.

Wirework.
Wm. Bain & Co., Ltd.
F. W. Potter & Soar, Ltd.
F. S. Ratcliffe (Rochdale), Ltd.

Wood Preservatives.
D. Anderson & Son, Ltd.
W. B. Harrison.

Woodworking Machinery.
Norman E. Potts, Ltd.

Works Equipment.
Constructors, Ltd. (Steel).

Wrought Ironwork.
F. W. Potter & Soar, Ltd.

Yellow Metal Rods.
Manganese Bronze & Brass Co., Ltd.
McKechnie Brothers, Ltd. (Extruded).

THE ENGINEER'S YEAR-BOOK ALPHABETICAL DIRECTORY OF ADVERTISERS' NAMES, WITH ADDRESSES, 1949.

PAGE

Acme Welding & Constructional Engineering Co., Ltd., Carlyle Avenue, Hillington, Glasgow, S.W. 2 ... ... *grey page in front of book* xii (I)
Albright & Wilson, Ltd., Oldbury, Nr. Birmingham ... ... *facing page* 48 (II)
Alldays & Onions, Ltd., Great Western Works, Small Heath, Birmingham, 11 *grey page in front of book* xxvi (I)
Allott Bros. & Leigh, Ltd., Ickles Forge, Rotherham ... ... *facing page* 44 (II)
Aluminium Union, Ltd., The Adelphi, London, W.C. 2 ... ... *facing page* 1206 (I)
Amalgams Co., Ltd., 186 Attercliffe Road, Sheffield ... ... *facing page* 1137 (I)
Anderson, D., & Sons, Ltd., Park Road Works, Stratford, Manchester *facing pages* 302 (I), 303 (I)
Angus, Geo., & Co., Ltd., Newcastle-upon-Tyne ... ... *grey page in front of book* lxxiv (I)
Archdale, James, & Co., Ltd., Ledsam Street, Birmingham, 16 ... ... *facing page* 1019 (I)
Arkinstall Bros., Ltd., Battery Works, Coventry Street, Birmingham, 5 *facing page* 3 (II) *and grey page in front of book* lx (I)
Arrol, Sir William, & Co., Ltd., 85 Dunn Street, Bridgeton, Glasgow ... *facing page* 477 (I)
Ashton, Thomas A., Speedwell Works, Sheffield ... ... *facing page* 906 (I)
Ashwell & Nesbit, Ltd., Barkby Lane, Leicester ... ... *facing page* 1336 (I)

Bain, Wm. & Co., Ltd., Lochrin Iron Works, Coatbridge ... ... *facing page* 211 (I)
Bakelite, Ltd., 18 Grosvenor Gardens, London, S.W. 1 ... ... *facing page* 988 (II)
Barimar, Ltd., Barimar House, 22-24 Peterborough Road, Fulham, London, S.W. 6 *grey page in front of book* lii (I)
Barr, Thomson & Co., Ltd., Netherton Iron Works, Kilmarnock *grey page in front of book* lxii (I)
Bell's Asbestos and Engineering Ltd., Bestobell Works, Slough ... ... *facing page* 1120 (II)
Bennis Combustion, Ltd., Little Hulton, Bolton, Lancs. ... *grey page in front of book* xv (I)
Beresford, James, & Son, Ltd., Mackadown Lane, Marston Green, Birmingham *facing page* 824 (I)
Bettles & Sons, Ltd., Stafford Road West, Fordhouses, Wolverhampton *grey page in front of book* xx (I)
Bigwood, Joshua, & Son, Ltd., Wednesfield Road, Wolverhampton ... *facing page* 1026 (I)
B.K.L. Alloys, Ltd., Factory Centre, Kings Norton, Birmingham, 30 *grey page in front of book* xxx (I)
Blairs, Ltd., Glasgow Engineering Works, Woodville Street, Govan, Glasgow, S.W. 1 *grey page in front of book* xl (I)
Booth, James, & Co., Ltd., Argyle Street Works, Nechells, Birmingham *inside front cover* i (I), ii (I)
Bound Brook Bearings (G.B.), Ltd., Birch Road, Witton, Birmingham, 6 *facing page* 862 (I)
Braby, Fredk., & Co., Ltd. 352-364 Euston Road, London, N.W. 1 *Buyers' Guide Section* (II)
Brackett, F. W., & Co., Ltd., Hythe Bridge Ironworks, Colchester *grey page in front of book* xxxiii (I)
Brett's Patent Lifter Co., Ltd., Foleshill Works, Coventry ... ... *facing page* 1120 (I)
Brightside Foundry and Engineering Co., Ltd., Sheffield ... ... *facing page* 675 (II)
British Aluminium Co., Ltd. (The), Salisbury House, London Wall, E.C. 2 *facing page* 1207 (I)
British Paints, Ltd., Portland Road, Newcastle-upon-Tyne ... ... *facing page* 55 (II), 977 (II)
British Thermostat Co., Ltd., Teddington Works, Windmill Road, Sunbury-on-Thames *facing page* 659 (II)

PAGE

British Thomson-Houston Co., Ltd., Rugby ... ... *grey page in front of book* xxxi (I)
British Timken, Ltd., Cheston Road, Aston, Birmingham, 7 ... ... *facing page* 853 (I)
Broadfoot, John, & Sons, Ltd., Ferryden Street, Whiteinch, Glasgow *grey page in front of book* lxxvii (I)
Brotherhood, Peter, Ltd., Peterborough ... ... *facing pages* 290 (II), 754 (II)
B.S.A. Tools, Ltd., Marston Green, Birmingham ... ... *grey page in front of book* xvi (I)
Bull's Metal & Melloid Co., Ltd., Yoker, Glasgow, W. 14 ... ... *facing page* 1175 (II)
Butters Brothers & Co., Ltd., McLellan Street, Glasgow ... ... *facing page* 804 (II)

Cam, Fred (Engineers), Ltd., 5 John Street, Sheffield, 2 ... *grey page in front of book* iv (I)
Carr, Richard W., & Co., Ltd., Pluto Works, Wadsley Bridge, Sheffield *facing page* 1159 (I)
Cementation Co., Ltd., Bentley Works, Doncaster ... ... *facing page* 381 (I)
Chatburn & Chantry, Ltd., 290 Penistone Road, Sheffield ... ... *facing page* 1168 (I)
Chemical & Insulating Co., Ltd., Darlington ... ... *facing page* 68 (II)
'Chemist & Druggist,' The, 28 Essex Street, London, W.C. 2 ... ... *facing page* 919 (I)
Chesterman, James, & Co., Ltd., Eccleshall Road, Sheffield, 11 *grey page in front of book* lxxv (I)
Churchill, Charles, & Co., Ltd., Coventry Road, South Yardley, Birmingham, 25 *grey page in front of book* lxxi (I)
Clarke, Chapman & Co., Ltd., Victoria Works, Gateshead-on-Tyne *between pages* 14 (II), 15 (II)
Clyde Alloy Steel Co., Ltd., Craigneuk Works, Motherwell ... ... *facing page* 1160 (I)
Clyde Blowers, Ltd., Livingstone Road, Clydebank, Glasgow *grey page in front of book* xxii (I)
Clyde Fuel Systems, Ltd., Queen Elizabeth Avenue, Hillington, Glasgow, S.W. 2 *facing Preface* (I)
Cochran & Co., Annan, Ltd., Annan, Scotland ... ... *facing page* 13 (II)
Cockburns, Ltd., Clydesdale Engineering Works, Cardonald, Glasgow, S.W. 2 *grey page in front of book* xxviii (I)
Constructional Engineering Co., Ltd., Titan Works, Birmingham, 12 *grey page in front of book* xxix (I)
Constructors, Ltd., Tyburn Road, Erdington, Birmingham, 24 ... ... *inside back cover* (I)
Cooper, John, & Son, Ltd., 32 Garden Street, Sheffield ... ... *facing page* 433 (II)
Copes Regulators, Ltd., 9 Southampton Place, High Holborn, W.C. 1 ... *facing page* 15 (II)
Cork Manufacturing Co., Ltd., South Chingford, E. 4 ... ... *facing page* 9 (II)
Craigpark Electric Cable Co., Ltd., Flemington Street, Springburn, Glasgow *facing page* 1377 (I)
Crockatt, W., & Sons, Ltd., 62 Darnley Street, Glasgow, S. 1 *grey page in front of book* xxiii (I)
Crosby Valve & Engineering Co., Ltd., 251 Ealing Road, Wembley, Middlesex *grey page in front of book* xxv (I)
Crosthwaite Furnaces and Scriven Machine Tools, Ltd., York Street Ironworks, Leeds, 9 *facing Buyers' Guide Section* (II)

Dallow Lambert & Co., Ltd., Spalding Street, Leicester ... ... *facing page* 1097 (I)
Davie & Horne, Ltd., Johnstone, Glasgow ... ... *grey page in front of book* lix (I)
Delta Metal Company, Ltd., Delta Works, East Greenwich, London, S.E. 10 *facing page* 1215 (I)
Dick, R. & J., Ltd., Greenhead Works, Glasgow, S.E. *facing pages* 905 (I), 916 (I), 917 (I)
Drayton Regulator & Instrument Co., Ltd., West Drayton, Middlesex ... *facing page* 71 (II)
Dunford & Elliott (Sheffield), Ltd., Attercliffe Wharf Works, Sheffield, 9 *facing page* 1158 (I)

Effingham Steel Works, Ltd., Washford Road, Sheffield, 9 ... ... *facing page* 1196 (I)
'Electronic Engineering,' 28 Essex Street, Strand, London W.C. 2 ... *facing page* 86 (I)
Elliston, Evams & Jackson, Ltd., 24 Ray Street, London, E.C. 1 ... *facing pages* 1498 (I), 1499 (I)
'Engineer,' The, 28 Essex Street, Strand, London, W.C. 2 ... ... *facing page Buyers' Guide Section* (II)
English Steel Corporation, Ltd., Vickers Works, Sheffield ...
Evershed & Vignoles, Ltd., Acton Lane Works, Chiswick, W. 4 ... *facing page* 1449 (I)
Eyre & Baxter, 55 Brown Street, Sheffield ... ... *grey page in front of book* lxxvi (I)

Fenner, J. H., & Co., Ltd., Winewall Mill, Colne, Lancs. ... ... *facing page* 918 (I)
Ferodo, Ltd., Chapel-en-le-Frith, Cheshire ... ... *facing page* 907 (I)

PAGE

Fibreglass, Ltd., Ravenhead, St. Helen's, Lancs. ... ... *facing pages* 69 (I), 17 (II)
Findlay & Co., Broadway Chambers, Hammersmith, London, W. 6 ... *grey page in front of book* lvi (I)
Fordsmith, H., Ltd., Hadfield Street Works, Cornbrook, Manchester, 16 *grey page in front of book* lvii (I)
Fry's Metal Foundries, Ltd., Tandem Works, Merton Abbey, London, S.W. 19 *facing page* 1160 (I)

General Galvanizers, Ltd., Lever Street, Wolverhampton ... ... *facing page* 1190 (I)
General Refractories Ltd., Genefax Ho., Sheffield 10 ... ... *facing page* 375 (I)
Glasgow Royal Technical College, Glasgow ... ... *grey page in front of book* xlv (I)
Glenfield & Kennedy, Ltd., Kilmarnock ... ... *facing pages* 802 (I), 803 (I)
Gleniffer Engines, Ltd., Temple Works, Anniesland, Glasgow, W. 3 ... *facing page* 210 (II)
Graphite Products, Ltd., 52 Battersea Church Road, Battersea, London, S.W. 11 *grey page in front of book* x (I)
Green & Boulding, Ltd., 162A Dalston Lane, London, E. 8 ... ... *facing page* 54 (II)

Hall Brothers (West Bromwich) Ltd., Eagle Works, Greets Green, West Bromwich *grey page in front of book* liii (I)
Harrison, W. B., 6 Bridge Street, Sunderland ... ... *facing page* 851 (I)
Hart, Thomas, Ltd., Lambeth Rope Works, Blackburn ... ... *facing page* 924 (I)
Heenan & Froude, Ltd., Worcester ... ... *facing pages* 43 (I), 1350 (I)
Hendry, James, Ltd., Main Street, Bridgeton, Glasgow ... ... *between pages* 912 (I), 913 (I)
Herbert, Alfred, Ltd., Coventry ... ... *grey page in front of book* lxvi (I)
Herbert, Edward G., Ltd., Atlas Works, Levenshulme, Manchester ... *facing pages* 1020 (I), 1021 (I)
Heriot-Watt College, Edinburgh ... ... *grey page in front of book* xlix (I)
Hoffmann Manufacturing Co., Ltd., Chelmsford ... ... *facing page* 849 (I)
Howden, James, & Co., Ltd., 195 Scotland Street, Glasgow, C. 5 ... *facing pages* 1309 (I), 666 (II)
Hudson, Robert, Ltd., Raletrux House, Meadow Lane, Leeds ... ... *facing page* 199 (II)
Humphreys, J. H., & Sons, Blackriding Electrical Works, Werneth, Oldham *facing page* 1063 (I)

Incandescent Heat Co., Ltd. (The), British Mills, Cornwall Road, Smethwick, Birmingham ... ... *facing page* 1125 (I)
International Combustion, Ltd., 19 Woburn Place, London, W.C. 1 ... *facing page* 1338 (I)
International Gas Detectors, Ltd., Great Wilson Street, Leeds ... ... *facing page* 1337 (I)
International Meehanite Metal Co., Ltd., 66 Victoria Street, S.W. 1 ... *facing page* 1191 (I)
'Ironmonger,' The, 28 Essex Street, London, W.C. 2 ... ... *facing pages* 301 (I), 432 (II)

Keelavite Rotary Pumps & Motors, Ltd., Allesley, nr. Coventry ... *facing pages* 766 (I) 767 (I)
Keeton Sons & Co., Ltd., Royds Lane, Sheffield, 4 ... *grey page in front of book* lxviii (I)
Kennedy, Allan, & Co., Ltd., Maritime Street, Stockton-on-Tees ... *facing page* 306 (I)
Kenyon, Wm., & Sons, Ltd., Chapel Field Works, Dukinfield, nr. Manchester *facing page* 674 (II)
King's College, University of London, Strand, W.C. 2 ... *grey page in front of book* xlvii (I)
Kirkstall Forge, Ltd., Kirkstall, Leeds ... ... *grey page in front of book* xliv (I)
Kleen-e-Ze Brush Co., Ltd., Hanham, Bristol ... ... *facing page* 989 (II)
Klinger, Richard, Ltd., Klingerit Works, Sidcup, Kent ... ... *facing page* 69 (II)

Lamson Engineering Co., Ltd., 6–8 Hythe Road, Willesden Junction, London, N.W. 10 *facing page* 1302 (II)
Latch & Batchelor, Ltd., Hay Mills, Birmingham ... ... *facing pages* 932 (I), 933 (I)
Le Bas Tube Co., Ltd., Cyclops Works, Millwall, E. 14 ... ... *facing page* 667 (II)

FOR CLASSIFIED DIRECTORY OF ADVERTISERS' SPECIALITIES, SEE PP. 1381, &c.

PAGE

Liptak Furnace Arches, Ltd., 68 Victoria Street, London, S.W. 1 *grey pages in front of book* v (I), vi (I)
London & Midland Steel Scaffolding Co., Ltd. (The), Burwood Ho., Caxton Street, S.W. 1 ... ... *grey page in front of book* xxxiv (I)

Macdonald, John, & Co. (Pneumatic Tools), Ltd., 175 Shawbridge Street, Pollokshaws, Glasgow ... ... *grey page in front of book* lv (I)
Madan, Charles S., & Co., Ltd., Atlantic Street, Broadheath, Altrincham, Lancs. *grey page in front of book* lviii (I)
Manchester College of Technology, Sackville Street, Manchester, 1 *grey page in front of book* xlvi (I)
Manganese Bronze & Brass Co., Ltd., Handford Works, Ipswich ...*facing pages* 863 (I), 1226 (I), 1242 (I)
Marrison, W. L., Ltd., Bower Street, Sheffield ... ... *facing page* 1066 (I)
Marshall Sons & Co., Ltd., Gainsborough, Lincs. ... ... *facing page* 45 (II)
Martin Bros. (Machinery), Ltd., Cornbrook, Manchester ... *grey page in front of book* xix (I)
Mathison, John, Ltd., Hutton Hall, Guisborough ... ... *facing page* 1351 (I)
Mavitta Drafting Machines, Ltd., Park Road, Aston, Birmingham, 6 *grey page in front of book* lxiv (I)
Mavor & Coulson, Ltd., Bridgeton, Glasgow, S.E. ... ... *facing page* 894 (II)
McKechnie Bros., Ltd., Rotton Park Street, Birmingham, 16 ... ... *facing page* 1214 (I)
McNeil, Charles, Ltd., Kinning Park, Glasgow, S. 1 ... *grey page in front of book* lxx (I)
Moore & Wright (Sheffield), Ltd., Sheffield ... ... *facing page* 1023 (I)
Morgan Bros. (Publishers), Ltd., 28 Essex Street, London, W.C. 2 ... *facing page* 722 (I)
Morris, H., & Co., Ltd., Glasgow ... ... *grey page in front of book* lxxviii (I)
Morris, Herbert, Ltd., Loughborough ... ... *facing page* 801 (II)
Morse Chain Co., Ltd., Letchworth, Herts. ... ... *facing page* 963 (I)
Munro, Alexander, & Co. (Johnstone) Ltd., 1 Rankine Street, Johnstone *facing pages* 1304 (II), 1305 (II)

Newall, A. P., & Co., Ltd., Possilpark, Glasgow, N. ... ... *facing page* 85 (I)
Newman, Hender & Co., Ltd., Woodchester, Glos. ... ... *facing pages* 1252 (II), 1253 (II)

Palatine Engineering Co., Ltd., Hawthorne Road, Bootle, nr. Liverpool *grey page in front of book* xxiv (I)
Parsons Marine Steam Turbine Co., Ltd. (The), Turbinia Works, Wallsend-on-Tyne *facing page* 291 (II)
Permutit Company, Ltd., Permutit House, Gunnersbury Avenue, W. *facing page* 49 (II)
Poles Ltd., Tyburn Road, Erdington, Birmingham ... ...*facing pages* 1410 (I), 1411 (I)
Pollard Bearings, Ltd., Tyburn Road, Erdington, Birmingham ... ... *facing page* 855 (I)
Pollard, Fredk. & Co., Ltd., St. Saviour's Road East, Leicester *grey page in front of book* xli (I)
Positive Lock Washer Co., Ltd., 78 McAlpine Street, Glasgow *grey page in front of book* xiv (I)
Potter, F. W., & Soar, Ltd., Phipp Street, Great Eastern Street, London, E.C. 2 *grey page in front of book* xi (I)
Potts, Norman E., Ltd., 130 Moseley Road, Camp Hill, Birmingham, 12 *grey page in front of book* l (I)
Premier Cooler and Engineering Co., Ltd., Shalford, nr. Guildford, Surrey *facing page* 182 (II)
Pulsometer Engineering Co., Ltd., Nine Elms Ironworks, Reading, Berks. *facing page* 810 (I)

Railway Signal Co., Ltd. (The), Fazakerley, Liverpool, 9 *grey page in front of book* xxxii (I)
Ransome & Marles Bearing Co., Ltd., Neeark-on-Trent ... ... *facing page* 852 (I)
Ransomes, Sims & Jefferies, Ltd., Orwell Works, Ipswich *grey page in front of book* xliii (I)
Ratcliffe, F. S. (Rochdale), Ltd., Crawford Spring Works, Rochdale *grey page in front of book* li (I)
Renold & Coventry Chain Company, Ltd., Renold Works, Manchester ... *facing page* 962 (I)
Reynolds Rolling Mills, Ltd., Tyseley, Birmingham ... ...*facing pages* 414 (II), 415 (II)
Richardson Westgarth & Co., Ltd., Hartlepool ... ... *facing page* 183 (II)

ALPHABETICAL DIRECTORY OF ADVERTISERS.

PAGE

Richford, E. M., Ltd., 8 & 9 Snow Hill, London, E.C. 1 ... ... *inside back cover* (I)
Riddell, Cousland, Wm., & Co., Ltd., Springfield Wire Works, Glasgow, S.E. *grey page in front of book* lxxiii (I)
Riley, Robert, Ltd., Milkstone Spring Works, Rochdale, Lancs. *grey pages in front of book* vii (I), viii (I)
Riley Stokers Co., Ltd., 19 Woburn Place, W.C. 1 ... ... *facing page* 1339 (I)
Ritchie-Atlas Engineering Co., Ltd., Atlas Works, Temple, Glasgow, W. 3 *grey page in front of book* lxxii (I)
Robey & Co., Ltd., Globe Works, Lincoln ... ... *facing page* 84 (II)
Rotherham & Sons, Ltd., Coventry ... ... *grey page in front of book* lxv (I)
Royal Technical College, Glasgow ... ... *grey page in front of book* xlv (I)
Ruberoid Company, Ltd., Commonwealth House, 1–19 New Oxford Street, London, W.C. 1 ... ... *facing page* 300 (I)
Russell Newbery & Co., Ltd., Essex Works, Dagenham ... ... *facing page* 211 (II)
Russell, S., & Sons, Ltd., Bath Lane, Leicester ... ... *grey page in front of book* liv (I)

Sanderson Bros., & Newbould, Ltd., Sheffield ... ... *grey page in front of book* xlii (I)
Scriven Machine Tools, Ltd., York Street, Leeds, 9 ... *facing Buyers' Guide section* (II)
Scottish Cables, Ltd., Deanside, Renfrew ... ... *facing pages* 1416 (I), 1417 (I)
Siebe, Gorman & Co., Ltd., Davis Road, Tolworth, Surrey ... ... *facing page* 605 (I)
Simon-Carves, Ltd., Cheadle Heath, Stockport ... ... *facing page* 12 (II)
Skefko Ball Bearing Co., Ltd., Luton, Beds. ... ... *bookmark*
Sparklets Ltd., Upper Edmonton, N. 18 ... ... *facing pages* 1261 (I), 1180 (II)
Stainless Steel Wire Co., Ltd., The Barracks, Langsett Road, Sheffield, 6 *facing page* 87 (I)
Stanton Ironworks Co., Ltd., nr. Nottingham ... ... *facing page* 723 (I)
Steel Equipment Co., Ltd., Greets Green, West Bromwich *grey page in front of book* ix (I)
Stein, John G., & Co., Ltd., Bonnybridge, Scotland ... ... *facing page* 363 (I)
Superheater Company, Ltd., 53 Haymarket, London, S.W. 1 ... ... *facing page* 16 (II)
Swain, J., & Son, Ltd., 2–4 Noel Street, W. 1 ... ... *grey page in front of book* lxvii (I)
Swan, Hunter & Wigham Richardson, Ltd., Wallsend-on-Tyne ... ... *facing page* 628 (I)

Taylor, Charles (Birmingham), Ltd., Bartholomew Street, Birmingham *grey page in front of book* xxxvii (I)
Tecalemit, Ltd., Great West Road, Brentford, Middlesex *grey page in front of book* xxxvi (I)
Technological Institute of Great Britain, 234 Temple Bar House, London, E.C. 4 *grey page in front of book* xlviii (I)
Thompson (John), Wolverhampton, Ltd., Wolverhampton ... *Buyers' Guide section* (II)
Timbrell & Wright, Ltd., Slaney Street, Birmingham, 4 ... ... *facing page* 1022 (I)
Towler Brothers (Patents), Ltd., Rodley, nr. Leeds ... *grey page in front of book* lxiii (I)
Trist, Ronald, & Co., Ltd., Bath Road, Slough, Bucks. ... *grey page in front of book* lxxix (I)
Turner Bros. Asbestos Co., Ltd., Rochdale, Lancs ... ... *facing page* 1 (II)

Unbrako Socket Screw Co., Ltd., Burnaby Road, Coventry *grey page in front of book* xvii (I)
United Kingdom Self-adjusting Anti-friction Metallic Packing Synd., Ltd., Fazakerley, Liverpool, 9 ... ... *grey page in front of book* xxxii (I)
United States Metallic Packing Co., Ltd., Soho Works, Allerton Road, Bradford *facing page* 70 (II)

Vickers-Armstrongs, Ltd. (Variable Speed Gear Dept.), Elswick Works, Newcastle-upon-Tyne ... ... *facing pages* 1114 (II), 1115 (II)
Vokes, Ltd., Guildford, Surrey ... ... *between pages* 242 (II), 243 (II)

Walker & Wilson, Ltd., Imperial Works, Hamilton, Lanarkshire *grey page in front of book* xxvii
Wallsend Slipway & Engineering Co., Ltd., Wallsend-on-Tyne ... ... *facing page* 298
Walters & Dobson, Ltd., Bailey Street, Sheffield, 1 ... *grey page in front of book* ii
Ward, Thos. W., Ltd., Albion Works, Savile Street, Sheffield ... ... *facing Contents*
Wardlows, Ltd., Carlisle Street, Sheffield, 4 ... ... *grey page in front of book* xxxv
Weir, G. &. J., Ltd., Cathcart, Glasgow ... ... *facing page*

FOR CLASSIFIED DIRECTORY OF ADVERTISERS' SPECIALITIES, SEE PP. 138

	PAGE
Wellman Bibby Co., Ltd., Parnell House, Wilton Road, London, S.W. 1 *facing page*	243 (II)
Wells, A. C., & Co., Ltd., Providence Mill, Hyde, Cheshire *grey page in front of book*	xlii (I)
Wesson, W., & Co., Ltd., Moxley, nr. Wednesbury ... *grey page in front of book*	xviii (I)
West's Gas Improvement Co., Ltd., Albion Ironworks, Miles Platting, Manchester *facing page*	615 (II)
Westwood, A. E., Ltd., Wesco Works, Hall Green, Brimingham, 28 *grey page in front of book*	xxxv (I)
Whitecross Co., Ltd., Warrington ... ... *...facing page*	944 (I)
Wickman, A. C., Ltd., Banner Lane, Tile Hill, Coventry ... *...facing page*	1027 (I)
and grey page in front of book	lxxi
Wild, M. B., & Co., Ltd., Argyle Street, Nechells, Birmingham, 7 *...facing page*	895 (II)
Wilkinson Rubber Linatex, Ltd., Frimley Road, Camberley, Surrey *grey page in front of book*	xxxix (I)
Williord & Co., Ltd., Park House Works, Sheffield, 9 ... ... *facing inside back cover*	(I)
Winn, Charles, & Co., Ltd., Granville Street, Birmingham, 1 *grey page in front of book*	xxi (I) *and* lxi (I)
Wrights Ropes, Ltd., Birmingham ... ... *...facing page*	847 (II)
Yarrow & Co., Ltd., Scotstoun, Glasgow ... ... *...facing page*	14 (II)
Young, Jas. (Contractors), Ltd., 177 Blairbeth Road, Burnside by Rutherglen *facing page*	469 (II)
Zwicky Ltd., Slough, Bucks. ... ... *...facing page*	825 (I)

FOR CLASSIFIED DIRECTORY OF ADVERTISERS' SPECIALITIES, SEE PP. 1381, &c.

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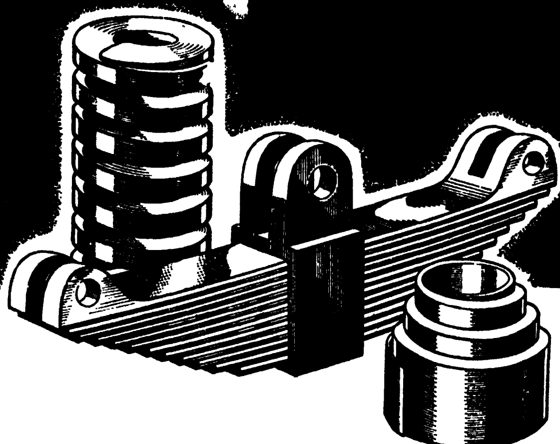

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